IN TWO SECTIONS—SECTION TWO

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of the

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Boiler Embrittlement

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In the introductory section of this paper, boiler embrittlement as a field of study is traced back toward the very invention and genesis of steam boilers. The study is traced through the miscellaneous researches of several countries up to the establishment of unified research in three institutions: Britain's National Physical Laboratory in 1917, America's University of Illinois in 1917, and Germany's Vereinigung der Grosskesselbesitzern in 1920.

In the second part of the paper, the chronological study is extended to date, but with emphasis on a critical evaluation of all published work which can be found from America, England, Germany, France, Sweden, Italy, and Russia. The roles of stress, caustic, chemical attack, "aging," and hydrogen are given especial attention along with all other possible factors in boiler embrittlement. These factors are then shown to be identical with those observed in the attack of steel by hydrogen at elevated temperatures, a subject given peculiar neglect in the literature on boiler embrittlement.

In the concluding section, the various factors involved in boiler embrittlement are interpreted from the viewpoint that "boiler embrittlement" is nothing other than an intermediate stage in a naturally occurring hydrogen-purification process. The arguments which have revolved around intercrystalline and transcrystalline paths of the fracture, the behavior of inhibitors and accelerators, the prerequisition of stress and of caustic, and the role played by the steel are discussed to show how logically all observations resolve themselves in the light of this new conception. This interpretation for boiler embrittlement is then corroborated visually by simple experiments made directly upon specimens from boiler plate which exhibited "boiler embrittlement." Plans for future research along entirely new lines are suggested.

SINCE the imperial Russian yacht *Livadia*, built by the British for the Czar in 1879, suffered a mishap with the steam boilers employed in its radical design, the study of embrittlement in boiler plate has commanded international attention. Yet, subsequent decades of investigation have succeeded principally in defining the defect known as "boiler embrittlement," while its nature and underlying cause remain essentially unsolved.

Indeed a glance at even recent literature on boiler embrittlement reveals disagreement on nearly all points. The use of sodium sulphate, for example, was adopted many years ago into the A.S.M.E. Boiler Code (185)[2] as a prerequisite for preventing embrittlement; yet the Symposium on Boiler Embrittlement held by the Society just recently found most of the contributors refuting everything previously claimed for Na_2SO_4 (405, 406, 410). Parr and Straub patented the use of phosphate as an inhibitor of boiler embrittlement (266); but Schroeder and Partridge report a case in which commercial trisodium phosphate apparently caused cracking (319). Similarly, embrittlement from NaOH depends upon the presence of Na_2SiO_3, according to Schroeder and Berk (315), and Straub and Bradbury (320); whereas Desch refutes that claim by embrittling steel with pure NaOH (361); and Rice mentions that Na_2SiO_3 may even inhibit embrittlement (397). Within one publication, Schroeder and Berk prove that nitrates are excellent inhibitors of caustic attack, and then proceed to reproduce the phenomenon of boiler embrittlement, in their opinion, with nitrate attack (398). The steel itself plays no role in boiler embrittlement, according to Parr and Straub (211); yet the entire boilermaking industry in Germany has been based upon metallurgical studies of the steel. Even the descriptions of the defect are at variance, principally in regard to the path which the failure follows, until Perry has ventured to remark that transcrystalline or intercrystalline characteristics can be selected at the will of the operator (311).

Careful study of the mass of literature on the subject appears to reveal the basis for this disagreement. Indeed, half a dozen industries comparable to boilermaking long ago met and solved what undoubtedly is the identical problem, as will be shown.

At this late date, the annual cost of "boiler embrittlement" to the railroads alone is $500,000 according to a recent estimate (375). Besides that value in dollars and cents, one must reckon the loss in steel replacements, disruption of war production, and danger to lives and property—important factors in the present emergency.

1 EARLY HISTORICAL REVIEW OF INVESTIGATIONS LEADING TO STUDY OF BOILER EMBRITTLEMENT

Genesis of Boilers

Hero of Alexandria, in his book "Pneumatica," written about 130 B.C. (1), describes what were probably the world's first steam boilers. One, an "aeolipile," was a primitive steam-reaction turbine; the other was a hollow altar containing air which, when heated by a fire kindled underneath, allowed the expanding air to force water from a vessel below into a suspended bucket, which in turn then descended, opening the doors of a shrine. When the fire was extinguished, the air contracted, the bucket emptied, and the doors closed.

Except for such primitive nonsense, credit for the first commercial employment of steam as a medium of making heat do work belongs to Capt. Thomas Savery in England in 1698. His boiler was simply a hollow oval vessel made of metal (12). Among subsequent names of historic stature stands Newcomen in 1705, and Watt in the latter part of the same century. Of historical interest to many a locomotive engineer is the name "Rocket," which was the first successful steam locomotive, built in 1829.

Usually made of cast iron, early boilers often had flat surfaces and leaden, or even wooden, tops. Hoboken's great engineer, Stevens, who, in 1805, invented the water-tube boiler (12) which is essentially the progenitor of the modern boiler, constructed a boiler with its upper part shaped like a truncated cone and made of wooden staves held together with iron hoops. Copper was sometimes used, especially for parts in contact with the fire. About 1720, a boiler in Cornwall supplying steam for a Newcomen engine was actually constructed of large granite slabs

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[2] Numbers in parentheses refer to Bibliography at end of paper.

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NOTE: Statements and opinions in papers are to be understood as individual expressions of their authors and not those of the Society.

(12). Little wonder that records from those days mention nothing of embrittlement!

Watt's "Wagon Boiler," so named because of its resemblance to the principal covered vehicle of that great era, introduced wrought iron as a structural material for steam boilers.

In 1857, steel was first used by the English Admiralty for the construction of marine boilers, but the results were far from satisfactory (7). To the *Bellerophon* goes the distinction of having the first inboard steel boiler. The commercial marine reported conflicting results, although land boilers made of Bessemer steel gave good service. Steelmakers were challenged to provide a more satisfactory material (3), which resulted in successful trials by the *Iris* and, in 1876, adoption of steel for steam boilers by the English Navy.

The use of steel then became generally adopted, although Parker (4), in 1878, stated that only two "of the modern form" were in existence in England at that time. Parker went on to give the first published comparison of advantages of steel over iron as a structural material for boilers, stressing the saving in thickness for any given pressure. In 1881, the same author noted that 1100 steel boilers were then in marine service (6).

Early Investigations of Explosions

In London, in 1815, a disastrous boiler explosion led to the appointment of a Parliamentary Committee, in 1817, to investigate the cause of that explosion and others which occurred about the same time. The recommendation of that learned body was that boilers thereafter should be made of wrought iron instead of cast iron or copper. Thereafter, boilers were periodically inspected by the Manchester Company, the pioneer inspection company of the world, from whom that practice subsequently spread to other countries (10).

Until 1830, boilers worked at about 7 psi, rarely over 10. The principle of high-pressure steam, then in its infancy, soon boosted the pressure to 30 or 40 psi, without bringing a simultaneous redesigning of the boilers. As a result, 288 explosions were reported in England between 1865 and 1870 (10).

In America, a parallel situation developed when the United States Treasury Department, in 1872, requested Thurston, President of this Society in 1881, to prepare a report on the causes and conditions of boiler explosions. Congress magnanimously appropriated $100,000 and the President was authorized to appoint a board to investigate "the singular mystery that has been supposed to surround their causes" (11). Thurston's comment then bridges the 60 years which have elapsed: "They (the Board) at once proceeded to spend money very freely—laid quite large plans; but for causes that need not be mentioned here, the expenditure of money was not as wisely made as it might have been!" His description of the futile efforts made to get various groups to co-operate in tracking down boiler failures casts a complimentary glow upon the contrasting success of the present Joint Research Committee on Boiler Feedwater Studies.

No report of that proposed investigation was ever published, though printing costs could hardly have exceeded the appropriation. Thurston discusses it, however (9, 11), and his hair-raising case histories of a dozen explosions, in which houses were demolished, locomotives turned into scrap, and bodies blown to hilltops a quarter mile away are quite sufficient to impress one that no effort made to solve and prevent boiler failures is unwarranted.

Thurston's conception, incidentally, was that a steam boiler is comparable to an explosive ordnance magazine, and he sought the answer to the explosion in the "expansive power common at the moment to the steam and to the water," which typified the general unawareness in that day of insidious processes within the steel, now called "boiler embrittlement." Most explosions, he concluded, were attributable to three causes; ignorance, carelessness, and utter recklessness.

Recognition of Brittleness in Boiler Steel During Fabrication

The Livadia. Prior to the *Livadia*, mentioned in the introduction, the study of boiler failures was primitive and largely nontechnical, resulting in conclusions to be compared with those of Thurston. Suspicions of the steel went no further than corrosion (6), and those failures wherein overheating, caused by local accumulation of dirt or sludge within the boiler, led to a craterlike eruption through the plastic metal (16).

The *Livadia* was an epochal event in yacht building, as well as in boiler making (5, 6, 8, 38, 59, 101). With a length of 230 ft, a beam of 150 ft, a double bottom, and triple sides forming two cell-layers each 6 ft through, this odd craft was to be fitted with ten boilers 14 ft 3 in. in diam; eight of them double-ended, 16 ft long, and two single-ended, $8^1/_2$ ft long. The boiler shells comprised three strakes of acid open-hearth $^3/_4$-in. steel plate with the lap joints treble-riveted, holes punched $^{11}/_{16}$ in. and reamed to $^3/_4$ in. The steel had passed Lloyd's tests, but one of the plates, after being punched, slipped out of the slings and developed cracks around the rivet holes (6, 101). It was assumed that punching had made the metal brittle, whereupon it was returned for annealing.

After completion of the boilers, they were tested hydrostatically. The first failed longitudinally at the joints of all three strakes. Another cracked before water ever entered it. Upon dismantling, the plates were found to be so brittle that pieces could be knocked off with a hammer.

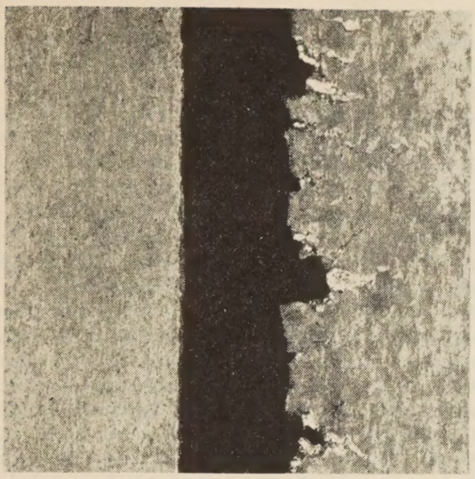
This event, then, turned the eyes of metallurgists toward the study of the steel itself; but only toward the type of brittleness which manifests itself during fabrication. Regardless of the possible relationship between this early trouble and the modern problem of "boiler embrittlement," it is important, for the present, to distinguish the two by pointing out that the cracking which concerns us today develops only after the steel has been exposed to service conditions.

At this date, but little finality can be read into the many explanations advanced then for this fabricational brittleness. On the other hand, our improved position in the science of metallurgy permits two rather significant conclusions to be drawn: (*a*) The steel of half a century ago contained excesses of impurities now known to lead to "aging," or loss of notch-impact toughness after cold-working; (*b*) brittleness from hydrogen was likely.

Because hydrogen is to be discussed in relation to contemporary troubles, the second conclusion warrants some expansion. In his extensive discussion of the *Livadia's* boilers, Parker (6) concludes that (*a*) ordinary chemical analyses of the steel reveal nothing; (*b*) further working to 50 per cent reduction before punching eliminates the trouble; and (*c*) the steel is sound until punched, when cracks develop for some distance around the holes.

When "ordinary chemical analyses reveal nothing," in relation to brittleness in steel, hydrogen is a prime wager even today. The second point argues against a role of "aging," which is *aggravated* by cold-working, and points directly to hydrogen; for hydrogen, and no other element, can be removed in critical quantities by cold-work (384, 387). His last point, although applicable to other types of brittleness, certainly applies to hydrogen brittleness, as demonstrated in Fig. 1. Two specimens of ordinary boiler steel were annealed in a laboratory furnace $^1/_2$ hr at 950 C (1742 F), one in H_2, and the other in air. When these plates were cold-deformed in a manner analogous to punching, severe cracking developed all along the edge of the hydrogen-treated specimen.

Belief that the *Livadia's* steel was similarly steeped in hydrogen



(a) (b)

Fig. 1 Edges of Two Specimens of Ordinary Boiler Steel, Cut From Same Sheet, Heated 1/2 Hr at 950 C, and Cold-Rolled to 50 Per Cent Reduction; ×3

(a, Annealed in air; b, annealed in hydrogen.)

during annealing can be based upon the known effect of a hydrogen anneal in removing nonmetallic impurities from the steel. Parker reports P and S diminished by one half near the surface; and C, which is most readily removed, was reduced to a trace. As Siemens, a famous metallurgist of that day, remarked in the discussion, such a condition cannot possibly be traced back to segregation in the ingot, but must have resulted from some subsequent treatment. One need not assume the annealing treatment was quite as severe as that in the experiment just cited; for, as Barnaby points out in the discussion, the failure was only one out of 1100 cases, a second set of plates causing no trouble at all. Treatment that severe would have caused 1100 failures out of 1100. Parker's remark that "absorption of gases in the annealing furnace" may have caused the embrittlement suggests that even they regarded hydrogen embrittlement as a serious prospect, although the defect was unknown then. He "intended to pursue the question of the possible absorption of gases" as "a fruitful source of inquiry;" and his observation that the embrittlement reduced the ductility without affecting the tensile strength is typical of hydrogen.

Incidentally, these "preservice failures," typified in Fig 1, are transcrystalline, i.e., through the grain, which distinguishes them again from the "service failures," or "boiler embrittlement," which are predominantly intercrystalline.

Recognition of Brittleness in Boiler Steel During Service: "Boiler Embrittlement"

In England. This transitional study of brittleness lasted for some years (37, 38, 39, 101), the discussion settling around brittleness observed during hydraulic-testing. Inconclusive comparisons of acid and basic open-hearth steels were made (38, 39). Annealing came under close scrutiny, because the opening of a crack indicated a role of internal stress (38). Steels which passed all tests of the British Board of Trade Surveyor and the Surveyor to Lloyd's Register would fail unexpectedly. Again, it is interesting to note that the fracture types discussed by Milton (38), and Arnold (37) are typical of those caused by hydrogen (404, 423).

According to Houghton (84), Milton wrote the first paper (38) on "boiler embrittlement" in 1905, wherein he noted that flanged steel plates in boiler service may crack, but that tensile and bend tests of the adjacent metal are entirely satisfactory. This fact stands as basic today (233, 398). The recognition of the fact that cracking might develop in service, rather than failure through the obvious means of corrosion or overheating, was the turning point in this field.

A growing pain of historic consequence in that regard was the famous case of the Dutch steamship, *Pahud*, an unclassed coasting vessel employed around Java, whose boiler exploded so violently that the boat sank (101). The boiler had been made in 1908, by a Dutch firm using basic open-hearth plates from a large German steelworks. German law required all plates to be annealed but, in this case, the plates were so large that the furnace equipment was not well suited. The boiler was of the triple-riveted, double-butt-strap type, and the rupture occurred along a row of rivets on a longitudinal joint. Butt straps were then removed from several similar boilers on other ships, and cracks between the rivets were found.

Wolff (97) blamed the failure upon fatigue, reporting an elaborate series of tests on the elastic movements of plates. He felt that short coastal service, i.e., "boiler-working," was responsible in developing intermittent and unequal expansions in the boiler shell, and cited the case of a Clyde-built vessel employed on the Yangtze. At the sudden approach of a typhoon, steam was raised rapidly; the shell plate split longitudinally, but was patched and thereafter served for 40 years.

Wolff also stated that the worst failures were from basic open-hearth steel, although acid steel fails, too. The fact that only in a few isolated cases did boilers fail, he felt, pointed to the steel as the source of the trouble. Houghton (101), however, thought the answer lay in the work of Ridsdale (24, 27), who had demonstrated that steel worked at blue heat develops cracks between the crystals near the surface. Service (108) remarked on the evident harmfulness of shearing; Bryden (100) blamed rolling, finishing at too low a temperature, and pointed out that the failures discussed by Houghton all occurred at the edges of the plates; Marquand (102) blamed overheating, or finishing at too high a temperature, as evidenced by the large grain size at the failure; McCance (103) showed that steel plate hardens to one half its thickness, during shearing whether sheared cold or at blue heat; Houghton (101) and Marquand (102) both hint at the bogy of "oxidized steel;" and Ridsdale (106) blames the weather, remarking that a cold floor on Monday morning, or snow and rain, if the plate is cooled outdoors, may lead to local brittleness. Houghton (101) quotes Colville as opining that the *Pahud's* boilers had been made from a "wild charge of steel dosed with silicon," which may be said to complete this shotgun pattern of logic.

Finally, Stead (109), who agreed with Wolff in his opinion on fatigue, obtained a "good" and a "bad" piece of steel, each passing specifications; but the "bad" steel, as a specimen 1 in. wide, would crack during one bend, whereas the "good" steel required numerous bends for breaking. The method of bending as a test for boiler plate was thereafter generally adopted.

In Germany. In Germany, these events were more or less paralleled, although the importance of annealing was recognized long before it was adopted elsewhere. Meunier, in his Report of the Chief Engineer to the Elsässischen Vereines von Dampfkesselbesitzern (28) for 1902, discussed the explosion of a boiler built in 1869, and proposed that the ductility loss occurs during service, in opposition to the general picture of that time. Meunier, therefore, antedates Milton in recognizing "boiler embrittlement." All boilers having 35 years of service should be replaced, in his opinion (35). Principally as a result of his work, the Reichs Chancellor issued a decree, known as the Würzburg and Hamburg Code (40) of 1905, which standardized boiler inspection both during fabrication and during service (41, 44).

Bach (33, 41, 42, 48, 49, 60, 65, 70) and Baumann (50, 70, 77, 157) were the authorities in Germany on boiler failures. It is an amusing commentary on the elusive nature of "boiler embrittlement" to find Baumann writing a monumental article 35 years ago on "The Present Status of the Question of Cracking in Boiler Plate" (50), in which he presumes to have the answer to boiler embrittlement. The same title was used almost verbatim only 3 years ago in Germany, and in substance composed recent papers in England (396) and America (297), as well as the present one, each one still claiming an answer.

From the start, German investigators suspected the steel. Sulzer (54) attempted to relate cracking to thermal strain by various physical tests. Bach (48) used specimens from a fractured boiler dome and tested them at various temperatures, emphasizing the necessity for designing boilers on the basis of the properties of the steel at the operating temperature (33). Bach (60, 70), Bauer (99), and Baumann (77) then showed that incipient cracks developed during riveting when too severe pressures were used, a line followed by recent investigators in this country (311). Boiler explosions (71) in Germany, in 1912, were blamed on improper heat-treatment of the steel (67, 167), and hot-working at too low temperatures (73). Bauer (99) and Baumann (77) also investigated segregations and inclusions as causes of cracking. Starck (184) blamed the brittle nature of ferrite crystals. Finally, the problem still unsolved, a disastrous boiler explosion in 1920, in Germany (283, 361), provoked the formation of the Vereinigung der Grosskesselbesitzern.

In America. Corresponding concerted study of boiler embrittlement began in England at the National Physical Laboratory (251, 265, 296, 361), in 1917, the first results of which were published by Rosenhain and Hanson (128), in 1920. In America, Parr pioneered present studies with a publication (94), also in 1917, which covered his work begun in 1912, as a result of boiler failures occurring around Urbana, Ill. (220, 233), and concluded with a research by Merica (92) on caustic embrittlement. Boilers had been installed in a building at the University of Illinois in 1874. Surface water was used until 1888, when a 160-ft well was dug. In 1898, operating pressures were increased and embrittlement began shortly, although the first official record occurred in 1910. First attempts, in 1912, to neutralize the alkalinity ended in 1915 with Parr's suggesting that sulphuric acid be added to the feedwater. Today, the work at the University of Illinois Engineering Experiment Station (92, 94, 181, 197, 200, 211, 223–226, 232–234, 243, 244, 254, 266, 320, 352, 353, 369, 416), at the United States Bureau of Mines (297, 300, 315–319, 331, 348–351, 380, 381, 398, 399, 411–413), and that of the Joint Research Committee on Boiler Feedwater Studies, represents America's principal contribution to the international problem of boiler embrittlement.

Everything prior to the authoritative investigations of these great international organizations for the study of boiler embrittlement, just named, may be considered as having historical interest only, although it serves well to orient the student of this subject. The following discussion, while also chronological and stepwise, will subordinate historical features to a critical examination of the manifold, and often divergent, data provided principally by those international organizations over the past 25 years.

2 CRITICAL STUDY OF PUBLISHED DATA ON BOILER EMBRITTLEMENT

Caustic Embrittlement

Today, the defect known as "boiler embrittlement," the subject of this paper, is still often called "caustic embrittlement" (326, 256, 259, 371, 378, 390), and perhaps correctly. In any event, the roots of present research certainly extend from the observation that caustic solutions develop a brittleness in steel which is characteristic of the defect which is now being considered.

Thomson (19), in 1894, and Stromeyer (59), in 1909, appear to be the first to discuss that action of caustic when they cited cases of boilers containing caustic in which embrittlement had proceeded to a degree where the rivet heads flew off. Stromeyer's explanation was that, in the presence of caustic, water reacts with the iron and forms iron oxide and hydrogen, "which leads to hydrogen embrittlement." In 1914, Andrew (81) enlarged upon the argument; but 1917 stands as the epochal year.

In that year, Desch, discussing Wolff's paper (97), described the embrittlement of caustic tanks and ventured the explanation that hydrogen entered the grain boundaries and "gradually opened up the metal until the crystals were distinct." To Desch, one of Britain's ablest scientists, therefore, goes the credit for anticipating by a quarter of a century the recent results of the National Physical Laboratory (396). Stromeyer (95, 96), in a classic experiment, proved that caustic embrittles steel only when stressed in tension. Two concentric, tapered rings were forced together, placing one under tension, the other under compression. Regardless of previous working or annealing, the compressed ring would not embrittle in caustic, whereas the ring in tension did. Stromeyer assumed that sodium carbonate in the boiler water breaks down to form NaOH under operating conditions of temperature and pressure, and that the NaOH caused the embrittlement (59, 95, 96, 150, 171). Furthermore, he noticed a "penetrating influence" of the caustic, which seemed to operate upon metal removed from the solution by $^1/_4$ in. of steel.

In the same year, Merica (92) and Parr (94) initiated the series of publications from the University of Illinois. Parr pointed out that normal feedwater has no NaOH and far too little carbonate to act on boiler plate, but that concentration is likely. Furthermore, depending upon temperature and the removal of CO_2, hydrolysis of the carbonate yields NaOH. Tests with steel, immersed in 13.6-normal NaOH at temperatures up to 280 C, clearly established the role of caustic in developing the hydrogen necessary for the hydrogen theory, whereupon Parr concluded, "the evidence seems to show that the embrittling effect of NaOH on steel is due to the evolution of hydrogen, and the absorption by the steel of the hydrogen in the nascent state." This statement, though similar to Desch's, missed similar fruitfulness for reasons soon to be disclosed. On the basis of the hydrogen theory, he suggested adding an oxidizing agent, such as $Na_2Cr_4O_7$, to depolarize the iron electrode. Sulphates, he added, might also be useful, at least as diluents. These ideas remain more or less secure today.

Certain basic facts regarding boiler embrittlement, recorded by these same investigators, are worthy of mention, namely, that embrittlement (*a*) always occurs below the water line; (*b*) is associated with waters having high alkalinity; (*c*) usually occurs in conjunction with a leak or seam where concentration of the soluble material to its saturation point is promoted; and (*d*) is accompanied by an exterior accumulation of a strongly alkaline incrustation.

In attempting to elucidate the role of hydrogen in causing boiler embrittlement, which is a phenomenon characterized by grain-boundary reactions at elevated temperatures, Parr and Merica drew an unfortunate comparison with the hydrogen-caused phenomenon of "pickling brittleness," widely discussed since the time of Ledebur (14, 15), and relating only to short-time tests and temperatures usually well below the boiling point of water. Discussion of that error will be reserved until its importance is established in succeeding paragraphs.

The Role of Stress

In the Intercrystalline Failure of Metals. As an alternative to the hydrogen theory, Parr (94) and Merica (92) had also suggested that the caustic ate out the "amorphous intercrystalline cement," a primitive metallurgical conception of grain-boundary material which is probably less stable than the body of the crystal. Andrew (81) had previously suggested such a thing, and Schroeder and Berk (398) base their entire argument upon essentially the same conception. Let us examine the broad aspects of intercrystalline failure from direct chemical attack to detect relevancy to boiler embrittlement, if any.

Since Roberts-Austen (13), in 1886, touched an Au-Cu-Ag alloy with $FeCl_3$ solution and found it breaking within a few seconds along several inches of the metal, many similar observations have been reported which may strongly influence one toward such an explanation for boiler embrittlement. Amagat (17), in 1893, forced mercury through 3 in. of cast steel at 3000 atm pressure, with indications of a selective and highly penetrative amalgamation. No previous flaw could be detected in the steel with a microscope. Cailletet, Colardeau, and Riviere (25) observed the same phenomenon at higher temperatures proceeding under relatively low pressures. Bridgman (66) finally discovered the role of stress, demonstrating that pressures exceeding the elastic limit would force mercury through the strongest steel.

Similarly, Desch found beta brass containing Al to granulate within a few seconds when placed in a solution containing a salt of mercury (82, 133); although cast Zn, Sn, and Pb corroded gradually without granulation (112). Copper cathodes in molten NaOH developed intercrystalline brittleness (127). Rawdon reports that mercury solutions for testing crack sensitivity of manganese bronze were adopted into the New York Board of Water Supply Specifications in 1915, and that the Aircraft Board adopted an immersion test of 15 min in mercurous-nitrate solution (105). Phosphor-bronze bars, immersed in mercurous-nitrate solution, cracked in 3 min, according to Arnott (131). LeChatelier complains of the disintegration of military canteens in the first world war (116). Made of an aluminum alloy, containing 3 per cent copper, the canteens disintegrated in service, showing oxidation around the grain boundaries. Again, copper develops intercrystalline brittleness when heated in Zn vapor (151); and Desch and White showed that electrolytic corrosion of beta brass is so pronounced at the grain boundaries that the zinc is removed, leaving an enrichment of copper, which may oxidize (79). Similarly, nickel steel has been reported to oxidize granularly, leaving an enrichment of nickel in the grain boundary (89).

Nitrate solutions are particularly effective, lead disintegrating into crystals when placed in dilute HNO_3 (119), or when used as an anode in nitric acid (86). Lead acetate has a similar effect on lead (129), although a single crystal is relatively immune. Porter reports $NaNO_3$ causing cracks within a few hours running the entire diameter of a 100-ft NaCl evaporating pan (146), an observation also made by Malcolm for smaller pans (141). Smialowski (288) and Andrews (335) have demonstrated the intercrystalline attack of steel by ammonium nitrate, Andrews proposing that grain-boundary carbide is the cause of the attack. The work on boiler embrittlement at the Bureau of Mines leans heavily upon this action of nitrates (398).

Among the first to discover the critical role of stress in these failures was Rogers, who claimed to have developed, in 1905, a mercuric-salt method for revealing internal stress in brasses (118). Smith (121) thought that Roberts-Austen's experiment with the gold alloy (13) confirmed the presence of a highly stressed surface film which led to fracture when fissured. Jones (140), in a paper much quoted in work from the Bureau of Mines, stated that service failures of steel from caustic clearly required stress. He proceeded to develop intercrystalline cracking in steel at temperatures up to 250 C, using $Ca(NO_3)_2$, NH_4NO_3, $NaNO_3$, $NaNO_3$ plus 5 and 10 per cent $NaHCO_3$, and KOH solutions, although neither $NaHCO_3$ alone, $(NH_3)_2SO_4$, $CaCl_2$, fused $NaNO_3$ + KNO_3 mixture, nor air would cause cracking. Around rivet holes, $Ca(NO_3)_2$ produced cracks in 28 days. In spite of the fact that some students of boiler embrittlement have attached great importance to this nitrate cracking, Jones admitted that the action of the KOH, which was the only nitrate-free solution with which he produced cracking, was unique; and he actually concluded that the caustic produces hydrogen which selectively weakens the grain boundaries. The nitrate, he thought, might supply an analogous agent, possibly nitrogen. Subsequent investigators, who have referred to Jones's work, never mention his own evaluation of his results, but emphasize the nitrate cracking, which is irrelevant to boiler embrittlement, as will soon appear.

In further proof of the importance of stress, Jonson found that brass immersed in ammonia and stressed in tension was unaffected until the elastic limit was reached, whereupon the metal fractured (85). Dickenson showed that molten tin and solder penetrate the grain boundaries of stressed brass (123); Merica found a difference in potential to exist between stressed and unstressed wires (87); Austin obtained intercrystalline embrittlement by stressing various metals in tension in solder (302); van Ewijk demonstrated the penetration of high-strength alloy steels by solders at 400 C (301), similar to Schuster's experiments with liquid brass on steel (269); Hodge and Miller showed intracrystalline penetration of austenitic "stainless" steel by the chlorides of iron and mercury (373). However, the most thorough research on the penetration of stressed steel by molten metals and alloys was reported by Schottky, Schichtel, and Stolle (242), who concluded that intercrystalline failure will result only if the stress is above some minimum, and only for those systems exhibiting solubility.

In "season cracking" of brass is found the widest discussion of intergranular cracking. "Season cracking" is one of those hapless terms which have no technical significance and simply pinch-hit until the phenomenon to which they refer is understood. In this case, the term comes from the resemblance of the defect to the cracks developing in timber upon seasoning.

With copper-zinc alloys, the defect is thought to be limited to metal having less than 80 per cent copper (98). Strain is essential, which lends a serious aspect to the problem because brass is principally valuable in the hard worked condition. Diegel, in 1906, appears to have been the first to discuss season cracking (43). Merica and Woodward reviewed the literature (93), up to 1917. Masterful papers on the subject include those of Hatfield and Thirkell (115), and Moore, Beckinsale, and Mallinson (144), both written more than 20 years ago. Both show the role of stress, the latter demonstrating the intercrystalline cracking produced by salts of Hg and NH_3 in brasses sufficiently stressed. Pure copper was not cracked by those salts, and only those brasses cracked in which Zn was greater than 6 per cent. Bassett (98), writing in an era not yet adapted to the automobile, cites instances of spun- or drawn-brass lamps on carriages breaking during winter storage in stables. Condensation of moisture during temperature changes plus absorption of ammonia from the stable atmosphere was indicated.

On the role of stress in causing cracking, Heyn's 1914 lecture before the Institute of Metals remains a classic (83). Heyn was the first to set forth clearly the underlying principles and to measure the stress. Incidentally, his method of measuring stress by cutting concentric rings from the specimen, although independently conceived, had been used 20 years previously by Howard at the Watertown Arsenal (18).

These examples have been cited because today there are those who attach great, but ambiguous, importance to the similarities with boiler embrittlement, probably following a suggestion made in 1919 by Rosenhain and Archbutt that boiler embrittlement was nothing other than the defect known as season cracking in brass, and as granular disintegration in lead (120). Such an opinion simply reverses the adage about the trees obscuring the forest, for the phenomenon of intercrystalline embrittlement in metal springs from a multitude of causes, none of the foregoing having any real resemblance to boiler embrittlement except that under the microscope the failure is intercrystalline in both cases. If comparison is valuable, and it certainly can be when correctly made, why not compare boiler embrittlement with the mass of untouched information on intercrystalline embrittlement and granular deterioration from hydrogen? Since the active presence of hydrogen in boiler embrittlement has been so amply proved, why not select for comparison the intergranular failure occurring in chambers for ammonia synthesis, hydrogenation autoclaves, in decarburizing steels with hydrogen, in general purification treatments using hydrogen, in the well-known "hydrogen sickness" of copper and its alloys; rather than with aqueous solutions containing chemicals having but little relationship to boiler feedwater? Why do investigators of boiler embrittlement continue to cite cases of intercrystalline-cracking from nitrates (398) when the NO_3^- ion is deliberately added because it is a good "inhibitor" of boiler embrittlement (319, 405)?

In "Boiler Embrittlement." Similarly, stress has been found to be essential to the intercrystalline-cracking known as boiler embrittlement (226), even to the extreme of Hatfield's suggestion (193) that the question of boiler-cracking was settled in 1898 when Stead showed fissures in the steel caused by machining and working (21, 22). Bauer early noticed that boiler-cracking was always preceded by cold-working, a condition difficult to avoid in processing boilers (90). Heyn and Bauer made an elaborate study of the stresses in boiler plate (68) and decided that tensile tests were poor indicators of suitability for boiler service (74). Impact tests were advised. A great deal of work in Germany followed that general line of thought as a result of Fry's development of an etching technique which brought out the strain lines in steel (149, 169, 179, 180, 183, 187, 196, 219). Hanson (136) and Rosenhain (148) noted that the outer plate and rivets of a 4-strake boiler embrittled worse than the inner plate and suggested that stress was consequently a more likely cause of boiler embrittlement than chemical attack. Sufficient reason for cracking could be found in internal stresses and cold-working, according to Whiteley (110, 204), Bolsover (216), Strauss and Fry (149), and Ness and MacCallum (210), although Houghton demurred (138). That corrosion only started at loci already initiated by excessive strain during cold-working was Fletcher's viewpoint (134). As a matter of fact, the argument at the 1921 session of the Faraday Society centered around the relative importance of stress and chemical attack in causing embrittlement, as did the 1925 meeting of the Vereinigung der Grosskesselbessitzern in Germany (183). McAdam concluded that boiler embrittlement was simply a pitting form of stress corrosion wherein hydrogen played no part and NaOH was unnecessary (240). Such a conclusion, of course, is ill-advised, for no traces of pitting are ever found (240, 319), NaOH is essential (181, 211, 233), and the defect depends upon hydrogen, as will be shown.

At the National Physical Laboratory, Rosenhain and Murphy (241) and Jenkins (280, 293) performed experiments in which specimens were stressed in various ways in various solutions at various temperatures, but no intercrystalline failure could be produced even in tests lasting 5 years. Tests in air at 300 C for 5 years also failed to reproduce boiler embrittlement (199), whereupon the safe assumption could then be made that stress alone is not the cause of that defect. Since that temperature lies in the range of "blue brittleness," the tests also eliminated the analogy originally proposed by Ridsdale (20).

To accentuate the specific nature of "boiler embrittlement," White and Schneidewind (245) outlined and illustrated the various types of cracking which follow from stress alone, stress in conjunction with corrosion, and from caustic attack. Of course, it must be recognized that all boiler failures are not caused by caustic embrittlement (23, 175, 261, 299). Surface cracks may even carry over from hot-working (239); and true corrosion and fatigue cracks are quite possible under boiler-operating conditions (181, 195, 220, 240, 312, 313). Nevertheless, the defect known as "boiler embrittlement" (135, 186, 213, 218, 255, 367, 368, 383, 397) is a specific phenomenon, whose cause clearly depends upon both stress and chemical attack.

Interestingly, research in Germany followed steadfastly the role of stress as it related to the properties of the steel (270, 272, 286, 354, 392), whereas Americans have emphasized the chemical aspect (198). In England, besides the researches already mentioned, some considered embrittlement to result simply from pre-existent intercrystalline spaces (124, 249), the so-called "Rose' Canals," which were proved to exist by the clever research of Tammann and Bredemeier (172), or from fissures developed during riveting (21, 22, 191, 193). Rosenhain and Hanson, noting carbonless bands and grain growth from critical amounts of cold-working, argued heatedly that they had at last found the answer to boiler embrittlement in large, brittle ferrite crystals (107).

Finally, Münzinger took a trip to America to examine our practice and wrote upon it at length in 1925, after returning to Germany. His message to the folks back home was that interesting results were being obtained in America from feedwater studies, the opinion being that the caustic concentrated in leaky seams (163, 165, 166). The fall meeting of the Vereinigung der Grosskesselbesitzern in that year concerned itself largely with Münzinger's report (183), whereupon that date may be said to mark the beginning of concerted international attention being paid to the combined effects of stress and chemical attack from NaOH.

The Role of Chemical Attack

Chemical attack, of course, must relate to chemicals in the feedwater. That is a principal weakness in the argument based upon nitrate-cracking when much more significant comparisons can be so readily found. In fact, Schroeder and his colleagues have frequently pointed out the well-known characteristics of boiler embrittlement which distinguish it from ordinary chemical attack, but provide, as their best answer, a theory based upon direct chemical attack. As mentioned by Parr and Straub (181), nitrates, chlorides, and sulphates may lead to corrosion because corrosion is associated with the H^+ ion; but the OH^- ion, which characterizes the feedwater of embrittled boilers, should inhibit such attack in boilers. That it does is borne out by fact, except for some extraordinary conditions in high-capacity boilers (365).

There is another well-established fact: namely, that boiler embrittlement is founded upon the reaction of iron with water to form hydrogen and black magnetic oxide

$$3\ Fe + 4\ H_2O = Fe_3O_4 + 8\ H$$

The oxide characteristically occurs as a smooth, adherent, black scale to be distinguished sharply from the highly localized accumulations one must expect from selective corrosion, as required by the theory of Schroeder and his colleagues. The rusting of iron in general, which they admit is the basis of the phenomenon, is notoriously undiscriminating in its attack; that is, it is non-

selective (137). Other objections to their argument will be treated as they develop in the discussion.

Because the role of feedwater and chemical attack has been so frequently discussed in this country, only a skeletonized review of essential points will be made here. In the monumental publication by Hall and his colleagues (192), and in the papers of Straub (223, 225), Chapman (236), and others (207, 231), the subject is considered in detail.

Briefly, caustic is kept in the water as a necessary prevention of dissolution of metal at any anodes which may develop, 100 ppm effectively inhibiting corrosion. If carbon dioxide escapes, the caustic may concentrate

$$Na_2CO_3 + H_2O = CO_2 + 2\,\underline{NaOH}$$

Even with this increase in NaOH, however, the solution is still harmless to steel. Further concentration is necessary, as early pointed out by Parr (94). In 1910, Stromeyer had reported iron tubes cracking in caustic evaporators when NaOH exceeded 40 per cent (63). In 1917, Parr had shown that iron did not react with NaOH solutions from $^1/_2$ to 2 normal, but that, at higher concentrations, 5 to 10 normal, the passivity vanished and the anodic reaction proceeded (94)

$$Fe + x\,OH = FeO_x + x\,H$$

At 150 to 200 C, however, H was generated in a 1-normal solution, a fact shown by Krassa (58) in 1909. Still, such concentrations never occur in boiler water; whereupon the assumption was made that the solution concentrated around rivets and in leaky seams to a point where the steel was attacked (181). That was the viewpoint taken back to Germany by Münzinger and discussed by Thiel, Kriegsheim, Ries, Rösing, and Baumann at the meeting of the Vereinigung der Grosskesselbesitzern (183), and elsewhere by Baumann (188), and by Splittgerber (170).

Demonstration of such concentration soon followed. Berl (189), in 1927, used glass capillaries to show that evaporation and the slow diffusibility of substances in such confined spaces quite readily led to concentration. His observation (190) is today generally accepted as factual (233), especially because the alkaline incrustation always noted around failed parts is virtual proof of the mechanism (233).

For the prevention of embrittlement, Parr's early ideas were based upon the hydrogen theory and concerned adding salts that would react with the NaOH or with the H to make it harmless. Such an argument is still valid today, although the studies have been advanced to include many other additions. Endless argument over the results which have been obtained are principally due to causes that will be considered after the discussion of another phase of the problem has provided a basis for an understanding.

The Role of "Aging"

No adequate discussion of boiler embrittlement can fail to mention the energetic figure of Stromeyer who, as early as 1905, made bending tests after successive intervals of time and found that the ductility of boiler steel decreased upon standing at room temperature (39). Although admitting that the brittleness may begin at sheared edges, Stromeyer nevertheless felt that the defect was more generally located and proposed that P was to blame when its content exceeded 0.05 per cent. The same thing is observed today (200). In 1907, Stromeyer (52, 53) proposed his famous "aging" theory for boiler embrittlement, which threw steelmakers into consternation because he effectively showed them that steel may be harmed by boiling in water. Tests made at temperatures from a wintry —16 F to that of boiling water and for periods up to 16 years showed that steel became brittle with age, and that increasing the temperature accelerated the process. In 1909, Stromeyer introduced N into the question of aging (59), along with impurities in general, although he was never able to provide an explanation for the effect that was either logical or decisive.

About 20 years later, Fry (178, 209) brought the "aging" argument to practical fruition with the development of a "nonaging" steel, "Izett" steel, so named from the initials of the German words "*immer zäh*," meaning "always tough." Fry believed, as did Stromeyer, that boiler steel aged into brittleness upon heating at 200 C. Subsequent to the time of Stromeyer, oxygen was identified as a likely cause of aging; consequently, Izett was a steel to which sufficient Al had been added so that the oxygen was stabilized as Al_2O_3.

Because oxygen may not be the only element responsible for aging, Izett steel need not be the final answer. Pfeil (212), for example, blames C; and N was considered along with P by Stromeyer (59). However, it is possible to coalesce these seemingly contradictory opinions and to show that each nonmetallic impurity may play a part.

Unfortunately, the valuable lesson to be learned from these investigations on aging was obscured by the research of Parr and Straub (211), in 1928. Although Parr and Straub themselves state that cracking will not occur if the steel is not stressed above the range of the yield point (181), a fact widely recognized, they tested a group of steels without regard to load/strength ratios, and concluded by comparing a flange steel, tested at its yield point, with a specimen of Izett having a 25 per cent overload. Even under these conditions, the Izett specimen was 200 to 300 per cent more durable than most of the fifteen specimens tested, and had a life exceeded only by two S.A.E. 1112 steels and the flange steel just mentioned. Neuendorff (252) also remarks that Parr and Straub's results are valueless because, "for reasons unknown to the reviewer," the flange steel had less load than the Izett. Those tests were made exceedingly critical to the problem of boiler embrittlement by the authors emphasizing the erroneous conclusion (211): "The embrittling effect occurs without regard to impurities or composition of the plate and that any defense of the embrittling distress attempting to charge it against the faulty iron is without basis in fact" (181), and "any ideas that embrittling distress may be due to faulty iron is without basis in fact" (200). Again, Straub remarks (200), "Pure iron was embrittled in the same manner as boiler plate." Obtaining "pure" iron is an accomplishment but few metallurgists yet dare boast; none would in 1927. In the same article, on the other hand, he observes that S and P accelerate embrittlement.

Along the same line, Parr and Straub (181) tested 3.5 per cent Ni steels, again without regard for load/strength ratios, and concluded that Ni exerts "no effect other than raising the yield point and, consequently, the initial stress necessary to cause embrittlement." On the contrary, examination of their data shows that the 3.5 Ni steel had 1000 per cent longer life than the best plain-carbon steel; that the best Ni steel had a yield point of only 40,000 psi and lasted $8^1/_2$ days, whereas six of the eight carbon steels had yield points above 40,000 psi and failed before 1 day had elapsed, one with a yield point of 49,000 psi failing in $3^1/_2$ hr. In the same vein, these investigators later (211) discounted the effect of sorbitic heat-treatment because it doesn't "stop" embrittlement, thereby losing for the science the fact that the finer dispersion of the carbide is definitely favorable.

Such work has been aptly categorized by Gillett (395) as "metallurgical naïveness," who with numerous instances reminds investigators of boiler embrittlement that they err when assuming that "steel is steel." Perry (311) is essentially in agreement with Gillett, and with metallurgists in general, when he states that the railroads find it "impossible to accept the prevailing ideas

regarding the relation of . . . boiler water to the failures," and that "factors other than boiler-water conditions contribute to boiler failures, and until such factors are corrected, failures are bound to occur regardless of the water used."

Izett steel, in spite of this, has been amply defended, though American investigators of boiler embrittlement seem poorly informed of it. Ulrich (227, 235) deliberately repeated Parr and Straub's tests, but with careful arrangement of the ratio of load-to-tensile strength so that an acceptable basis for comparison resulted. In addition, his values were corrected for the variation of the tensile strength between room temperature and 250 C (154, 201), which apparently was overlooked in the work of Parr and Straub. The Izett showed a marked superiority over flange steel in respect to embrittlement in caustic solution, a result also reached by the German Official Testing Laboratory (194). Finally, American investigators likewise reported recently that aluminum-killed steels are more resistant to cracking (398).

Comparing Bessemer, open-hearth Duplex, and Izett steel with respect to embrittlement from cold-working and from heating, Epstein (248) found that Izett suffered no increase in tensile strength and but little decrease in ductility upon heating to the blue-brittle range, so that it was considerably more ductile than boiler steel; and that aging around rivet holes was so shallow that the notch toughness was unaffected. Fry (238) pointed out that boiler steels, if nonaging, would promote welding practice for boilers. Studying boiler plate at room temperature after cold-working, Bauer found a progressive decrease from 6.3 to 2 in notch-impact values over a period of 1 year, with no end point indicated (132, 252). In a similar study, Goerens reported a great improvement from an addition of 3 per cent Ni (160). Lastly, Herzog and Portevin (342) demonstrated that nitrate solutions, generally so destructive to steels, do not readily fissure killed steels, which shows the necessity of considering the metallurgical quality of the steel in such problems.

The Role of Hydrogen

The Fe-H_2O Reaction as a Source of Hydrogen. Since Ledebur (15), in 1889, discussed rusting brittleness in iron along with pickling brittleness, the hydrogen produced by the reaction of Fe and H_2O has received universal attention. In 1903, Whitney (32) originated the electrochemical theory for the corrosion of iron by water, wherein H^+ ions are replaced by metal ions, according to the theory of Nernst

$$Fe + 2\,H^+ = Fe^{++} + 2\,\underline{H}$$

Walker and his colleagues (55, 56) extended the theory in 1907, and 2 years later Friend (57) measured the hydrogen produced by steam acting on iron foils and powders at temperatures up to 350 C. Friend showed that magnetic iron oxide resulted from the action, an observation substantiated by every case of boiler embrittlement (181), and presented curves showing that at the lower temperatures the reaction undoubtedly still proceeded, but more slowly.

In 1919, Fuller (114) published a research proving that tap water reacts with iron to produce hydrogen which diffuses through the iron, and that steam greatly aggravates the action. His observations are notable, because some have raised the question whether the nascent condition of the hydrogen in rusting really compares with the hydrogen developed during pickling, which is known to penetrate iron. It does, though the reaction is considerably slower.

Schikorr (221, 222) developed 18 atm pressure of hydrogen by reacting iron turnings with deaerated water without any limit being indicated. The evolution was doubled between 25 and 35 C, which is consistent with the logarithmic nature of chemical reactions. When an iron plate was used, a coating of Fe_3O_4 developed in 2 hr, but no measurable hydrogen was produced. This suggests that the gas may have been absorbed by the steel, as shown by Fuller (114). Powder has such a large ratio of surface to volume that reaction, rather than occlusion, would be favored. Baylis (176), and Berl and his co-workers (190, 206, 215, 229, 246), made similar observations, the latter reporting that NaOH added to the water in quantities above a certain minimum greatly increased the evolution of hydrogen.

In actual tests on boilers, Fellows found hydrogen both in saturated and in superheated steam, and in the water both as it entered and as it left the economizers (217). Fellows made the significant remark that a commercial process for producing H_2 is the steam-iron reaction. His attention, however, was centered upon the oxidizing phase of the reaction, and he did not note that his curve, showing increasing oxidation of the steel with increasing temperature, likewise represents increasing hydrogenizing, by virtue of the decomposition of H_2O, and the standard reaction

$$3\,Fe + 4\,H_2O = Fe_3O_4 + 8\,\underline{H}$$

That H_2 may be a predominating noncondensable gas in steam was demonstrated by Hall and his co-workers (192) with a laboratory boiler. Straub and Nelson (369) collected great amounts of H_2 from water and caustic solutions allowed to react with steel. In the same line come the researches of Potter, Solberg, and Hawkins (330, 414, 415), which were prompted by the fact that modern boiler installations are using temperatures as high as 1200 F, and pressures of the order of 1400 psi, truly simulating in severity the commerical process for making hydrogen, mentioned by Fellows. Again, however, those investigators began with their attention turned to the oxygen rather than the hydrogen, principally because they were studying scaling. That gas finally forced their attention by diffusing through the steel and vitiating their results, and again by unexpected quantities suddenly evolving from the steel. Their method of measuring oxidation was to employ measurement of the hydrogen which was produced, since it would be a chemical equivalent. As a result, those investigators provide some interesting facts attesting the prominence of hydrogen in steam boilers.

In another field, Dällenbach (277) reports that the penetration of hydrogen through the steel walls of mercury-arc rectifiers from the cooling water upsets their service (298).

Finally, Norton (377) reports a research establishing further the penetration of iron by hydrogen from tap water. What is of special significance to investigators of boiler embrittlement is the fact that sodium chromate, an inhibitor known since the days of Parr (94), stopped the diffusion of H; and that a sand-blasted steel tube coated with Na_2SiO_3, a great accelerator of embrittlement, developed so much hydrogen on the opposite surface during 15 min of heating at 200 C that the experiment had to be discontinued.

These researches can leave no doubt that in steam boilers an unlimited supply of hydrogen is available; furthermore, that that hydrogen develops under conditions favoring its diffusion into the steel.

Action of Hydrogen on Steel at Ordinary Temperatures—Transcrystalline Brittleness. One of the prolonged arguments in the literature on boiler embrittlement is that concerning the path taken by the fracture, i.e., is it around grains or through grains?

When steel is exposed to nascent hydrogen at ordinary temperatures, as in acid-pickling, cathodic electrolysis, and the rusting reaction, the hydrogen atoms enter the iron lattice and diffuse through the body of the grain; not through the grain boundaries

(321). Consequently, the deleterious effect of the gas at ordinary temperatures is *intra*granular (186), rather than intergranular. The gas may enter grain boundaries, of course, but its pressure there cannot exceed the pressure within the grain unless secondary reactions with impurities set in. Although Pfeil (182) and Williams and Homerberg (162) appear to have recorded such an instance at ordinary temperatures, such reactions are unlikely over a short period of time.

In previous publications from Battelle Memorial Institute, the action of hydrogen within the grain at ordinary temperatures has been sufficiently discussed (384, 386–389, 404, 418–420),[3] so that anything more than a summary statement here is unwarranted. Briefly, each grain of the steel is intrapenetrated with a vast, geometric system of ultramicroscopic imperfections. These are commonly observed when the metal is cold-deformed or etched. They may also be observed when the metal is stressed internally by precipitations such as the Widmannstätten action, or during nitriding, or carburizing, when the supersaturated phase may be seen to collect along the crystallographic planes that once contained minute voids. Those same imperfections collect hydrogen when it is a supersaturated phase; and it always is, practically speaking.

Now, hydrogen can only enter the iron lattice when H_2 is dissociated into atomic H. That is why pickling, cathodic electrolysis, and rusting are prolific sources of absorbable H, for the H atom actually meets the iron lattice before it meets another H atom with which to combine to form stable, insoluble H_2. In a gaseous-hydrogen atmosphere, only one out of approximately 10^{30} of the gas particles is an atom. Consequently, gaseous hydrogen can be stored in steel containers without measurable loss. Conversely, no one has yet conducted a pickling experiment under such great pressures of H_2 that the pickling action was reversed. Nor has any investigator of boiler embrittlement ever carried the bomb test to a point where the pressure of H_2 was sufficient to reverse the oxide-forming reaction.

By the same token, it must be conceded that the hydrogen atoms within the steel, when encountering the crystal imperfections just discussed, or any other discontinuity, will recombine in the direction of equilibrium, which, at room temperature lies far toward the side of tremendous pressures of the insoluble, molecular gas. Actual measurement of the pressures which may develop in that manner is not difficult. Bardenheuer and Thanheiser (205) obtained nearly 5000 psi, using a dilute NaOH solution and found no more indication of an end point than in a similar test at Battelle Memorial Institute wherein hydrogen penetrated $^1/_4$ in. of solid steel and became trapped under a pressure which ruptured the gasket at 500 psi (387). Secondary observations of blistering and swelling of heavy plate confirm the theory in indicating that that gas can rupture the strongest steel (387, 404).

[3] References (287, 340, 358, 384–389, 400–404, 417–427) constitute a complete bibliography of published work from Battelle Memorial Institute on hydrogen in iron and steel.

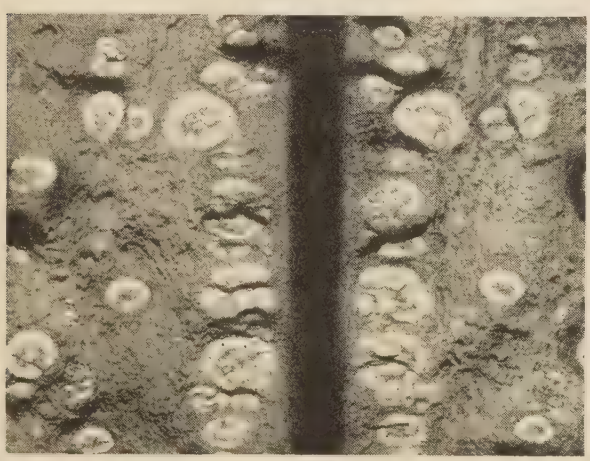


Fig. 2 Localized Transcrystalline Hydrogen Embrittlement Called "Fish Eyes," Observed on Fracture of a Tensile Specimen of Weld Metal; ×3

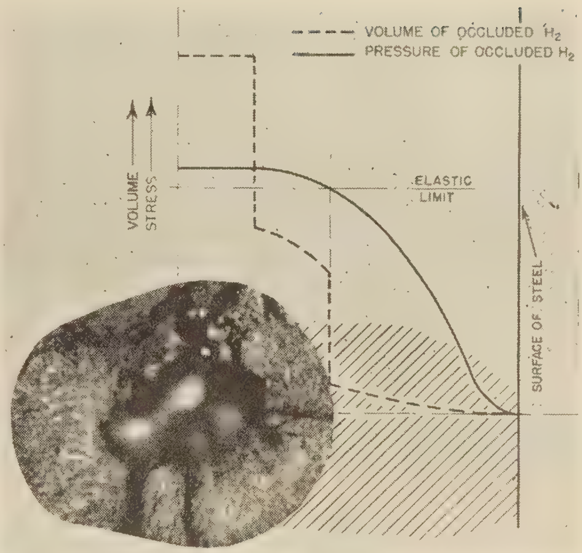


Fig. 3 Enlargement of "Fish Eye," Superimposed on Sketch Designating Probable Pressure-Volume Relationships of Occluded Hydrogen

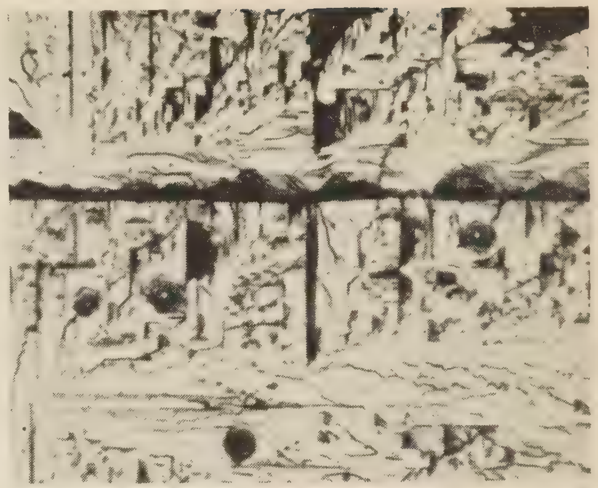
(Note shiny interior suggesting presence of reducing gas such as hydrogen.)

When steel absorbs sufficient hydrogen to congest those slip discontinuities, "*intra*granular embrittlement" results.

In the following illustrations, this type of embrittlement, which is caused by hydrogen alone, is clearly illustrated. "Fish eyes," a welder's nuisance, appear in Fig. 2. In this case, the hydrogen has been obtained through dissociation of the gas at high temperatures, rather than through the energy of an aqueous reaction, as in pickling or rusting (26, 388). The shiny appearance of the embrittled zones is characteristic of hydrogen embrittlement and expresses the fact that the fracture in these zones lies along flat, transgranular imperfection planes which have been distended and extended by the trapped gas. Reflection of light from those flat, extensive facets is the cause of the brightness. One of the "fish eyes" is enlarged and superimposed upon a sketch in Fig. 3, which designates the probable pressure-volume relationships of the occluded gas. Note that, throughout the brittle zone, the gas is under pressure which exceeds the elastic, or cohesive, strength of the steel.

The two views in Fig. 4 were made possible by the fact that the fracture is transgranular across flat, crystallographic planes. Single facets under high magnification were observed by a technique recently developed at Battelle Memorial Institute (423). Note the system of "imperfection planes" which intersect the main cleavage plane being examined and reveal the crystallography of the metal. Certainly these illustrations give striking proof that the temporary brittleness from hydrogen at ordinary temperatures, variously called "pickling britteness," "acid brittleness," "hydrogen embrittlement," and so forth, is definitely and in-

(a)



(b)

Fig. 4 Direct View at High Magnification of Single Cleavage Facets of Hydrogen-Embrittled Iron Broken in Impact
(a, Armco iron, purified in hydrogen and cooled at a sufficiently rapid rate to be embrittled by remnant gas; ×200. b, Purified Armco iron, embrittled cathodically; ×500.)

herently transcrystalline. Incidentally, specimen (b) in Fig. 4, though identical with "fish-eye" brittleness, is a steel embrittled cathodically, and therefore applies directly to the present argument.

Curiously enough, White and Schneidewind (245), and later Schroeder and Straub and their colleagues, used a research by Pfeil (182), in 1926, as illustrative of the type of fracture caused by hydrogen during pickling or cathodic electrolysis (385); yet Pfeil stands almost alone in reporting intercrystalline failure from pickling brittleness. This selection, it is interesting to note, actually counteracted their own unfortunate choice of pickling brittleness as a parallel case of boiler embrittlement, and probably sustained interest in hydrogen for a few years. Closer examination of Pfeil's work, however, shows that he actually obtained transcrystalline fracture, just as everyone else has, when ordinary specimens are used, and he discusses it at some length. However, when he went to the great trouble of growing crystals 1 in. or so in diam, the specimen broke through the grain boundaries, which is hardly surprising and is too far-fetched a case to be used excusably for comparison with boiler steel. Unhappily, Pfeil elected to display this unusual specimen upon the first page of his work, whereupon a great deal of misapprehension has resulted.

What investigators of boiler embrittlement have done, then, is to compare boiler embrittlement, which is predominantly intercrystalline, with ordinary pickling brittleness, which is distinctly transcrystalline; but they selected a misleading example purporting to show that pickling brittleness was intercrystalline, while the fact that, *under the conditions of boiler service, hydrogen attacks steel in a manner differing distinctly from pickling brittleness and which is truly intercrystalline is scarcely mentioned in the entire literature on boiler embrittlement.*

Action of Hydrogen on Steel at Elevated Temperatures—Intercrystalline Brittleness and "Boiler Embrittlement." Knowledge of intercrystalline attack of steels by hydrogen at elevated temperatures has stood in the literature at least since the invention of the Haber-Bosch process for ammonia synthesis (257, 267, 268) in 1908. In his Nobel Prize lecture in Stockholm (257), Bosch points out that one of the first problems which had to be faced was intercrystalline hydrogen embrittlement.

True, ammonia synthesis, such as the Haber-Bosch (257), or the Claude (289) process, utilizes pressures up to 1000 atm and temperatures from 400 to 600 C (752 to 1112 F); and hydrogenation processes, such as the Fischer-Tropsch, have conditions of temperature and pressure but little less severe at about 700 atm and 450 to 550 C (842 to 1022 F)(329); whereas only extraordinary steam units attain that temperature range, and never those pressures. Nevertheless, the difference is only of degree, for all processes supply H to the steel, although the rate varies. Boilers, accordingly, require months or years to fail in most cases, whereas the attack of boiler-type steel under the conditions of ammonia synthesis is so severe that its use is positively precluded. For example, Barber and Taylor (275) report steel actually pulverizing in 1000 hr from grain-boundary hydrogen attack in the hydrogenation of heavy liquid hydrocarbons. Investigators in that field have so little occasion to observe ordinary pickling brittleness that the following type of statement (275) abounds in their literature: "In all cases of hydrogen brittleness, the fracture takes place along the crystal boundaries." Compare this with the conclusion of Schroeder and Berk in their recent publication (398) on boiler embrittlement: "No intercrystalline cracking was found in any of the specimens," the "specimens" being steels loaded cathodically with hydrogen at room temperature, or for only short periods of time at elevated temperatures, in an attempt to reproduce boiler embrittlement under conditions which could not possibly result in anything but the irrelevant "pickling brittleness," if they became embrittled at all.

Among the major contributions in the literature on intercrystalline hydrogen attack are the researches of Vanick in America (156, 158, 161, 202, 203); Inglis and Andrews (262), and Sarjant and Middleham (314) in England; Naumann (329, 346) in Germany; Jacqué (334, 344, 345) in France; Giacomo (362) in Italy. Vanick began in 1922 with a steam-boiler failure which he associated with gases in the steel (156), but unfortunately did not blend that research with his subsequent conclusive studies on the behavior of steel under conditions of ammonia synthesis.

In 1923, Vanick (158) tested plain-carbon steel and wrought iron in NH_3. Below 300 C, reactions were slow, but at higher temperatures intergranular disintegration set in rapidly. De-

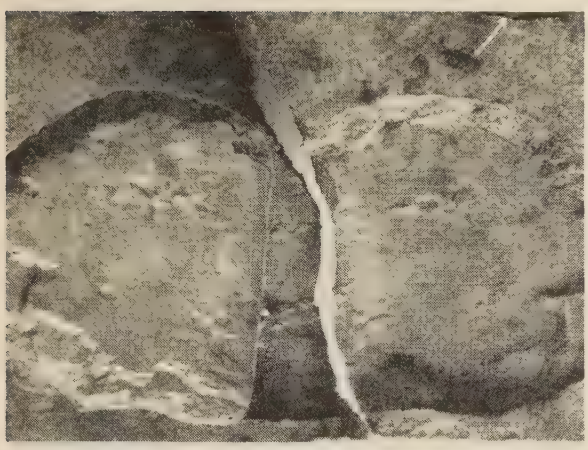


Fig. 5 Plain-Carbon-Steel Plate After Exposure to NH_3 at 500 C and 12,000 Psi; ×3

(Note large, deep-seated blister which is peeled back, and smaller blister showing as a fissure, indicated by white arrow. The fracture through this zone illustrates a second hydrogen activity, i.e., intergranular embrittlement.)

carburization leading to fissuring was the principal reaction he observed, in exact agreement with the latest English research on boiler embrittlement (396). The deterioration, i.e., "internal decay," he so aptly called it, progressed evenly from the cool to the hot end of the tube, which is a significant demonstration that the intergranular hydrogen attack of steam boilers and ammonia chambers would differ in degree, or rate, as would be expected from the difference in operating temperatures. The fact that boiler embrittlement seems restricted to temperatures below 250 C should not confuse the present argument, since boiler embrittlement depends rather precariously in its fulfillment upon such factors as seam concentration and boiler-water impurities. How such factors change at temperatures above 250 C is but little known.

Vanick also observed that inclusions were definitely reduced in these tests, some being completely removed, by the infiltering H. When situated suitably with regard to the surface of the plate, these inclusions led to blisters from the accumulation of gaseous reaction products (275).

An example appears in Fig. 5. The specimen is carbon steel after exposure to NH_3 at 500 C and 12,000 psi. Note the large deep-seated blister which is folded back, and the smaller one showing as a fissure. The fracture through that zone reveals the simultaneous hydrogen embrittlement. In the micrograph, Fig. 6, may be seen the only remaining islands of pearlite which could be found in the entire section, and those appear nearly completely decomposed. The straight crystallographic lines are "needles" of iron nitride. In other words, the equilibrium with the external gas has demanded that both nitrogen and hydrogen penetrate the steel. At any discontinuity within the steel, those gases will tend, of course, to re-establish their relationships at 12,000 psi. Secondary reactions with carbon and other nonmetallics to form products such as insoluble CH_4 are likewise driven toward their respective equilibria, which lie sufficiently far to the right to cause effective removal of a visible precipitate such as iron carbide.

Williams and Homerberg (162), in 1924, likewise stated that hydrogen reduced inclusions in their tests on the role of hydrogen in boiler embrittlement (168). They recognized the two distinct forms of hydrogen embrittlement being discussed in this paper and concluded that water as a reaction product distends the grain boundaries and leads to subsequent failure of boiler plate. Unfortunately, they had used electrolytic methods and did not defend themselves when subsequent confusion over hydrogen embrittlement obscured the importance of their results. Their tests lasted about 30 days, which was apparently just long enough for sufficient reaction products to accumulate to cause some grain-boundary weakness. Many years before, Law (45, 51) discovered that hydrogen caused "pickling blisters" on steel sheet (31), but insisted they were formed from H_2O produced by H reacting with the oxide inclusion usually found there. Neither Law nor Williams and Homerberg modified that statement; whereupon Law lost the value of his argument, because water is not found in blisters, nor is oxide always present (46); and Homerberg and Williams became discredited by similar observations. At the ordinary temperatures of pickling and electrolysis, H reacts but slowly with impurities in the steel (153), and blisters, as well as the brittleness, are simply caused by molecular hydrogen. Higher temperature favors those reactions.

By the subsequent work of Vanick (161, 202); Vanick, de Sveshnikoff, and Thompson (203); Barber and Taylor (275); Maxwell (294, 310); Giacomo (362); Puchner (379); Jacqué (334, 344, 345); Wright and Habart (355); Kosting (281); Bosch (257, 267, 268); Naumann (329, 346); Sarjant and Middleham (314); Newitt (376); and others (328, 336, 343, 363, 382, 391); the following general facts for hydrogen attack at elevated temperatures became clearly established:

1 H from the gas phase dissolves atomically within the lattice of the steel, leading to temporary brittleness.

2 H inside the steel then reacts with nonmetallic impurities, especially C, next with O, but depending entirely upon the stability of the oxide. S, N, and P may be attacked to varying extent, depending upon the stability of their respective phases. The insoluble gaseous products congest in grain boundaries, since they are insoluble in the grain.

3 When the pressure of these insoluble products exceeds the cohesive strength of the grains, the gas forces intercommunication and separation which manifests itself in loss of mechanical properties, cracking, swelling, blistering, or subsequent penetration of a corroding chemical.

4 The attack is favored by increasing temperature, pressure, and stress, is strongly influenced by the constitution of the steel and possibly by its heat-treatment.

Discussion of these facts has been available in textbooks for years (289, 372); yet, scarcely mention of it can be found in the entire literature of this country on boiler embrittlement.

Perhaps some actual cases of intercrystalline hydrogen at-

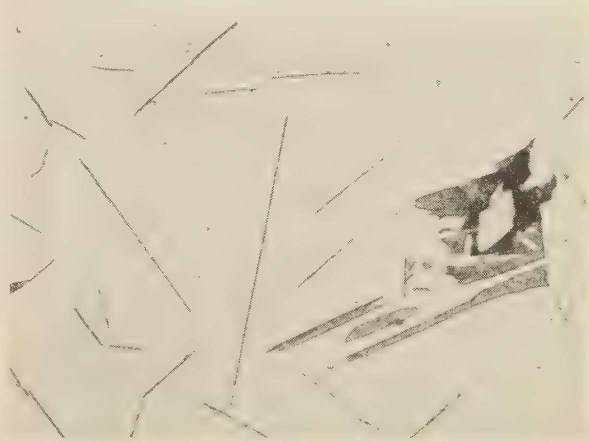


Fig. 6 Micrograph of Specimen in Fig. 5 Showing Nitride Needles and Advanced Decarburization; ×500

Fig. 7 Section of Carbon-Molybdenum Steel From Catalyst Basket, or Container, Showing Blisters From Hydrogen and a Fracture From Hydrogen Embrittlement; ×3

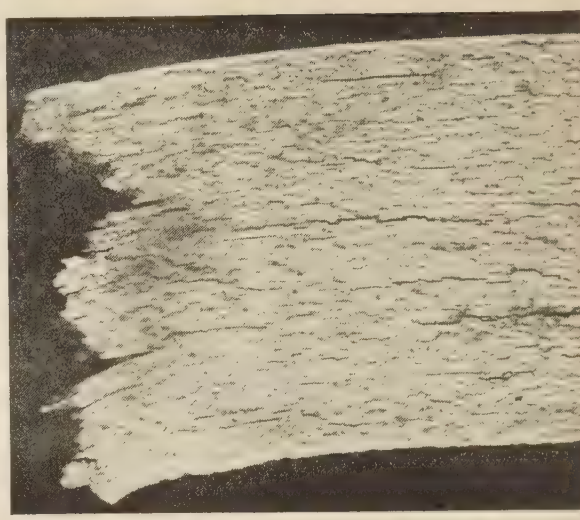


Fig. 8 Cross Section of Fracture in Fig. 7, Showing Deterioration From Hydrogen; ×3

Fig. 9 Micrograph of Specimen in Fig. 8, Showing Distinctly Intergranular Nature of Hydrogen Attack; ×500

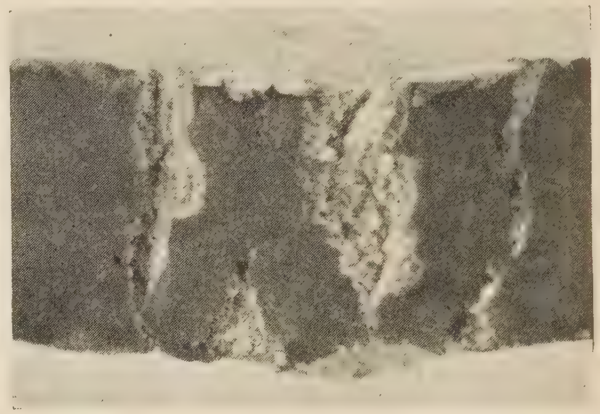


Fig. 10 Section of Carbon-Steel Tube After 1 Year in Hydrogen at 400 C and 400 Psi, Bent With Interior Wall in Tension to Show Deterioration From Hydrogen; ×3

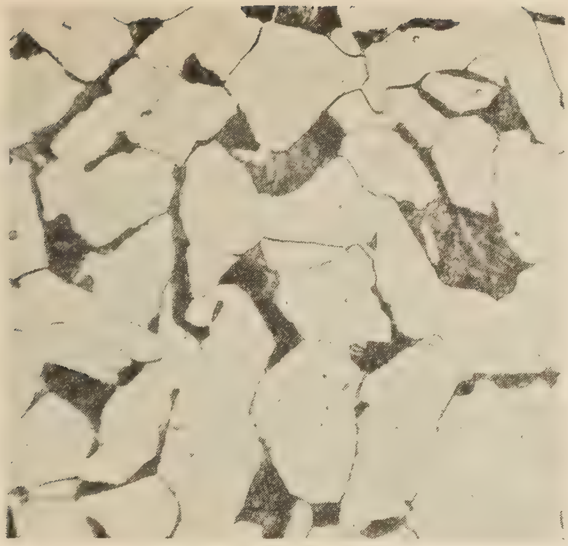
tack under conditions differing from steam boilers only in degree will be of value. In Fig. 7 is a macrograph of the surface of a carbon-molybdenum steel which served as a catalyst basket in an ammonia-synthesizing unit. Note the blisters in surface layers and the fracture occurring through embrittled metal, both defects being caused by hydrogen reacting with impurities in the steel. In Fig. 8, a section through that specimen shows the savage deterioration caused by the hydrogen. The micrograph, Fig. 9, shows intercrystalline attack with much greater definiteness than is obtained in cases of boiler embrittlement, if a distinction is to be drawn. Note the complete decarburization. The general absence of visible decarburization in boiler embrittlement in no way reduces the significance of these observations, as will appear later.

After 1 year of service in hydrogen at 400 C and 400 psi, a carbon-steel tube showed the deterioration evident in Fig. 10, where a section of the tube has been bent outward. In Fig. 11, micrograph (*a*) shows the condition of outer layers of that steel—carbide in the grain boundaries, but still largely unattacked; and micrograph (*b*) shows complete decarburization and marked intercrystalline fissuring in the inner layers.

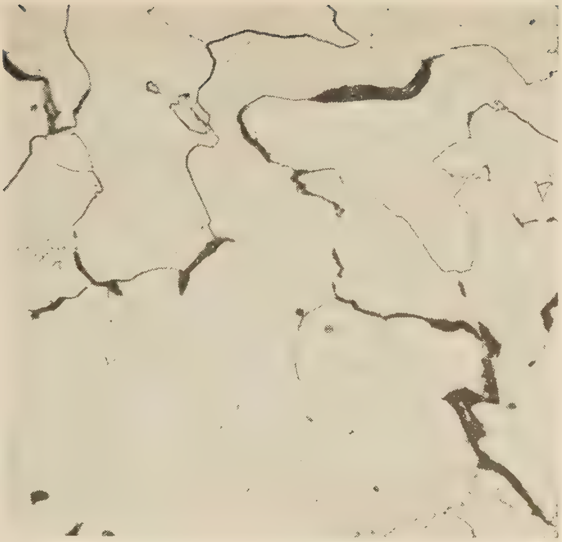
The next two illustrations demonstrate a point that is vital to an understanding of boiler embrittlement, for it largely bridges the gulf that seems to stand between the operating conditions of steam boilers and the hydrogen-attacked units under discussion. Fig. 12 shows a section of a carbon-steel tube from service in high-purity hydrogen at 400 psi similar to that in Fig. 10. But, in this case, the temperature was 100 deg lower, i.e., 300 C, whereas the time of service was doubled to 2 years. Correspondingly, deterioration is practically identical for each case. A macro-etch shows the depth of the decarburized zone. In Fig. 13, micrograph (*a*) again shows pearlite between the grains, attacked but not destroyed; while micrograph (*b*), taken in the decarburized zone, again shows the complete removal of carbide and the marked intergranular fissuring. Though it cannot be stated with certainty, the accumulation of pearlite toward the grain boundaries of the ferrite seems to be a preliminary action in the removal of carbon from steel by hydrogen.

Correspondingly, K. G. Jones[4] points out: "At pressures above 400 psi, with temperatures above 200 C, we know that hydrogen will cause decarburization over a long period of time. Carbon steel would not be satisfactory in ammonia over 200 C at

[4] Personal correspondence; see acknowledgment.



(a)



(b)

Fig. 11 Micrographs of Specimen in Fig. 10
(a, Outer surface, showing carbide in grain boundary, but still largely unattacked; b, inner surface, showing complete decarburization and marked intergranular fissuring; ×500.)

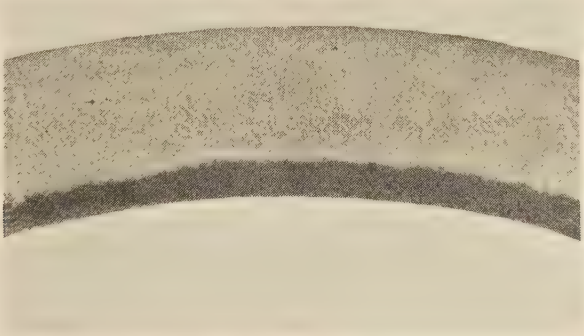
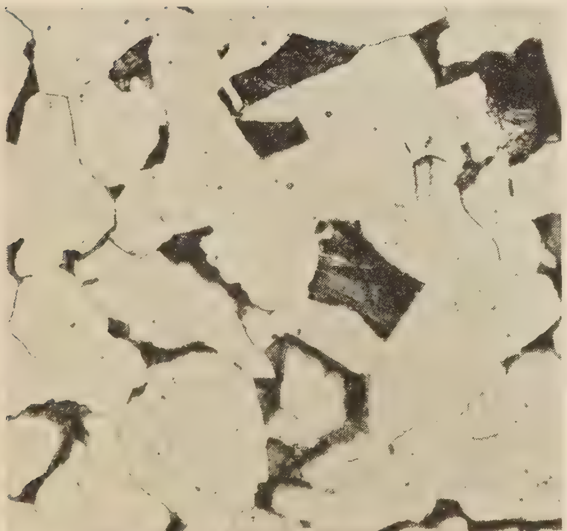
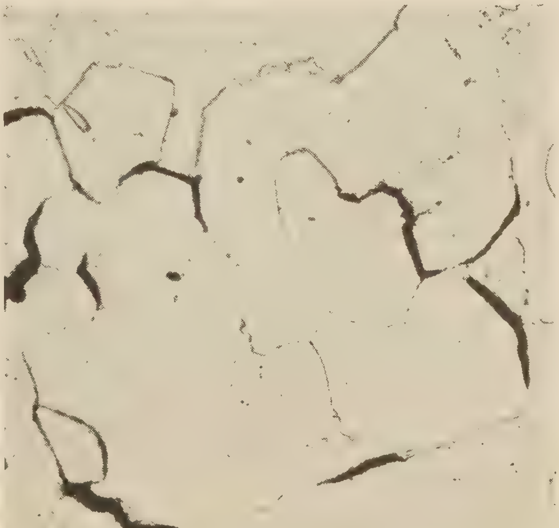


Fig. 12 Carbon-Steel-Tube Section Similar to That in Fig. 10, but After 2 Years at 300 C in High-Purity Hydrogen at 400 Psi
(Macro-etch delineates depth of decarburization; ×3.)

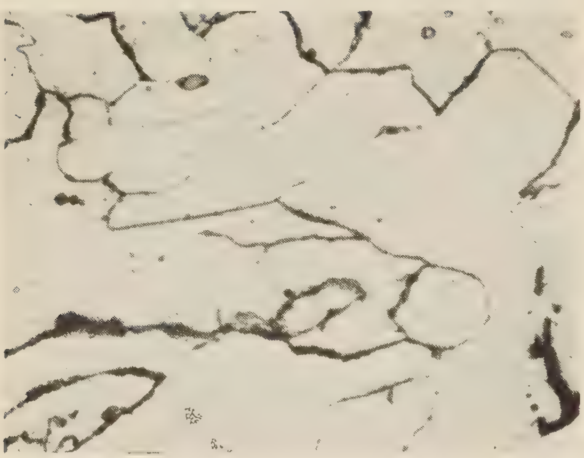


(a)

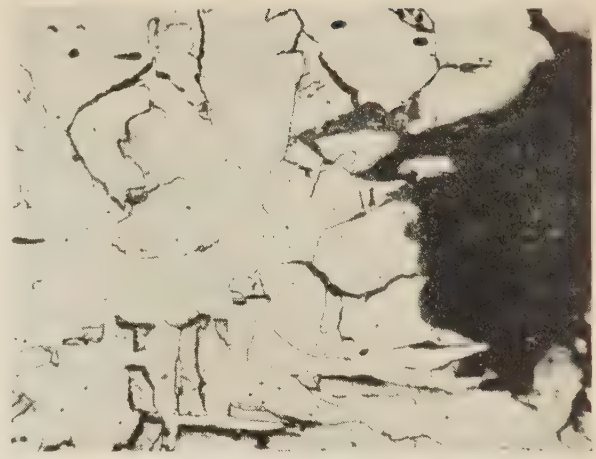


(b)

Fig. 13 Micrographs of Specimen in Fig. 12
(a, Toward external surface, showing pearlite between ferrite grains, but only slightly attacked; b, decarburized zone showing complete removal of carbon and distinct intergranular failure; ×500.)



(a)



(b)

Fig. 14 Micrographs of Carbon Steel After Exposure to Ammonia at 12,000 Psi and 500 C; ×500
(Note (a) complete decarburization, marked intercrystalline fissuring, and, in micrograph (b) the adjoining corrosion which is obviously secondary to the hydrogen attack.)

our plant pressure of 12,000 psi." Compare this with the recent monumental report from the National Physical Laboratory on boiler embrittlement (396), wherein it is reported that boiler plate, embrittled in caustic solutions at temperatures slightly above 400 C, analyzed up to 10 volumes of gas per volume of metal, that the gas was predominantly methane, and that, upon reheating the steel to 625 C, internal fissures actually became recarburized on their surfaces.

Hydrogen attack in high-pressure plants is influenced markedly by pressure (289, 372). For example, Tongue (289) reports that H_2 at 1 atm pressure caused no decarburization of a high-carbon steel in 1 week at 500 C, but at 200 atm the carbon content of the surface layers decreased from 1.4 to 0.1 per cent in 16 hr. Autoclaves in hydrogenation processes never reach even the lowest H_2 equilibrium pressure shown in Fig. 17, which will be described shortly. At those lower pressures, Newitt (376) reports that H attack can be observed at temperatures as low as 200 C, even in their comparatively short-time tests. Inglis and Andrews report slight attack at 150 C, over a period of 2 years (262).

In Fig. 14 are shown two micrographs from a specimen similar to that in Fig. 6. No nitrides are evident, but decarburization is complete. In micrograph (a), intergranular failure is outstanding; in (b), the same type of intergranular failure is shown near a locus of outright corrosion. Is it not obvious that the corrosion is subsequent to the actual deterioration of the steel by hydrogen? "Selective corrosion" in this case is quite evident, but it has been *enabled* by the embrittling factor; it did not *cause* the embrittlement.

Plainly, the principal difference between the two fields of investigation, boiler embrittlement and hydrogen attack of reaction chambers, is that with boilers a longer period of time is required to complete reactions which are rapid at higher temperatures. The fact that in one case ammonia produces the soluble H, in another case hydrocarbons, or pure H_2, and in another case steam or NaOH, is quite irrelevant, since the ability of steam to cause steel to absorb H requires no further demonstration. Other distinctions, such as the lack of visible decarbonation in most cases of boiler embrittlement, are superficial and will be resolved during later discussion.

Consequently, a glance at the cures that have been provided for H attack in reaction chambers should be well repaid. Briefly, the industry developed both mechanical and metallurgical devices. Double-walled units were used with the nitrogen passing through the interspace to prevent hydrogen from penetrating the outer mantle (267). Another double-walled unit used an outer mantle built to withstand stress and drilled with holes to permit the escape of hydrogen that penetrated the inner wall, which was "gastight" and grooved on the outside to aid the gas in reaching the holes in the outer plate (267). Vanick (161), and later Lewkonja and Baukloh (263), tested an Al coat on iron and found good resistance, except that such a coating is impracticable because of its imperfections. Pier (284) patented a shell of 1–6 per cent Cr steel with a lining of Zn or Zn alloy. A British patent concerns a lining of Ta or Ta alloy (314).

Liquid steel was treated with H to remove O, C, S, P, and other impurities which might later react with H during service. Such steel, it is reported, actually did withstand H attack (289). An English patent suggested similar treatment of the plant parts with H while red-hot before the final forging or rolling. In test, such steel remained unchanged after 300 hr, whereas ordinary steel embrittled in 24 hr (289).

Most of the research, however, has had to do with the composition of the steel. Plain-carbon steel, which takes months or years to fail under the milder conditions of steam-boiler service, fails in a few hours in the more severe service of ammonia synthesis (274, 307). Consequently, alloy steels must be used. Research soon showed that the principal attack was on carbon in the steel, and that the attack could be lessened by stabilizing that element with alloy elements forming more stable carbides than does iron, much as oxygen is "killed" in steels by adding Al. Chromium received the most and the earliest attention, and, thereafter, numerous other carbide formers, such as Ti, V, Zr, Cb, Ta, and W.

On these additions, Krupp steelworks in Germany holds the most interesting patents (282). Their subject matter is well covered by Naumann (329, 346), whose paper (346), in 1938, is easily the outstanding contribution on the subject of H attack of steel and its prevention. His findings are largely contained in the diagram, Fig. 15, in which the effect of various alloy additions upon the resistance to H attack is illustrated. Si, Ni, and Cu had no effect. Mn, Cr, W, and Mo increased the resistance in the order named; and Ti, V, Zr, and Cb showed no special effect until a limit was reached, beyond which a sudden and astonishing resistance to the highest temperatures studied showed itself. This phenomenal action clearly depends upon carbide

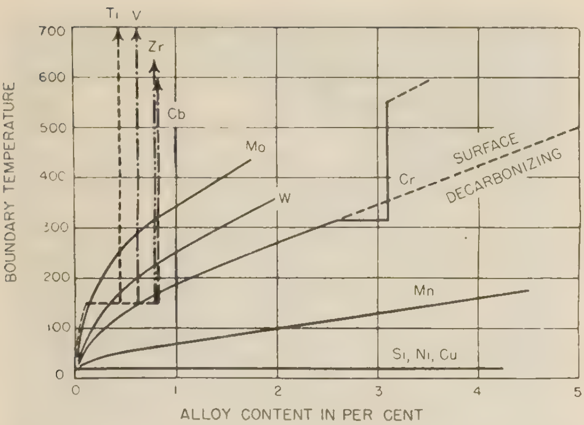


Fig. 15 Effect of Alloying Elements on Resistance of Steel to Hydrogen Attack at Elevated Temperatures; Naumann

formation, and hence upon carbon content; but for ordinary mild steels, such as boiler plate, less than 1 per cent of any one of these elements is all that is required. The maximum effect occurs when the vanadium content, for example, is greater than 5.6 times the C content, the formation of V_4C_3 then being assured. In any event, the patents state, C should be kept below 0.30 to 0.35 per cent.

Again, it is interesting to compare a statement from the recent publication on boiler embrittlement by Schroeder and Berk (398), who finally demonstrate for the first time in this country that killed steels show "distinctly better resistance" to embrittlement, but who conclude that that effect cannot be due to deoxidation because vanadium "which is also considered an efficient deoxidizer" doesn't show that improvement. They refer to tests of their own on steels having a vanadium content of only 0.25 to 0.30 per cent; and, referring to the curves in Fig. 15, one can readily understand why such steels would not reveal the stellar role of that element. Also, metallurgists now regard vanadium as a weaker deoxidizer than aluminum.

In another field, intercrystalline attack by hydrogen is also sufficiently prominent to warrant some attention, i.e., the so-called hydrogen embrittlement of copper, or "copper sickness." Vanick had tested copper in an ammonia-cracking unit in 1924, and found, after 4 months at 400 C, that it had expanded by 30 per cent of its diameter and fissured to its core (161). In the catalyst well, where the hydrogen is expectedly more nascent, a copper tube fissured in 6 days along a temperature gradient from 200 C to 600 C. If the copper had been deoxidized with boron, only slight fissuring resulted; and an oxygen analysis of that copper showed a content only 10 per cent of the ordinary. Bosch also points out that the failure of Fe and Cu in H_2 is identical (257).

Forty years ago, Heyn showed that copper heated in hydrogen at temperatures above 600 C increased in volume and developed fine fissures from H reducing cuprous oxide to form steam which became trapped within the metal (34). Also, tin oxide in bronze led to the same result. Archbutt, about the same time, removed the oxygen from copper by heating it in hydrogen, and found similar fissuring and bloating (36). Bengough and Hill found that CO, as well as H, diffused into the copper and formed insoluble compound gases (61); although Sieverts and Krumbhaar, studying the solubility of N_2, CO_2, H_2, and CO in copper found only the hydrogen soluble (62).

Embrittlement from H_2, CO, and steam was established by Ruder, who showed that temperature controlled the reactions to a large extent (88), and that hydrogen at elevated temperatures causes intercrystalline attack (91). Dry H_2 incepted embrittlement at 400 C, moist H_2 at 600, steam at 700, and CO above 800 C. Copper, deoxidized with boron, was not affected at all. Similarly, Johnson found that copper, deoxidized with ferrosilicon, was not affected by H even at 780 C (80); and Bengough and Hill had reported an analogous role for arsenic (61), on the argument that the arsenic stabilizes the oxygen in copper. Because carbon does not dissolve appreciably in copper, H attack principally concerns oxygen. Al, P, Mg, and numerous other oxide stabilizers came into use (80), all on the basis of the assumption that the hydrogen-oxygen reaction led to intergranular failure just as in boiler embrittlement. Redding noticed embrittlement of copper sheets simply from remanent machining oil left on the surface during annealing (147), and Ellis reported a 70 per cent loss in strength and a 90 per cent loss in ductility when vaseline was smeared on copper plates before annealing in a closed furnace (143). Pilling noticed embrittlement during brazing copper in a gas flame (145). In all cases, the source of hydrogen is evident.

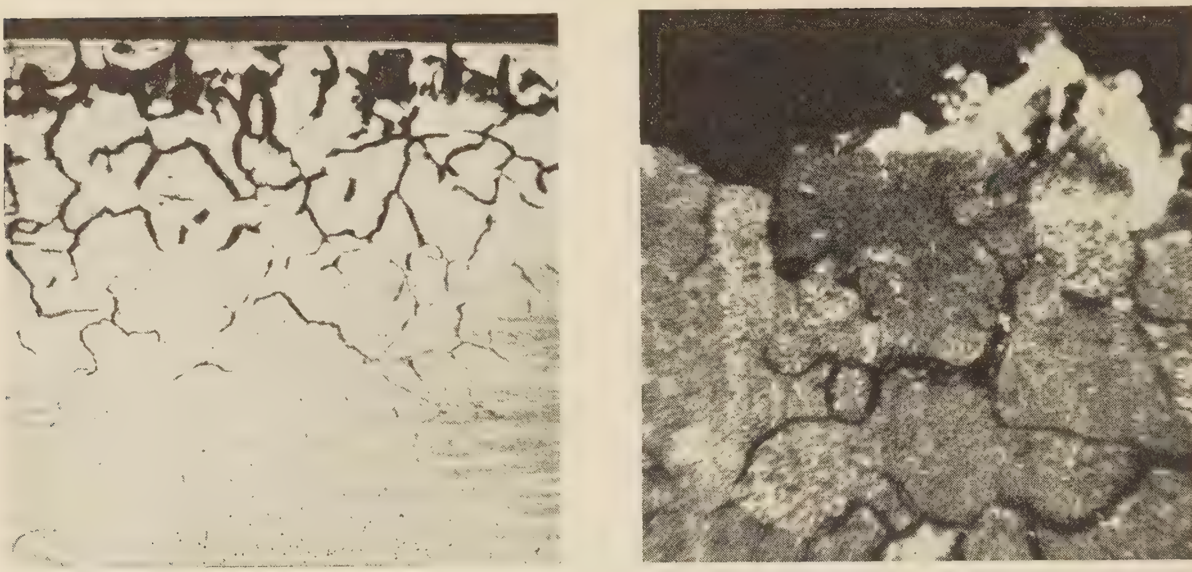
Determining the relative diffusion rates of various gases through copper, Pilling (104) lists H_2O at 65, CO at 17, and CO_2 at 0.6, when H is taken at 1000. If Cu is treated with H_2 while liquid, corresponding to the patent recently discussed for steel, a sound ingot results and embrittlement is prevented. However, if oxygen is absorbed, even during casting, the hydrogen-oxygen reaction is enabled (125). Sulphur in the grain boundaries also leads to intercrystalline failure (125); so oxygen is not alone, just as it is probably not alone in permitting boiler embrittlement, as evidenced by the incompleteness of "killed" steel. Moore and Beckinsale, in an outstanding paper (143), determined that as the oxygen content of copper became less, H attack began at higher temperatures. Bamford and Ballard (122) likewise discussed similar behaviorisms of hydrogen in brasses.

Finally, in purifying steel, especially in decarburizing, a hydrogen treatment at elevated temperatures is not only well known, but stands almost as the only means by which nonmetallics may be removed from solid steel. And, again, the grain boundaries are the path for occlusion and migration of the gaseous phase. The same Andrew whose work on caustic embrittlement (81) is quoted so widely in the literature, several years earlier had written on decarburizing steel with hydrogen (64, 69). Other investigators soon showed that H diffusing through the steel is attracted to impurity atoms, such as C, N, O, etc. (325, 347), and that insoluble reaction products develop whose removal constitutes the purification treatment (78, 111, 116, 117, 130, 152, 250, 253, 256, 263, 264, 279, 290–292, 303–305, 322–325, 338, 339, 347). The H removes not only C and O, for practicable desulphurization with H has been demonstrated (303, 304), at temperatures above that of boiler service, of course.

Other metals than steel and copper suffer similarly from these deleterious equilibria of hydrogen and nonmetallic impurities (237); and the scope of those reactions even extends into welding (337, 370).

During the hydrogen purification treatment, intergranular weakness is exhibited (263), as would be expected, because the seat of the reaction with the nonmetallics is in the grain boundary. In Fig. 16 are shown two typical examples of intergranular weakness in steels treated in hydrogen, but removed before the reaction products have escaped. Consequently, the grain boundaries are congested with the expansive gases, and the steel suffers from intergranular hydrogen embrittlement. If the treatment is allowed to go to completion, those insoluble gases escape, the grains weld together again, and the steel becomes ductile iron.

By now it should be firmly established that steel, when ex-



(a) (b)

Fig. 16 Micrographs of Free-Machining Steel Interrupted During a Purifying Treatment in Hydrogen at Elevated Temperatures

(a, Unetched, lightly polished; ×10; b, etched; ×30. Note distinctly intergranular fissuring.)

posed to hydrogen at sufficient temperature and for a sufficient period of time, suffers the following action:

1 Hydrogen enters the steel atomically.
2 Hydrogen reacts with the nonmetallic impurities.
3 Gaseous reaction products congest the grain boundaries to disruption, which is embrittlement.
4 Disruption permits escape of the gaseous products to the atmosphere.
5 Ductility returns.
6 The steel has become purified iron.

Because it is universally agreed that hydrogen does enter boiler plate, this progressive action positively must be an entity in "boiler embrittlement," whatever its degree of importance. Obviously, the action never even approaches completion; but only the first three steps are necessary for embrittlement.

Beginning with this firmly established background, let us now attempt to determine the ranking of hydrogen attack as the cause of "boiler embrittlement."

3 THE STATUS OF "BOILER EMBRITTLEMENT"

Steam-Iron Equilibria: The Case of the Closed Boiler

In view of the consideration just paid intercrystalline hydrogen attack, an interesting comparison is afforded with recent statements of Schroeder and Berk (398), regarding the possible role of hydrogen in causing intercrystalline embrittlement in boilers. Those investigators succinctly dismiss hydrogen from all consideration by subjecting the hydrogen "theory" to two peculiar tests. They state that the theory "may be tested, first, by determining whether or not hydrogen will produce intercrystalline cracks," whereupon they conduct room-temperature and short-time tests at 250 C, which of course produce only transcrystalline brittleness, or none at all. Not a word of recognition is given in their report to any of the references just reviewed, in which intercrystalline failure from hydrogen attack is the only consideration.

Upon drawing the astonishing conclusion that hydrogen does not cause intergranular weakening, Schroeder and Berk proceed with a second and, in their opinion, a decisive test. They state: "The action of hydrogen might also be tested in a second way, namely, by determining whether or not intercrystalline cracks can be produced in its absence;" whereupon they resort to an "anodic" treatment, whose complete irrelevancy can be established in numerous ways. The simplest is probably to point out that boiler embrittlement has none of the visual evidence necessary for a selective anodic attack. There is the common observation that "embrittled specimens are not corroded but are covered with a thin shiny . . . layer of black iron oxide" (181); as well as Schroeder's own observation that boiler embrittlement is not corrosion or corrosion pitting (240). What can anodic attack, which almost by definition is direct attack beginning at the surface and working inward, possibly have to do with such a phenomenon? To a metallurgist, who is aware that the intercrystalline failure of metals covers a vast field of individual conditions and causes, it is inadequate to test a specific case like boiler embrittlement with a nonspecific test.

The hydrogen mechanism, on the other hand, is an indirect attack, and therefore provides an immediate and a simple answer, since the gas does not deteriorate the metal until after it has passed through a finite surface layer of solid metal, so that its reaction products may become trapped. Once inside the iron in atomic solution, the gas then may react with impurities to produce retained gases with resultant deterioration of the plate. Or, the deterioration may later breach the surface, of course, as demonstrated in Fig. 14.

From their inadequate tests, Schroeder and Berk draw a conclusion which threatens to suspend an understanding of boiler embrittlement for many more years. They state: "At present there seems little reason to believe that hydrogen causes boiler cracking or that it plays an essential part in such failures" (398). Similarly, the University of Illinois Engineering Experiment Station states: ". . . we should divorce the idea of hydrogen embrittlement from the phenomenon of caustic embrittlement . . . that type of embrittlement due to hydrogen . . . is not the effect that we get in the so-called embrittlement caused by caustic"

(181). Rech regards Parr and Straub's work, in which they compare boiler embrittlement with pickling brittleness, as "eliminating the hydrogen theory" (220).

On the other hand, the 1941 Report on "Intercrystalline Cracking in Boiler Plate" by the National Physical Laboratory in England, names hydrogen as the fundamental cause of boiler embrittlement. The English investigators studied the defect by analyzing the gases in boiler plate after laboratory tests under somewhat exaggerated conditions. They do refer to Naumann's research on hydrogen attack, just discussed, and conclude that ". . . cracking produced by the action of sodium hydroxide was almost identical with that produced by gaseous hydrogen under high pressure."

Reversibility of the Steam-Iron Equilibrium. Besides the reasons already given for the present misunderstanding of the role of hydrogen in boiler embrittlement, there is another which is almost more surprising because it concerns elementary chemical principles, and the work on embrittlement in this country has been dominated by chemists. For example, Partridge and Schroeder state: "That oxides would be reduced by hydrogen passing through the steel, as postulated by Williams and Homerberg, seems unlikely . . . in view of the evidence . . . that the reaction proceeds rapidly in the opposite direction, iron removing oxygen from water and forming a coating of oxide with the simultaneous liberation of hydrogen."

Surely the error in such a statement is evident. In the chemical reaction

$$A + B = C + D$$

one side of the equation represents reactants, the other products, and the direction of the reaction depends only upon the concentration of the various components in respect to their concentrations when at equilibrium. Consequently, the most avid reaction going from left to right that one can conceive can be made to go in the opposite direction by reducing the concentration of the reactants sufficiently. Thus hydrogen and oxygen explode to form water, but the water in turn can be broken down into hydrogen and oxygen again if means are available for removing the tiny amount of dissociation product which must always be in equilibrium with the compound. Electrolysis, for example, is an accelerated means for removing the 10^{-7} mols of hydrogen ion present in 1 liter of water, which shifts that strong reaction into reverse and the water disappears.

Within a steam boiler the reaction[5]

$$3\ Fe + 4\ H_2O = Fe_3O_4 + 4(2H = H_2)$$

visibly always goes to the right because one of the products escapes as a gas. The equilibrium concentration of H_2 never becomes realized, and the entities on the right are continuously generated so long as free iron remains in contact with steam. Researches already discussed have amply demonstrated the continuous presence of hydrogen in the steam coming from steel boilers. The *rate* of generation, of course, depends upon the many factors which influence that contact, and must not be confused with the *equilibrium* now to be considered.

On the other hand, the reduction of Fe_3O_4 by hydrogen to produce water vapor and metallic iron is well known in metallurgy. That is the reverse of the action in the steam boiler and gives practical proof of the well-established fact that the equation just given is truly reversible. In fact, the chemical relationships for that equilibrium over a great temperature range have been so well developed both in this country (260) and abroad (253) that it is difficult to reconcile the complete lack of recognition paid it in literature on boiler embrittlement.

Briefly, when both Fe and Fe_3O_4 are present in excess, as they are in the steam boiler, their chemical activities remain constant for any given temperature and may be conveniently placed at unity. The equilibrium constant for the reaction

$$K = \frac{(Fe_3O_4)(H_2)^4}{(Fe)^3(H_2O)^4}$$

may then be greatly simplified

$$K' = H_2/H_2O$$

in which the two gases may be expressed in terms of their partial pressures

$$K' = P_{H_2}/P_{H_2O}$$

This expression reveals the highly significant fact that, at any fixed temperature, the ratio of hydrogen to steam is fixed whenever iron and its oxide are present together. In terms of the "phase rule"

$$F = C - P + 2$$

there are three components and four phases when water and steam are both present; then

$$F = 3 - 4 + 2 = 1$$

and the system is seen to be univariant, having but one degree of freedom. That is, when temperature, for example, is specified, all other entities become fixed. Thus, for a boiler operating at 200 C, the steam-gage pressure automically becomes established at 211 psi, as is well known. What is not so well known, were one to judge from the published researches on boiler embrittlement, is that the potential pressure of hydrogen becomes fixed for exactly the same reason.

Notice the following interesting features of this system:

1 The *ratio* of H_2 to steam for any given temperature is fixed at some constant value by the thermodynamics of the Fe–Fe_3O_4–H_2–H_2O system.

2 The actual steam pressure, P_{H_2O}, for any given temperature is fixed at some constant value according to a similar thermodynamic law for saturated steam, as every boiler operator knows.

3 Consequently, the hydrogen pressure, P_{H_2}, becomes fixed at some constant value easily determined from the equilibrium constant $K' = P_{H_2}/P_{H_2O}$

In other words, the operator fixes the temperature of his boiler, which in turn fixes the pressure of the steam, which in turn fixes the potential pressure of hydrogen—a pressure which the boiler will always strive to attain, but in vain, unless the boiler is sealed to permit that gaseous product to accumulate.

That immediately suggests the question: *What maximum pressure of hydrogen could theoretically develop in a closed boiler?*

As already mentioned, the data for $K' = P_{H_2}/P_{H_2O}$ are readily available in the literature (253, 260). Because Fe_3O_4 is the stable form of iron oxide below 560 C (394), the system is unique, and its data are therefore applicable to the steam-iron equilibria wherever they occur. The universal agreement on the form of the Fe-H_2O equation underlying boiler embrittlement is especially fortunate, and should remove the last doubt about the validity and applicability of the values which are plotted in Fig. 17.

The lower curve represents the variation of steam pressure with temperature, and is more or less familiar to all boiler operators.

[5] In the reaction, the double designation of the hydrogen ($2\ H = H_2$) is used to avoid a common, futile argument over the form the hydrogen actually takes, and to remind one that both the atomic and molecular forms do exist in keeping with their own equilibrium and that of the whole chemical system.

The upper curve represents the equilibrium pressures of hydrogen, P_{H_2}, corresponding to the saturated steam pressure for each temperature.

Those curves are undoubtedly surprising to many. Observe that the ordinate is logarithmic to allow encompassing the tremendous span of pressure involved. Even at 350 C, where the ratio nears a minimum, 10 lb of hydrogen pressure are required for every pound of steam; at 250 C, the ratio has risen to 100 to 1; and at 100 C, nearly 1,000,000 psi of H_2 must dilute the mere 15 psi of steam if equilibrium is to be reached.

The bewildering magnitude of such figures should not be allowed to confuse their significance. They simply show that the steam-iron equilibrium in a steam boiler at 100 C lies so far toward the right that for all practical purposes it is irreversible. Hydrogen could never accumulate under sufficient pressure to reduce the iron-oxide scale as rapidly as it was being formed, since the boiler obviously would explode before 1,000,000 psi could be reached.

The Grain Boundary as a Tiny, Theoretical Boiler. Within the body of the steel there is no steam, of course, but there may be hydrogen and nonmetallic impurities, such as the same iron oxide found within the boiler. The equation

$$3\ Fe + 4\ H_2O = Fe_3O_4 + 4(2H = H_2)$$

then must proceed *toward the left*, which is the reverse of the over-all boiler reaction.

In other words, hydrogen diffusing through the steel *must* tend to reduce each particle of iron oxide it encounters, just as iron ore is reduced by hydrogen in metallurgical practice. The reduction, of course, is governed by the same thermochemistry which governs the boiler reaction. Consequently, we can determine the potential, or limiting, pressures of H_2 and H_2O within the steel just as they were determined for the boiler. In fact, the locus of the oxide constitutes a tiny, theoretical boiler itself, wherein steam must be formed as a product of the hydrogen-oxide reaction. Because this is simply the reverse of the boiler reaction, the ultimate equilibrium is shown by the curves in Fig. 17, with an exception soon to be noted.

As a result, it now appears that cumulative gases within the steel could lead to congestion and disruption of the grain structure just as the cumulative gases could explode a sealed boiler if the steam-iron reaction were able to go to completion.

This, then, would seem to be the fundamental nature of the phenomenon of boiler embrittlement.

The same action is demonstrated in the hydrogen attack of catalyst chambers and hydrogenation units, as previously discussed. There, however, products besides H_2O are plainly involved, just as they undoubtedly are involved in boiler embrittlement (396). Parr and Straub (211), for example, found that high S and P led to more rapid failure, whereas, the general conclusion of work already reviewed is that O, C, and possibly S, N, and P are affected by the hydrogen (262). Straub, discussing White and Schneidewind's paper (245), mentions an instance of cracking being associated with nitrogen.

All nonmetallics with which hydrogen reacts, such as C, S, P, and N, must be expected to contribute to the total accumulation of gaseous products. Because P_{H_2} is fixed, these simultaneous equilibria are obviously similarly fixed. No calculations will be made here other than to point out that, for the equilibrium involving iron carbide and methane (253)

$$Fe_3C + 2\ H_2 = 3\ Fe + CH_4$$

$$K = (P_{H_2})^2/P_{CH_4}$$

$$\log K = -7990/T + 8.75$$

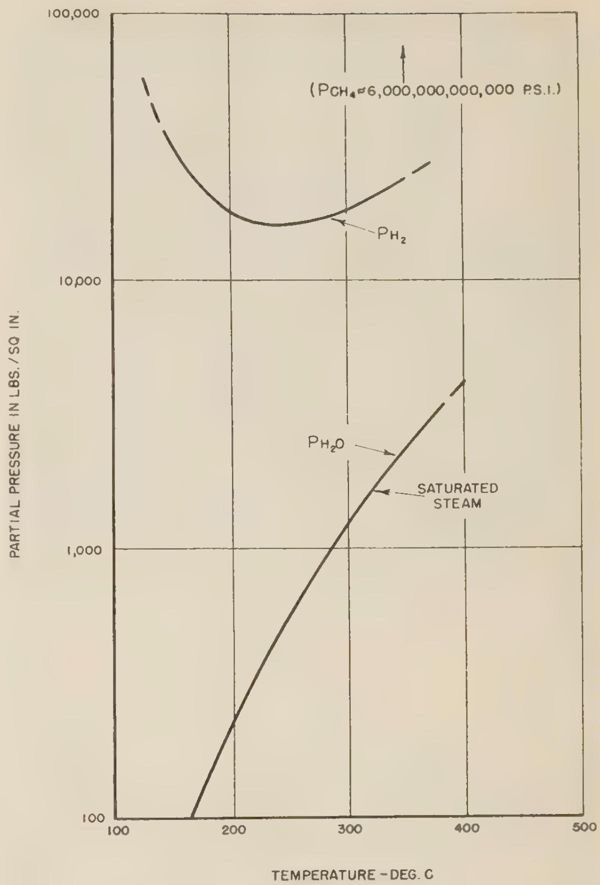


Fig. 17 Curves Showing Equilibrium Pressures of Steam and Hydrogen Over Iron and Its Oxide for Boilers Operating With Saturated Steam

whereupon, even at 350 C, the partial pressure of methane necessary for equilibrium with the steam and hydrogen prescribed for that temperature is approximately 5,760,000,000,000 psi! And at lower temperatures that figure rapidly increases! An arrow in Fig. 17, indicates that the methane pressures lie far off the graph.

Here again, the fantastic magnitude of these figures must not be allowed to contribute an impression of unreality. That monstrous value for P_{CH_4}, within its admittedly wide limits of accuracy, contributes the thoroughly sound information that iron, under the prescribed hydrogen pressure (see Fig. 17) of approximately 20,000 psi, could never be *carburized* by methane superimposed under any pressure available to us mortals. Or, from a more practical standpoint, the reaction to form methane under those conditions is irreversible. So long as carbon and hydrogen encounter one another within the steel, they will tend to add to the accumulation of insoluble gases. The concept is borne out by the fact that carbon may be removed from steel by hydrogen even at temperatures so high that methane is largely decomposed.

In this preliminary study, no conclusions will be drawn regarding the relative importance of the gaseous products in causing the grain disruption recognized as boiler embrittlement. Obviously, that is an important consideration, since it would indicate which nonmetallics in steel should be avoided. The beneficiation provided by deoxidizing steel with Al is undoubtedly

significant, as is the fact that such steel still ultimately fails in Schroeder's embrittlement detector. The literature on hydrogen attack concerns itself largely with carbon and methane, whereas decarburization is rarely noticed in boiler embrittlement. High-pressure boilers nowadays approach the service of hydrogenation units (365, 393); then decarburization is clearly indicated (365). Similarly, the English report marked decarburization from caustic embrittlement in tests on boiler plate at supernormal temperatures (396). Each of the gas reactions, of course, has its own equilibria and distribution of products and reactants over a temperature range. The same reaction need not be expected to predominate in these different services. The matter is best left temporarily simply as granular disruption by gases whose pressure can be expressed as

$$\Sigma P = P_{H_2} + P_{H_2O} + P_{CH_4} + P_{H_2S} + P_{NH_3} + \ldots\ldots\ldots\ldots$$

Equilibrium Versus Actual Boiler Conditions. Let it be understood that the foregoing calculations are based upon the ideal conditions of chemical equilibrium, but that they cannot be classed as being "only theoretical," since those equilibria have been determined and redetermined from actual experiment and have been made practicable. Of course, the values at high pressures are extrapolations and are subject to a consequent inaccuracy whose consideration is irrelevant at this time.

However, the question very naturally arises: *How closely are these calculated gas pressures likely to be approached in actual boiler service?*

Consider a unit, such as a boiler, separated into two compartments by a diaphragm of steel. In compartment *A*, the diaphragm has a clean fresh surface. In compartment *B*, the surface has such a heavy layer of Fe_3O_4 that for the present experiment it is inexhaustible. Compartment *B* is evacuated, leaving nothing but the Fe_3O_4, which is a product of the boiler reaction. Into compartment *A* saturated steam at 200 C is introduced, which is a reactant in the boiler reaction.

Immediately the steam reacts with the fresh iron surface of the diaphragm in the typical boiler reaction

$$3\ Fe + 4\ H_2O \longrightarrow Fe_3O_4 + 4(2H = H_2)$$

Now, regardless of the actual amount of the hydrogen which enters the steel, some quantity enters which is *that particular quantity injected by steam at 211 psi.*

In due time, some of the hydrogen will reach the Fe_3O_4 layer on the opposite surface of the diaphragm. There the well-established reaction for the reduction of iron oxide by hydrogen will occur

$$Fe_3O_4 + 4(2H = H_2) \longrightarrow 3\ Fe + 4\ H_2O$$

which is the reverse of the boiler reaction and has already been amply discussed, as have the pressures of the gases which must result.

Nevertheless, to recapitulate: H_2O will be produced in compartment *B* until enough steam is generated to reoxidize the iron and send H back into compartment *A* just as rapidly as the H_2O in that compartment is doing the same thing in the opposite direction. Obviously, the steam pressure in compartment *B* must then become 211 psi, since that is the steam pressure in compartment *A*.

At the same time that P_{H_2O} is increasing in compartment *B* enough H must precipitate from the iron without reacting with the oxide, of course, so that P_{H_2} is allowed to increase with P_{H_2O} n accordance with the curves in Fig. 17. Likewise, in compartment *A*, the portion of hydrogen which does not enter the steel will form molecular hydrogen until the steam is diluted with that same P_{H_2} that is ultimately attained in both compartments. The unit is then at equilibrium.

This fictitious experiment, of course, is simply a convenient means for representing the relationship between the steam boiler and the grain boundaries and inclusions within the steel which contain impurities reducible by hydrogen.

Reconsidering the "phase rule," an interesting thing will be noticed; within the grain boundary, or compartment *B*, there are only three phases, since water cannot exist at 200 C, until P_{H_2O} becomes 211 psi; then

$$F = 3 - 3 + 2 = 2$$

and the system is bivariant; whereas within the boiler, or compartment *A*, the system remains monovariant.

Thus the system is not completely described when the temperature alone is fixed. One more variable must also be stabilized. Ultimately, P_{H_2O} is that second variable which becomes fixed in the ideal unit when the steam therein reaches saturation and the fourth phase forms. But until the pressure of one of these gases is fixed, only the ratio of P_{H_2}/P_{H_2O} can be known, without any restriction as to the actual pressures of either gas. The total pressure may then be a matter of thousands of atmospheres, or only a few millimeters, so far as the chemical laws are concerned.

Without doubt, that brings us to the critical point in the phenomenon of boiler embrittlement, since the injection of hydrogen by the steam in the boiler cannot continue at the same rapid rate it had at that first instant when the surface of the iron was fresh and exposed. A covering of oxide immediately develops, which retards subsequent oxidation.

Without fear of confusion, we should now shift our attention from thermodynamics to kinetics; from ideal computations of equilibrium to actual rates of reaction, and how those rates may interfere with and upset the equilibrium. Plainly, the rate at which H is injected into the steel determines the concentration of the H within the steel; and that concentration in turn determines P_{H_2} within the grain boundary, or compartment *B*. This linking of rate with concentration is entirely valid, since the steel represents a reservoir for hydrogen through which the gas must obey the law of diffusion, in which rate and concentration are interdependent.

In other words, the gas ratio, and then the total pressure of $P_{H_2} + P_{H_2O}$ within the grain boundary, can become established by the fixing of P_{H_2}. Thus if H were supplied to the diaphragm in compartment *A* by some device, permitting a much greater rate of infusion than from steam at 211 psi, a superficial equilibrium could be imposed. The concentration of H within the metal would become abnormally high and the pressures of H_2 and H_2O in compartment *B* would adjust themselves to some supernormal values.

In the same way, if Fe_3O_4 in effect covers a part of the surface of the diaphragm in compartment *A* so that less H per unit time is injected into the steel, the gases in compartment *B* will adjust themselves to an equilibrium corresponding to steam in compartment *A* at something less than the actual pressure.

This, then, seems to be the explanation for the fact that boiler embrittlement depends upon agents like caustic and stress which will counteract the natural protection against hydrogen injection afforded by Fe_3O_4. Similarly, "accelerators," such as sodium silicate counteract the depression of hydrogen absorption. Without these counteracting agents, all agree, boiler embrittlement will not occur; conversely, it is possible that every steam unit would suffer embrittlement if it were not for the retarding and protecting effect of this cumulative reaction product, Fe_3O_4.

Is Failure Intercrystalline or Transcrystalline?

Since time immemorial for this subject, an argument has raged over the path followed by cracking in embrittled boiler

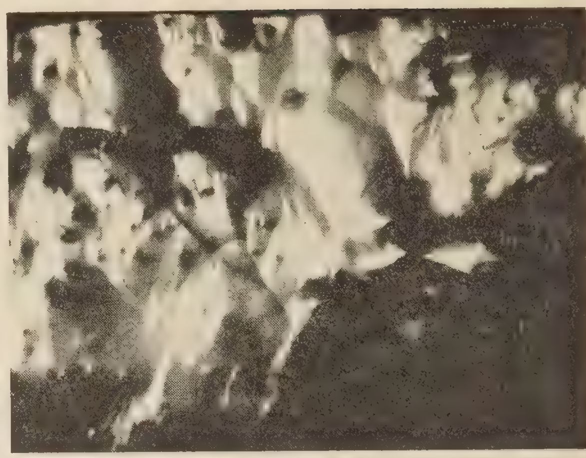



FIG. 18 DIRECT MICROGRAPHIC STUDY OF GRAIN BOUNDARIES ON A FRACTURED FACE OF INGOT IRON, EMBRITTLED INTERCRYSTALLINELY BY HYDROGEN

(Specimens fractured after interrupting a purification treatment in hydrogen; ×200.)

steel (97, 139). Many have given positive assertion to the fact that the cracks are exclusively intercrystalline (181, 220, 232); but others have noted that the cracks sometimes pass directly through the grain (137, 183, 406, 410), Perry (311) mentioning that either type may be shown "at the will of the operator." One case is even reported of cracking along twinning planes (133)! Numerous reasons have been advanced (128, 136), most of them superficial, none satisfying.

Now it should be clear why the cracks do exactly what they are observed to do. Let us refer again to Fig. 17. Only steam and hydrogen are shown there; but it is evident that at lower temperatures, the ratio of H_2/H_2O acquires such a predominance of H_2 that hydrogen must rely upon itself to embrittle steel. Its habitat is within the grain; consequently failure, when it occurs, is through the grain--transcrystalline. At higher temperatures, the ratio decreases until the sum of $P_{H_2} + P_{H_2O}$ is measurably greater than P_{H_2} alone; consequently failure, when it occurs, selects the habitat of the mixed gases, which is the grain boundary, since the gaseous product, H_2O, is insoluble in the grain.

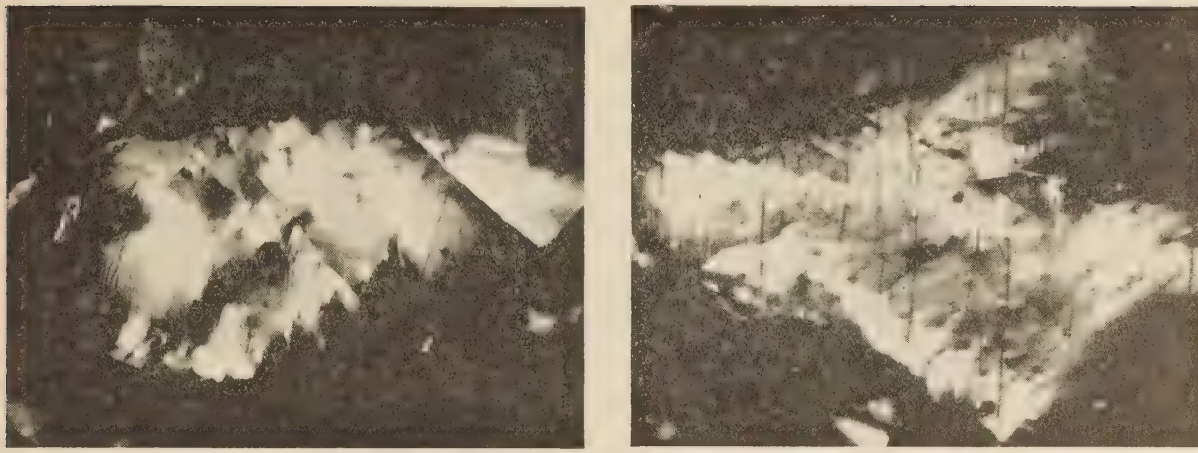
The gaseous products which are not shown, viz., CH_4, H_2S, NH_3, etc., have a role similar to that of steam with but one important distinction. That is, at lower temperatures some of their equilibria lie far toward the gaseous product, as is the case with methane, but without effect because the rate of reaction is so slow at ordinary temperatures. Consequently, grain-boundary failure, which depends upon an accumulation of those gaseous reaction products, tends to be suppressed at lower temperatures. Williams and Homerberg (162) found some reduction of oxide inclusions during cathodic electrolysis, as did Law years before (51). In fact, Pfeil's case of intergranular failure which has caused so much misunderstanding was unquestionably real.

In ammonia synthesis, then, intergranular attack is the rule; in acid-pickling and cathodic electrolysis, including short tests in caustic solutions at temperatures up to 250°C, or so, reaction products are seldom allowed to accumulate, and simple transcrystalline hydrogen brittleness is the rule. In boiler service, the tendency is definitely toward intergranular attack, because the extended period of service compensates for the low temperature; but the condition is sometimes intermediate. Some grains then break transgranularly, expressing the fact that the total gas pressure in the boundary may not always exceed that of pure hydrogen alone within the grain. The fact that sudden shock in the cracking process may also drive a fracture through a grain should not confuse this explanation, which applies to run-of-mill embrittlement.

Fortunately, this explanation need not rest upon theory alone, for the action can be directly followed through a micrographic technique recently developed at Battelle Memorial Institute (423). Samples of iron and steel showing various types of embrittlement by hydrogen have been fractured and studied at high magnification directly upon the untouched fractured face, as in Fig. 4. Micrographs showing both end stages and intermediate stages of embrittlement are presented in Figs. 18 to 20, inclusive. Fig. 18 is a view of a grain boundary as a surface instead of the customary line in sectioned specimens. Ingot iron, containing oxygen as its principal impurity, was heated in hydrogen at 1000 C for some days and was then withdrawn and fractured in impact. The specimen corresponds to those in Fig. 16, which were embrittled by accumulation of hydrogen reaction products in the grain boundary. Note the rough surface in the second view, suggesting that some impurities were still in place.

Fig. 19 represents the intermediate stage, wherein the pressure of grain-boundary gases is but little greater than the pressure of elemental hydrogen within the grain. The fracture, consequently, is intergranular, but transgranular planes of weakness have opened to accept the path of the fracture should a grain boundary hesitate. The first micrograph shows just a few crystallographic traces, and represents the specimen as it was removed from the hydrogen treatment. The second micrograph shows another specimen subsequently charged with hydrogen electrolytically, whereupon transcrystalline weakness was greatly developed. Nevertheless, that piece still chose to fracture predominantly through the grain boundaries in spite of the cathodic treatment. Evidently the pressure of residual gases in the grain boundary remained predominating—which may explain Pfeil's test, incidentally, since he may have occluded such products during his treatment for grain growth.

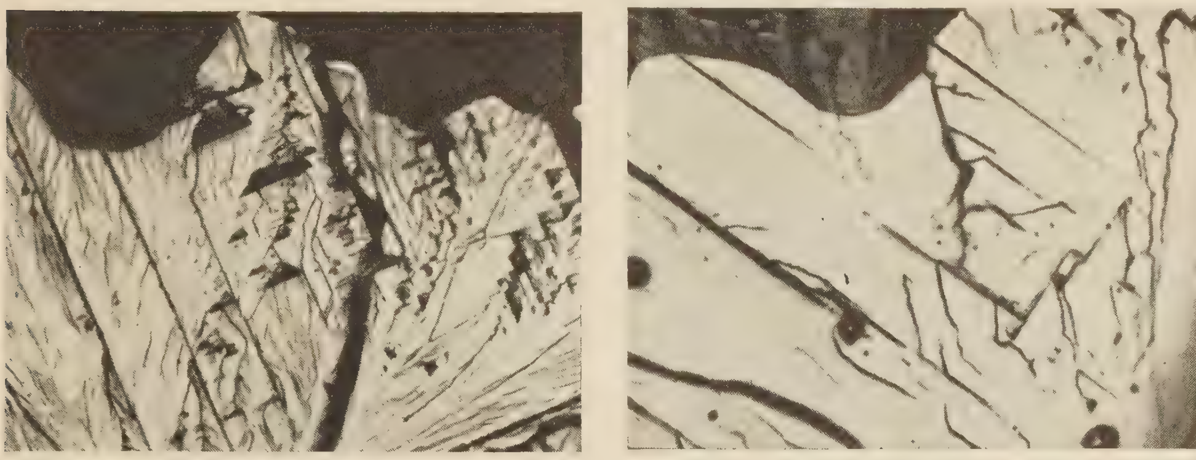
In any event, Fig. 19 verifies the discussion concerning the curves in Fig. 17, and explains why in certain cases the fracture of embrittled boiler plate occasionally chooses to pass through a grain. Fig. 20 is comparable to Fig. 4, and shows the other extreme. These specimens were purified in hydrogen, too, but the treatment was carried to completion. The grain boundaries were relieved of their burden during a subsequent treatment in vacuo, the metal healed, and cathodic electrolysis developed a strictly transgranular weakness.



(a) (b)

Fig. 19 Micrographic Study Similar to Fig. 18, but Showing Crystallographic Markings Developed by Elemental Hydrogen Within Grain

(a, As removed from hydrogen treatment; b, additional cathodic treatment to emphasize transcrystalline weakness; ×200.)



(a) (b)

Fig. 20 Same Technique Used as in Figs. 18 and 19, Except That Now the Fracture Is Entirely Transcrystalline, as in Fig. 4

(Micrograph a, ×350; micrograph b, ×1000.)

Fig. 18 therefore may be said to represent hydrogen attack under conditions of ammonia synthesis; Fig. 20 pickling-brittleness; and Fig. 19 boiler embrittlement.

The key to another puzzling observation in boiler embrittlement may be contained in this same explanation. The fact that the metal adjoining a crack is not brittle (38 has been held by some to disprove the hydrogen theory (233). On the contrary, that observation is in exact accord with the present argument, since escape of the gas through the surface of the fracture will permit ductility to return in the surrounding metal, just as it is known that boiler steel will recover its ductility if allowed to lie around awhile at temperatures well below those for annealing (155, 177). Although the compound gases are highly insoluble, they do ultimately lose at least their high compression, just as hydrogen does in a shorter period of time in the case of pickling brittleness. How could a theory of selective corrosion possibly fit these facts?

On the other hand, Straub's broad statement (233) that the adjacent metal retains its original properties may be going too far. In spite of the tendency to recover upon standing, more careful examination would very likely show deterioration of the proportional limit, if not the tensile strength and ductility, in agreement with well-established characteristics of intercrystalline H attack (262, 314).

Effect of Stress

Stress is doubtless important to cracking in boilers, for it plays a similar role in all other forms of hydrogen activity. Certainly the crack itself is caused by superimposed stress since hydrogen itself only embrittles—never cracks—sound steel (404, 418–420, 423). There is, however, an outstanding paradox in existing conceptions on boiler embrittlement which has received no attention and yet is critical to the entire subject.

Long ago there was established as a fundamental in metallurgy that, at temperatures where steel is elastic, the grain boundaries appear to be stronger than the grain itself (47, 113, 137). As Humfrey, an English metallurgist examining a boiler failure 30 years ago (75), observed: "It is a well-marked fact that the joints

between . . . the individual crystals . . . are considerably stronger than the gliding or cleavage planes within them." A moment's reflection recalls the fact that "cold-work," almost by definition, causes plastic deformation of the metal along slip planes within the crystal. In other words, stresses "beyond the yield point," which are so necessary to attaining boiler embrittlement, can scarcely be conceived to weaken the grain boundary, because those stresses are relieved by the imperfections called slip planes operating within the grain. As a matter of fact, even at the ultimate breaking load, after the stress necessary to start slip has been greatly exceeded and failure finally takes place, that failure is still through the grain, taking place along major planes of imperfection within the crystal commonly called cleavage planes. If the grain boundary is weakened at all, stress must disrupt the grain itself even more. How, then, can one adopt a theory of selective corrosion, a theory which depends upon the preferential weakening of grain boundaries by strain, and find it at all congruous with the facts? Furthermore, cold-working increases the solubility of steel in aqueous solutions, it is well known; but the increased rate of solution involves the grain itself in most cases, because cold-working makes the metal anodic, and because the effect of the cold-work is upon the grain itself. The selective-corrosion theory would seem to require the grain boundary to become electropositive from cold-working, when the exact opposite appears to be the case.

True, other forms of cracking, such as "season-cracking" in brass, and those discussed earlier, depend upon stress and yet have no conceivable relationship with hydrogen. But those cases remain unsolved. Even though they appear to represent selective grain-boundary attack, no one has yet provided an acceptable explanation for the phenomena. And if anyone wishes to take advantage of this exception, he must first explain why boiler embrittlement is notable for occurring underneath a smooth, black layer of iron oxide which always attests its inception and source; whereas every case of intercrystalline failure from plain chemical attack, to the best of the author's knowledge, shows the expected granulation on the surface. The present gas theory is tenable in that regard, since hydrogen dissolves into the iron without physical disturbance until it reaches the scene of its subsequent reactions.

In the present specific case of boiler embrittlement, hydrogen affords a simple and ready answer to this seeming paradox of stress. Any stress which exceeds the yield point likewise opens the imperfection or slip planes which house hydrogen in the elastic range (refer to Figs. 4, 20). Two factors then come into play: (*a*) More hydrogen is produced by the steam-iron reaction, because the iron is made more active chemically and because infiltration of the liquid into those imperfection planes permits a greater reactive area; (*b*) more hydrogen is absorbed because of the greater chemical activity and because the opened rifts afford sites for occlusion. In other words, in cold-worked steel, a greater movement or flow of reactive hydrogen gas occurs through the metallic membrane of surface grains, permitting a sufficient supply to reach inner grain boundaries where resultant gaseous products accumulate until they disrupt the grain-to-grain contacts, and the matrix is brittle.

Although stress is admittedly vitally important to embrittlement of steam boilers, it must not be classed as a truly fundamental factor, except in actual cracking. The fundamental factor in the destruction of ductility is the equilibrium between infusing hydrogen and the impurities within the steel. Stress aids in the attainment of, or the approach to, that equilibrium. Whether it simply breaks off the protective Fe_3O_4 layer, or whether its action is more subtle, cannot now be determined. It is plainly an accelerating factor which increases the rate of hydrogen infusion beyond some value that is probably critical for boiler conditions, below which the gases either diffuse from the steel faster than they can develop embrittling pressures, or embrittling pressures are not attained in the lifetime of ordinary boilers. The actual cracking does depend upon superimposed stress, and should not be confused with embrittlement, which is simply a condition of the metal wherein it is unable to respond to stress except by rupturing.

Such a conception of stress can be strengthened by numerous observations. Even in "pickling-brittleness," Langdon and Grossman long ago observed that cold-working accelerated embrittlement (126); and that cold-worked steel occludes inordinate quantities of hydrogen is common knowledge (384–389). At temperatures of the hydrogenation plant, where stresses are likely to fade rather quickly, previous cold-working greatly increases the intercrystalline failure caused by hydrogen (332), as it does in ammonia synthesis (372). Körber noted a similar effect with regard to decarburization in hydrogen (327); and, at Battelle Memorial Institute, specimens of carbon steel heated in hydrogen showed a tremendous increase in decarburization when the steel was simultaneously deformed in tension. In conditions of ammonia synthesis, stress is exceedingly important in accelerating deterioration; but with longer periods of time quite the same results are obtained without stress.

Effect of Caustic

Similarly, the effect of caustic, as important as it is, must be regarded simply as an accelerating, or catalyzing factor. A great many reasons have been advanced to explain the admitted fact that boiler embrittlement, as it is observed, depends upon NaOH; and there is no reason to doubt that that chemical concentrating in seams and crevices leads directly to the trouble. But again, the ultimate gas pressure obtainable in the grain boundaries, which is the basis for embrittlement, is the same whether Fe_3O_4 is produced by caustic or by pure distilled water. The gases which can form are dependent only upon impurities within the steel, and their partial pressures are controlled only by temperature, for the whole gaseous system is proscribed by involving saturated steam, as already explained.

Consequently, caustic must be placed in a category with stress as an agent which accelerates the infusion of hydrogen so that the gases will occlude more rapidly than they disperse, leading to embrittlement in an observable period of time. Whether the caustic itself reacts with the iron (229), whether it increases the electromotive force of the solution with respect to the iron (211), or whether its function is to dissolve the oxide reaction product, which would otherwise seal off the steel for further attack by H_2O (360, 365), is at present immaterial. Tajc even suggests that NaOH is "neither an active ingredient of the corrosion process, nor a factor in determining the stability of the resulting iron oxide" (333). He concludes that NaOH peptizes the oxide to form a colloid, which befits the third suggestion just listed.

Be that as it may, the important feature is that caustic is the agent in boilers which provides hydrogen infusion at a rate sufficiently rapid to surpass that critical value above which embrittling pressures of gas develop and below which the gases dissipate rapidly enough to make harmless the reactions which have been discussed.

There can be little doubt that gaseous hydrogen under 20,000 psi, the pressure corresponding to the potential pressure of the hydrogen produced on the boiler wall, would produce embrittlement in boiler steel at 250 C in the absence of the steam-iron reaction, since intercrystalline H attack from less pressures and at lower temperatures has already received discussion.

Of course, the attack would depend upon the impurities in the boiler steel, for that pressure of hydrogen alone is insufficient to

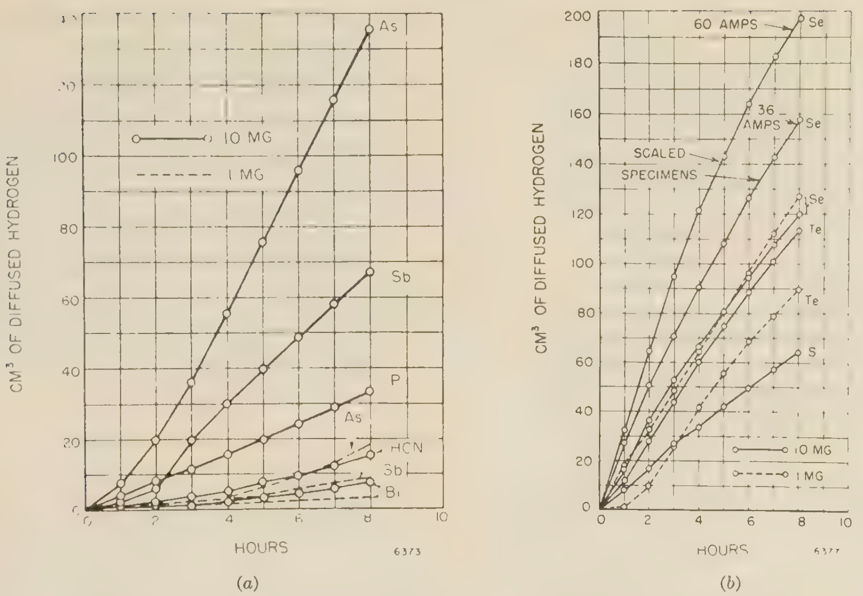


Fig. 21 Hydrogen Absorption From 10 Per Cent H_2SO_4 Electrolyte
(*a*, Traces of elements from Group V of the Periodic Table added to the electrolyte. *b*, Same for Group VI. Steel surface = 10 sq in., T = 25 C. Baukloh and Zimmermann.)

cause embrittlement (refer back to Fig. 3). That, then, leads to conclusions regarding the role of the steel.

Effect of Composition and Heat-Treatment of the Steel

The foregoing suggests an important point, that boiler embrittlement is a metallurgical study as well as a chemical one. In fact, the work of Schroeder and of Straub, and their respective collaborators, has advanced the chemical aspects of the problem so far beyond the metallurgical aspects that the rest of the problem is now almost purely metallurgical. Their tests, however, also demonstrate that "killed" steel, i.e., steel whose oxygen is stabilized or removed, is superior in its resistance (398), a fact discovered 15 years earlier by the German investigator who invented Izett steel. Aluminum oxide, of course, is much more resistant to chemical attack than is iron oxide; and the ultimate equilibrium pressure of H_2O over Al and Al_2O_3 is so negligible that that oxide effectively precludes grain-boundary congestion from steam. In other words, iron containing no impurities other than Al and oxygen probably would never show hydrogen embrittlement until the pressure of the hydrogen itself became great enough to cause the transcrystalline type of brittleness.

Likewise, the general role of "aging," first mentioned by Stromeyer many years ago, should now be clear, since "aging" is nothing other than the precipitation of certain impurities from the mother lattice. As separate, precipitated phases, most of those impurities are amenable to H attack. A truly "nonaging" steel, such as is approached by Izett, would simply have no important quantities of nonmetallics with which the hydrogen could react.

In tests reported by Solberg, et al (414, 415), steel alloyed with Cr showed increased resistance to attack by steam, which now assumes significance in the light of the analogous results already discussed on H attack. Boilermen often decry the use of alloy steels because of expense; but in industries where the effects of hydrogen are recognized, nothing but alloy steels receive consideration. Pilling and Straub (409), it is interesting to note, just last year obtained a patent for alloy steels containing Ni, Cu, Al, Be, and Ti in amounts up to 3 and 5 per cent to resist caustic-cracking. Even the ammonia-synthesis industry seldom resorts to such alloys. According to Krupp's experience, a low-carbon steel with 1 per cent or less of vanadium may do the job. That would certainly bear investigating.

The problem is to obtain steel as low in nonmetallics as possible, while retaining necessary mechanical properties, and rendering those impurities which are present as harmless as possible both by alloying and by heat-treatment. No improvement need result from removing oxygen, if C, S, or N, for example, are left in harmful quantities. Stromeyer many years ago, discussing Houghton's paper (84), blamed embrittlement principally on N and P, and stated that when $P + 5\ N > 0.08$ per cent, failure results.

By heat-treatment, some nonmetallics can be put in solution. Rapid cooling from the anneal then tends to prevent their precipitation. At least, their migration to grain boundaries can be prevented, which acts to prevent accumulation of gaseous products in the grain boundary. Obviously, heat-treatment can only be a palliative. In carbon steels, a "sorbitic" heat-treatment is advised, because a finely disseminated pearlite results. Grain-boundary carbides seem especially harmful.

Inhibitors and Accelerators

When hydrogen is accepted as the underlying cause of boiler embrittlement, an understanding of several hotly argued points on the action of so-called inhibitors and accelerators is possible. At the outset, Parr (94) concluded that oxidizing agents should inhibit embrittlement if hydrogen were its cause. In 1926, Parr and Straub stated that "NaOH is the only salt in the boiler which will embrittle steel" (181), and showed that Na_2SO_4 stops embrittlement when present in amounts approximately 3 times the total alkalinity as Na_2CO_3 (197). Then, in 1936, Straub and Bradbury (320) and Schroeder and Berk (315) simultaneously revealed a powerful role played by minute impurities in the caustic solutions that had been used in test (364). Principal among these was Na_2SiO_3, although the presence of Ca, Mg, and Al (320) and TiO_2, GeO_2, Sb_2O_3, and $SbCl_3$ (331) exerted an effect, too. Phosphates, chromates, tannates, acetates, besides

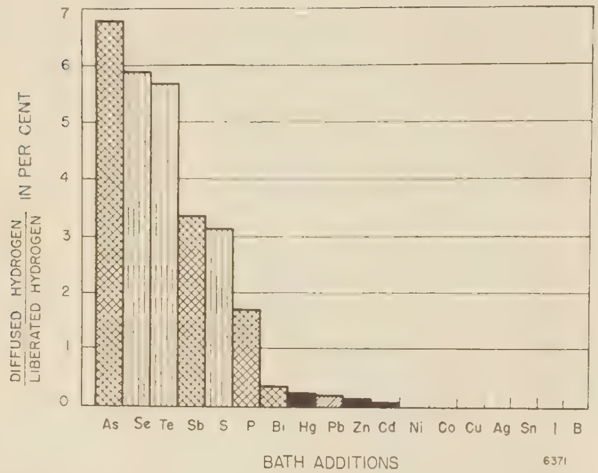


Fig. 22 Diagram of Ratio Absorbed/Liberated Hydrogen as Affected by Accelerating Elements Added to the 10 Per Cent H_2SO_4 Electrolyte in 10-mg Additions per Liter
(Current density 10 amp per sq ft, T = 25 C. Baukloh and Zimmermann.)

sulphates, were found to retard embrittlement by Parr and Straub (211), though a paper by Schroeder and Partridge (319) raises some question about Na_3PO_4. Schroeder, Berk, and O'Brien then state that dilute HNO_3 can be substituted for NaOH (331), although the year before Schroeder, Berk, and Partridge (317) found that sufficient quantities of oxidizing agents inhibit embrittlement; whereas in the recent A.S.M.E. Symposium on Boiler Embrittlement (405–408, 410, 412, 416) Bardwell and Laudemann (405) claim that their dilute addition of $NaNO_3$ has eliminated embrittlement in detector specimens on the whole Chesapeake and Ohio Railroad to date.

Finally, Na_2SO_4, the old stand-by written into the Boiler Code (185), itself failed to stop embrittlement (206, 311, 341, 397, 405, 406, 410), although Straub pointed out that the sulphate is ineffective unless a certain minimum of NaCl is also present (416). In the meantime, the National Physical Laboratory in England refuted Schroeder's silicate theory by obtaining embrittlement in pure NaOH, thereby proving that silicate can only be an accelerator. Rice even points out that the silicate may hinder embrittlement (397).

Desch undoubtedly comes closest to providing rhyme and reason when he observes that an inhibitor often acts as an accelerator under different conditions of temperature and pressure (361), similar to Straub and Bradbury's observation that sulphate protection disappears as the steam pressure increases, but that then another type of inhibitor fortunately becomes effective (352, 353).

In 1926, two Russian investigators, Alekseev and Polukarov (173, 174), discovered that a trace of As or Hg in an aqueous electrolyte led to rapid embrittlement of the steel cathode. Later, other elements were added to the list (250, 273), until Baukloh and Zimmermann (306), in 1936, provided the astonishing results shown in Figs. 21 and 22. The curves, in Fig. 21, show the tremendous increase in hydrogen absorption during cathodic electrolysis caused by seemingly negligible amounts of elements from groups V and VI of the Periodic Table. Arsenic present almost as a trace, for example, may increase the hydrogen absorption 100 times over that from a "pure" solution. As a matter of fact, in greater quantities these elements may act as inhibitors, which is significant in view of the analogous observations just recounted for boiler embrittlement (211, 361, 397). Körber and Ploum (250) even go so far as to suggest that iron containing none of these impurities will not dissolve in acid which contains none of them. In any event, these results offer significant comparison with the seemingly discordant findings of investigators on boiler embrittlement, for inhibitors and accelerators in hydrogen-caused boiler embrittlement would very reasonably be those inhibitors and accelerators here discussed. A diagrammatic illustration of the effect of these accelerators in aiding infusion at the expense of liberated portions of the gas stands in Fig. 22.

In the canning industry, the presence of sulphur and such elements has also been found to increase hydrogen evolution so greatly that the cans may swell and burst (247, 309).

In regard to inhibitors,[6] there is another interesting factor in boiler embrittlement which warrants study. Bardenheuer and Thanheiser (205), for instance, show very clearly that any inhibitor has its optimum range of temperature, and perhaps other conditions. When that range is exceeded, its inhibiting nature may disappear. One such instance is illustrated in Fig. 23, in which a plot of the ratio of absorbed to liberated hydrogen indicates that the absorbed fraction, which is the important fraction, of course, may increase without any evidence of a change in the visible evolution of gas.

The point to be made is that this type of action could operate at the steel-steam interface to promote or to inhibit the absorption of hydrogen, giving rise to the terms "inhibitors" and "accelerators." The suggestion simply amounts to an elaboration of Parr's original ideas. Christmann (258), for example, found that H from the steam-iron reaction reduces Na_2SO_4 in boiler water to form H_2S. Perhaps the inhibiting action of the sulphate is such a chemical one, removing a portion of the H before it can penetrate the iron. On the other hand, the argument over sulphate, i.e., whether its effect depends upon precipitation of a solid phase (181, 197, 297), may be explained by assuming that its precipitation on the steel would be a better barrier to hydrogen in spite of the fact that its chemical activity is then identical with that of the saturated solution (297). Na_2SO_4 is anhydrous above 33 C, whereupon a precipitated coating would protect the steel from steam attack. Na_2SiO_3, an accelerator, is strongly hydrated. Thus, Norton (377) observed a tremendous diffusion of hydrogen through an iron tube when it was coated with Na_2SiO_3, the accelerator, and heated to the boiler temperature of 200 C.

In further agreement, Norton found that sodium chromate, an inhibitor, completely stopped the diffusion through steel of hydrogen from tap water.

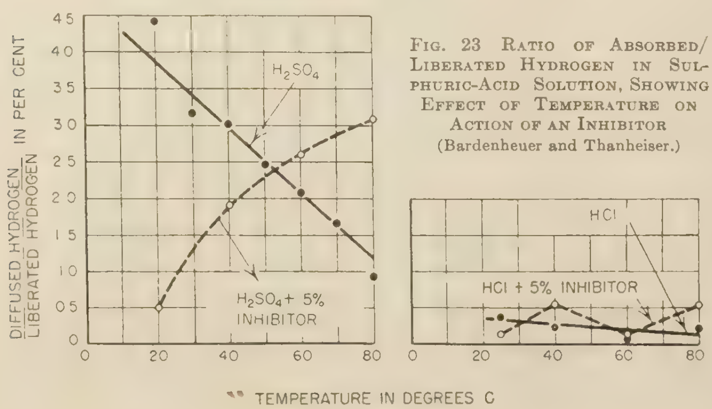
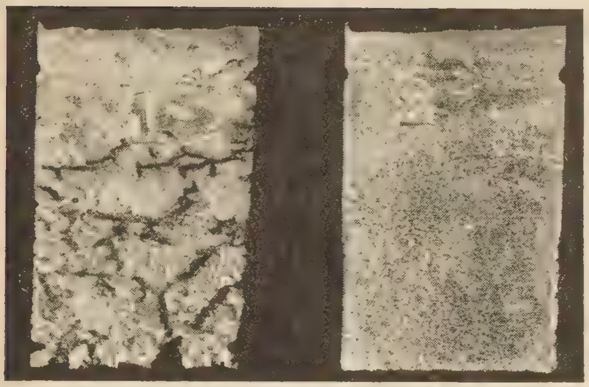


Fig. 23 Ratio of Absorbed/Liberated Hydrogen in Sulphuric-Acid Solution, Showing Effect of Temperature on Action of an Inhibitor
(Bardenheuer and Thanheiser.)

[6] For a more detailed discussion of inhibitors and accelerators, see reference 384.



(a) (b)

FIG. 24 CLIPPINGS FROM SHEETS SHOWN IN FIG. 1, AFTER DEEP-ETCHING IN HCl

(a, Annealed in hydrogen; b, annealed in air.)

(a) (b)

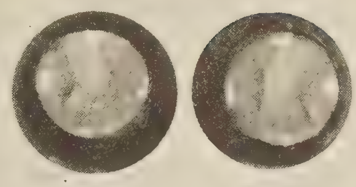
(c)

FIG. 25 TENSILE SPECIMEN OF WELD METAL CONTAINING HYDROGEN-EMBRITTLED ZONES, OR "FISH EYES," SHOWING PREFERENTIAL CHEMICAL ATTACK FROM CONCENTRATED HCl

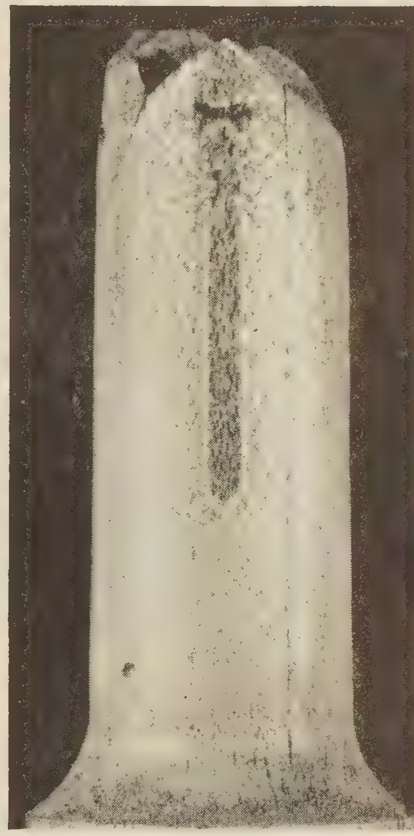
(Fractured faces, etched and unetched; a, original fracture, b, after deep-etching; c, side view.)

Also, Langdon and Grossman long ago observed that Na_2SO_4 inhibited cathodic embrittlement from hydrogen (126). And Hill points out that Na_2SiO_3 is widely used in flotation, possibly because it has an interfacial effect on gas absorption (308).

Such observations suggest returning to the original opinions of



(a)



(b)

FIG. 26 PHENOMENON IN FIG. 25, ARTIFICIALLY REPRODUCED BY EMBRITTLING ORDINARY STEEL TENSILE SPECIMEN ONLY ALONG FOUR THIN STRIPES AT QUADRANT POSITIONS

(a, Tensile fracture showing synthetic "fish eyes." b, Side view of half the specimen after deep-etching in HCl.)

Parr (94) on adding agents from the standpoint of frustrating hydrogen absorption. Simple tests at proper temperatures, using boiler steel as a diaphragm, with the various solutions on one side and a means of collecting transfusing hydrogen on the other side, might soon provide an empirical basis, at least, for selecting inhibitors suitable for any particular operating condition. Though Schroeder's embrittlement detector is a clever qualitative test, these diffusion experiments would provide quantitative information, and rapidly. Furthermore, that information would be so fundamental that running individual destructive experiments upon new boiler installations might become unnecessary, since the results could be predicted.

Why these substances affect hydrogen absorption as they do is

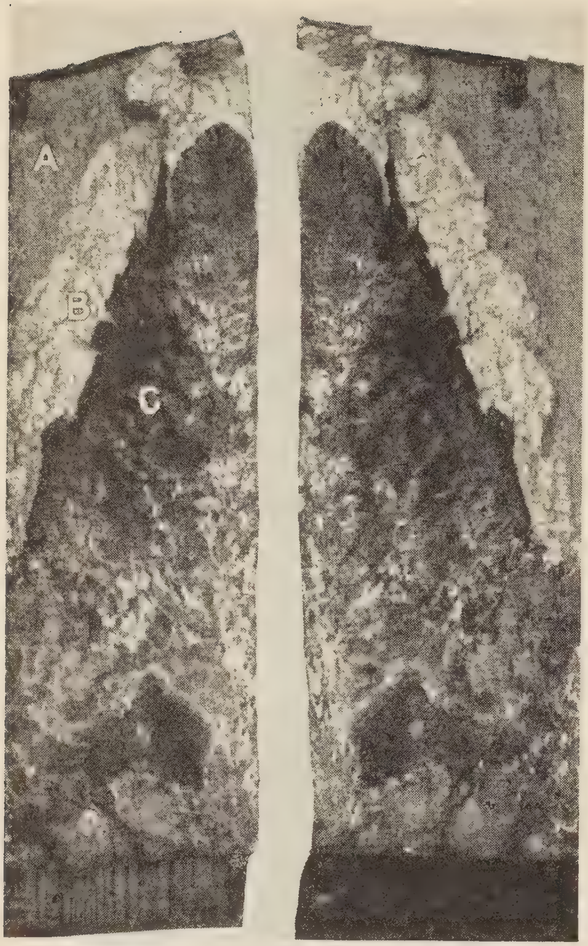


Fig. 27 Fractured Faces of a Partly Cracked Boiler Plate

(*A* shows a zone of seemingly normal, ductile metal farthest removed from the source of attack; *B* is a zone of hydrogen-embrittled metal preceding *C*, which is a zone of direct chemical attack, progressing into the original crack and into the intergranular voids opened by the hydrogen attack. ×2.)

as yet unknown. That they do is sufficient basis for prosecuting such an investigation.

Visual Proof of the Hydrogen Theory

In earlier sections of the paper, transcrystalline brittleness from elemental hydrogen was described as a function of pressure within crystallographic planes of the grain; and micrographs actually showed these planes opened by the gas during embrittlement. That they truly are opened, and with some permanent set, is shown in the illustrations which follow. In Fig. 24 are two clippings from the respective edges of the two sheets shown in Fig. 1. It will be recalled that one was embrittled by hydrogen and the other was not. Deep-etching in HCl developed severe fissuring in the embrittled specimen, as may be seen. In other words, the solution was able to penetrate more rapidly into metal that had been embrittled by hydrogen.

In Fig. 25, a similar instance is shown in regard to the "fish eyes" discussed earlier. The fractured faces of a tensile specimen of weld metal are shown as (*a*) and (*b*), the former being an original face; the latter the corresponding face after deep-etching in HCl; (*c*) is a side view of specimen (*b*). Note the savage attack of hydrogen-embrittled portions, attesting the distending and weakening forces of that trapped gas. In Fig. 26 the explanation is put to a more severe test. A tensile specimen of ordinary steel akin to boiler plate was coated with Glyptal except along four longitudinal stripes at quadrant positions. Cathodic electrolysis developed four synthetic "fish eyes," as may be seen in the fractures at (*a*). After 1 year of standing at room temperature, during which time all brittleness must have vanished, the specimen developed the marked preferential chemical attack shown in Fig. 26 (*b*). Thus hydrogen embrittlement truly distends and weakens the body of the grain. It must be remembered that these three figures pertain to transcrystalline brittleness, but also that they illustrate how hydrogen paves the way for subsequent chemical attack, a question which may have some significance in boiler embrittlement.

In intercrystalline H attack, such as boiler embrittlement, the compressed gas in the grain boundaries must have a similar effect, the only difference being that it opens the grain boundaries to subsequent chemical attack instead of the planes within the grain.

Accordingly, the whole story of boiler embrittlement seems to

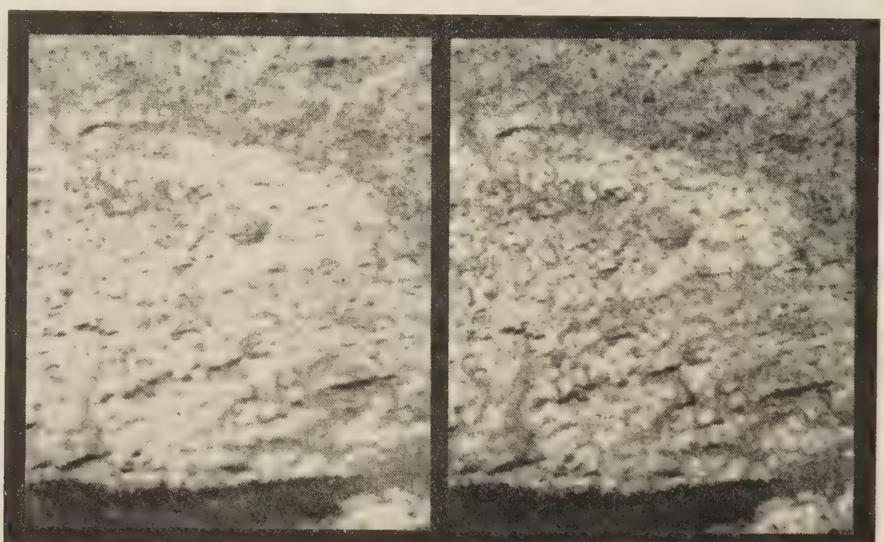


Fig. 28 Stereomacrograph of Zone *B* in Fig. 27, Showing Porosity From Hydrogen Attack and Progression of Boiler Embrittlement; ×10

lie within the single illustration, Fig. 27. Several samples of embrittled boiler plate were kindly furnished by Dr. Schroeder and Mr. Berk. Among other tests, these plates were broken open along visible cracks. A typical fracture through a crack beginning at a rivet hole in the outer plate of a double-plate boiler shell is shown in Fig. 27.

There are three obvious zones on the fractured faces; *A* is the gray fracture typical of normal, ductile metal and composes the zone furthest removed from the concentrated caustic solution in the leak around the rivet hole toward the inside of the boiler. Adjacent to that leaky seam and rivet hole is a zone of badly corroded metal, namely, zone *C*, obviously caused by penetration of the corrodant into the actual crack. In between these two is zone *B*.

Zone *B* is bright and clean, macroscopically and microscopically typical of metal reduced and purified in hydrogen. Zone *B* appears brittle, which is also typical of hydrogen attack. Zone *B* is intermediate between seemingly normal metal and cracked corroded metal; and that zone therefore represents an "advancing front" for this phenomenon of boiler embrittlement. The question: "Does the caustic enter the steel by creating fissures, i.e., selective corrosion, or are fissures created by another agent, such as hydrogen, which then permit the caustic to enter the steel?" seems to be answered clearly in Fig. 27.

Such an order of events is neither untoward nor unique. Long ago it was shown in the case of copper that granular deterioration from oxidation depended upon the grain boundaries first being opened by hydrogen embrittlement (142). Fig. 14 demonstrates the same thing for steel.

To convey the full significance of Fig. 27 as clearly as possible, a stereomacrograph[7] is included in Fig. 28. With the depth afforded by that type of illustration, one can more clearly see the marked porosity of the embrittled zone where nonmetallics have been removed from grain boundaries, seams, and inclusions, though photographic reproduction does this test an injustice. Into that porous mass, the corroding chemical can be seen advancing on one side; and on the other side one may observe that cavities also exist, but that the action is incomplete. Consequently, the metal there still appears ductile, although

FIG. 29 SECTION OF RAILROAD RAIL FRACTURED THROUGH A CRACK SHOWING DUCTILE METAL, HYDROGEN EMBRITTLEMENT, AND CHEMICAL ATTACK SIMILAR TO BOILER PLATE IN FIG. 27

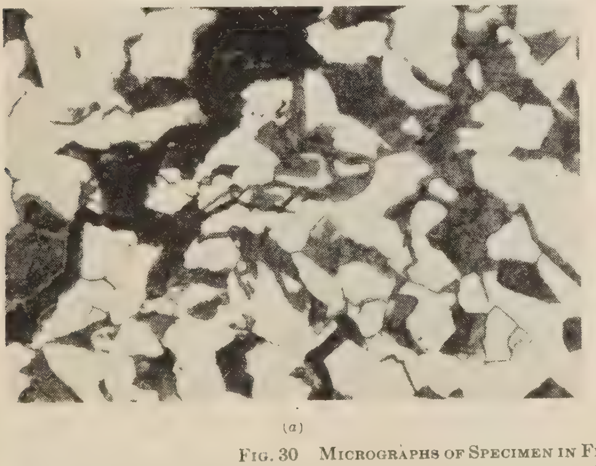
it must have contained grain-boundary gases under pressures which only lacked attaining that critical pressure at which the grains are forced apart and the metal is embrittled.

The whole appearance is in such exact accord with the present discussion that its importance can hardly be overestimated. The brightness of the brittle zone attests the strong reducing action of hydrogen; and the absence of flat cleavage facets in that zone shows that the attack is predominantly intercrsytalline, and that that fracture type was therefore imposed by the hydrogen function, since the fracture would normally be transcrystalline. The obvious and destructive porosity pervading the entire advance zone is sufficient testimonial *that boiler embrittlement and intercrystalline hydrogen attack are certainly identical.*

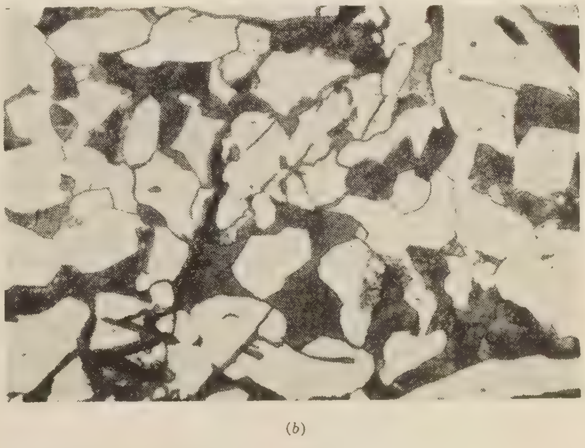
Actual cracking, it is well to repeat, is enabled—not caused—by the embrittlement. Use of the term "boiler embrittlement" for cracking is indiscriminate, since "embrittlement" only indicates a susceptibility to cracking, without further implication. Cracking first occurs when embrittled metal is subjected to stress. In boilers, that stress may be operational, or it may possibly arise from volume changes in the metal caused by the gas reactions, or both. The present study does not concern those stresses.

An observation with *trans*crystalline brittleness, directly analogous with that in Fig. 27, is shown in Fig. 29. A railroad rail, annealed in hydrogen for a short period at 1200 C, absorbed so much of the gas that it developed a large transverse crack from its own cooling stresses upon removing it from the furnace.

[7] This stereograph may be examined with a pocket viewer of the Keystone type from which the back has been removed, or with a matched pair of low-power magnifying glasses. Many people can obtain fusion by holding the prints squarely before the eyes at the shortest possible focal distance and "staring through" them at an imaginary distant object.



(a)



(b)

FIG. 30 MICROGRAPHS OF SPECIMEN IN FIG. 27, TAKEN IN CORRODED ZONE *C*; ×500
(Note the thin fingers of gray material in and around the ferrite, which probably represent iron oxide formed from water vapor penetrating fissures opened by hydrogen attack.)

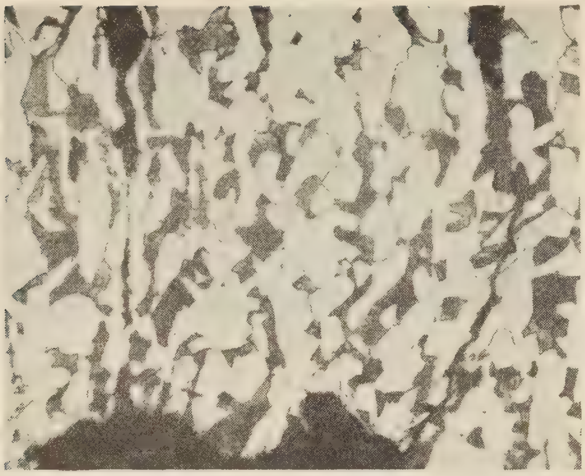


Fig. 31 Micrograph Taken Close to Boundary of Zones A and B in Specimen of Fig. 27, Showing Attack and Removal of Non-Metallics by the Gas Reaction; $\times 250$

After reheating in air to a temperature which confers ductility upon normal rail steel, the specimen was fractured through the crack, just as the boiler plate was fractured. The same three zones were visible, as may be seen in Fig. 29, though their disposition is considerably different, in accordance with the different conditions. Note the thin fibrous rim around the specimen, which is always present in hydrogen-embrittled specimens because the hydrogen cannot be held under embrittling pressure so near to the surface for any appreciable period of time. Then there is the shiny, brittle zone, and next the zone of chemical attack, this time from oxidation during the heating in air. The hydrogen embrittlement, then, enabled both the cracking from the superimposed thermal stress and the subsequent chemical attack.

Micrographic study of the boiler plate in Fig. 27 further confirms the statements which have been made. In the corroded zone C, penetration principally along grain boundaries is easy to find. Near the crack, all fissures lead plainly to the surface, and all are filled with corrosion product. Deeper in the metal, the cracks cannot be followed to the surface. Because they are likewise filled with corrosion product, the surmise is well made that they do connect with the surface but that the two-dimensional section does not reveal it. Also, deeper in the metal under zone C, the corrosion product is neater, taking on a smooth, gray appearance resembling iron oxide. It is likely that water vapor—steam—penetrates these finer fissures ahead of the liquid. These features may be seen in Fig. 30.

In the hydrogen-embrittled zone B, the removal of nonmetallics is everywhere evident, just as was plainly observed in a macrostudy of the untouched fracture. Empty holes betray the former presence of inclusions; but there is no trace of chemical corrosion. The action is quite the opposite, in that the metal is purified, and empty holes remain empty. The corrodant definitely has not reached the hydrogen-embrittled zone which is seen to lie in front of the zone of chemical attack.

No attack of the carbide is visible, in contrast to the specimens attacked by hydrogen at higher temperatures. However, C is only one of the nonmetallics which may serve as a basis for gas embrittlement, and reason has already been given for suspecting O in preference to C for boiler embrittlement. The English investigators found visible decarburization in boiler plate subjected to caustic embrittlement at temperatures above those of usual boiler service; and decarburization in high-pressure boilers has already been mentioned. At the temperatures of ordinary

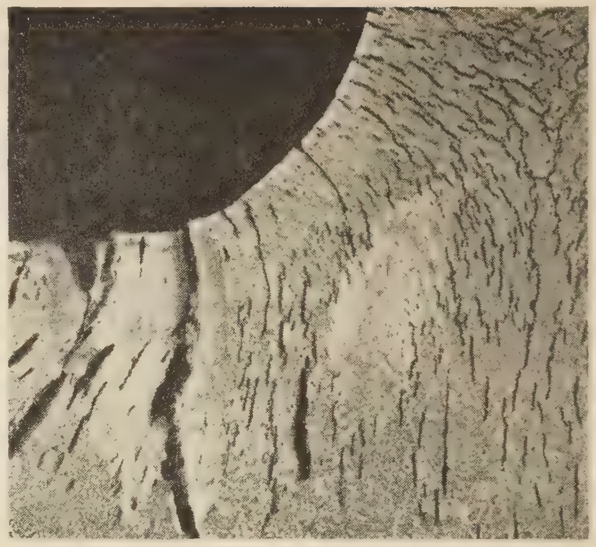


Fig. 32 Apparently Sound Boiler Plate After Deep-Etching; Compare With Figs. 24 to 26

boiler service, another equilibrium than that of C-Fe-H may well predominate. Again, the embrittlement is likely due to an accumulation of several gases.

On the other hand, one must recognize that outright visible removal of carbon need not be a prerequisite for embrittlement. The conversion of a gas, such as methane, from a solid state, such as iron containing carbide and dissolved hydrogen, demands a volume increase of roughly 5000 times for 1 atm pressure and at boiler temperatures. If the reaction takes place within the solid, in the absence of a fortuitous cavity, its formation immediately imposes upon the surrounding metal a compression of approximately 75,000 psi, which can account for embrittlement without further ado. The equations already discussed, with their monstrous values for potential methane pressures, clearly indicate that a mere 75,000 psi will do nothing to reverse or discourage the reaction.

Fig. 33 Fracture of Central Plate in a Three-Ply Boiler Showing a Brittle Zone Advancing From Both Sides With a Ductile Zone in Between

(The central position of this plate has allowed solutions to concentrate on both sides, with resultant hydrogen attack proceeding from two directions toward the center.)

Nor would the amount of carbide lost by this efficacious reaction necessarily be visible under the microscope. Visible decarburization represents removal of the carbon from the steel, which *follows* embrittlement and,

FIG. 34 MICROGRAPH OF BOILER STEEL EMBRITTLED IN SCHROEDER'S EMBRITTLEMENT TEST
(Note definite presence of both inter- and transcrystalline fissures; ×300.)

consequently is in no way necessary for embrittlement. The higher temperatures of hydrogenation favor removal of the trapped gas, and decarburization becomes visible.

In ductile zone *A*, inclusions are only partly attacked, in agreement with earlier conclusions. In Fig. 31, a single micrograph shows two inclusions lying as close to the boundary of zones *A* and *B* as can be determined. Although it is almost too much to expect, the inclusion on the brittle side is completely removed, and the one on the ductile side is only partly removed. Each is typical of the zone on its side, and no point will be made over the surprising proximity of these two cases. The important thing to note is that the inclusions, and material in the rolling seams, have been attacked and largely removed by the gas reaction, as was surmised from macroscopic observation in the stereograph. The absence of visible decarburization again points to oxygen and oxides as being particularly vulnerable to the conditions for boiler embrittlement. The nature of the crack running from each inclusion out to the fractured face is not clearly understood, but may indicate either brittleness at the time of fracture, or previous rupture caused by the trapped gases. The specimen, incidentally, was polished with care to retain all nonmetallics, though it was perfectly clear from macroscopic observation of the freshly fractured face that zone *B*, and zone *A* to some extent, contained only vacuous pores where nonmetallics formerly were.

In Figs. 24 to 26, the accelerated attack of hydrogen-embrittled metal was demonstrated, and the opinion was advanced that grain-boundary gases should do the same. A section of the boiler plate submitted by Schroeder and Berk was accordingly deep-etched, with the result shown in Fig. 32. The mass of fissures was visible only after the etch treatment, which certainly suggests a close analogy with the action shown in Figs. 24 to 26.

That plate, which happened to be the center strake of a three-ply boiler, was fractured through a visible crack, with the interesting results shown in Fig. 33. In its central position, concentration of the solution could occur on both sides. Consequently, there are two brittle *B* zones, one proceeding from each side of the plate. In between is the ductile *A* zone not yet overrun. The action is plainly stronger on one side than on the other, corrosion being rather limited to the side having the principal concentration of chemicals.

Micrographic study revealed similar conditions to those in the other specimen.

Schroeder and Berk also submitted some boiler plate which had been cracked in synthetic solutions containing 500 ppm of NaOH. Micrographic examination of those specimens revealed a typical example of combined intercrystalline and transcrystalline failure. An example is shown in Fig. 34. In their rigorous, short-time tests does it not now seem reasonable that the compound gases causing intergranular failure should lag and transgranular failure from elemental hydrogen should correspondingly be favored? No further comment should be necessary regarding the balance of gas reactions and its relationship to the failure shown in Fig. 34.

CONCLUSION

A Concise Definition of "Boiler Embrittlement." This preliminary study now affords a new conception of the phenomenon known as "boiler embrittlement," a conception which can be simply stated as follows:

"Boiler embrittlement is nothing other than an intermediate stage in a naturally occurring hydrogen purification treatment."

Thus, one begins with a boiler made of steel, which is the element iron containing certain nonmetallic impurities then:

1 The steam-iron reaction, which is enabled to progress more or less continuously, depending upon various factors, at loci where a caustic solution has concentrated, provides a more or less continuous infiltration of hydrogen through that steel.

2 The hydrogen reacts with the nonmetallics to form gaseous products within the steel.

3 These gases accumulate under pressure within the grain boundaries and at other seats of the nonmetallics until their pressure exceeds the cohesive strength of the grains and the granular structure is stressed to disruption.

That condition is recognized as "boiler embrittlement," and a stress applied to such metal may cause it to crack. The process, if allowed to continue, would then follow these well-established steps in the purification of steel by hydrogen at elevated temperatures:

4 Disruption of the grains extends communication of the gaseous accumulations until a portion escapes into the atmosphere, constituting the first actual removal of nonmetallics and incepting actual purification.

5 That degassing continues until the reactions are complete and insufficient gas remains to force itself through the structure out to the surface.

The material is then no longer steel, but is purified iron containing a small amount of residual gases. Embrittlement has vanished.

Direction and Aims of Future Research. As stated in the introduction as an opening premise, the present paper does not purport to provide a preventative of boiler embrittlement. Instead, it has attempted to interpret the work that has been done to date to show that a rational picture can be developed upon the new basis. With such an understanding, however, the battle is half won.

Simply stated, future research need have only two phases, one concerning a metallurgical study of steel to find an inexpensive

boiler plate for future boilers that will be relatively immune to hydrogen embrittlement from caustic attack, and the other concerning a chemico-metallurgical study of the role of inhibitors and accelerators in controlling hydrogen absorption so that the life of present boilers may be extended.

Both studies would be comparatively simple; and both promise extravagant results, because so little has as yet been done in direct relation to boilers, and yet from related fields a wealth of information is available.

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413 "Apparatus for Testing the Embrittlement Cracking Characteristics of Solutions," by W. C. Schroeder and A. A. Berk, U. S. Patent 2,283,954, May 26, 1942.
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416 "Results of Laboratory Embrittlement Testing of Boiler Waters," by F. G. Straub, Trans. A.S.M.E., vol. 64, 1942, pp. 393–396.
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425 "Scratch Blisters," by C. A. Zapffe and J. L. Yarne, ibid., vol. 25, 1942, pp. 180–190; abstract: "Relation of Oxidation to Enamel Defects," by C. A. Zapffe and J. L. Yarne, *Ceramic Industry*, vol. 36, May, 1941, p. 44.
426 "Relation of Surface Oxidation to Certain Defects in Enamel Coatings on Steel," by C. A. Zapffe and J. L. Yarne, ibid., vol. 25, 1942, pp. 191–194; abstract: "Relation of Oxidation to Enamel Defects," by C. A. Zapffe and J. L. Yarne, *Ceramic Industry*, vol. 36, May, 1941, p. 44.
427 "Chips, Fish Scales, and Shiners," by C. A. Zapffe and J. L. Yarne, ibid., vol. 25, 1942, pp. 194–203; also discussion, ibid., pp. 203–205; abstract: "Fish Scales, Shiners, Chips in Enamel," by C. A. Zapffe and J. L. Yarne, *Ceramic Industry*, vol. 36, May, 1942, pp. 43–44.

Discussion

W. C. Schroeder[8] and A. A. Berk.[9] The author is to be congratulated on the very large amount of data he has accumulated for his paper. Part of this information describes the two known effects of hydrogen on steel. Stated briefly these are as follows:

1 Hydrogen may enter or be trapped in steel at temperatures below the atmospheric boiling point of water to cause serious losses of impact strength. Fractures resulting from this brittleness are transcrystalline.

2 Hydrogen, under high pressure, at temperatures considerably above those at which riveted boilers operate will enter and decarburize steel, causing intergranular fissures to develop. At higher temperatures (1000 F), and a hydrogen pressure of 5000 psi, there is also a tendency for the inclusions in the steel to be reduced.

In contrast to the action of hydrogen at room temperatures and at very high temperatures, no intergranular fissure due to this gas has ever been reported to take place at temperatures from 250 to 460 F (15 to 480 psi), which includes the entire range in which embrittlement failures have occurred in boilers. This is particularly evident in the paper under discussion; for despite the very extensive bibliography, there are no illustrations or discussion of failures produced by hydrogen gas in the temperature range in which boiler embrittlement occurs. The author, nevertheless, postulates that such failures could occur by the following mechanism:

1 Boiler water concentrates by extreme evaporation in the boiler seams.

2 The strongly caustic solutions then react with boiler steel to form iron oxide and hydrogen.

3 The hydrogen enters the steel and reacts with carbides, oxides, sulphides, phosphides, and nitrides to form the corresponding hydrogen compounds.

4 The hydrides are not soluble in the grains, so that they congest in the grain boundaries, causing internal stress and fissures.

5 If the metal is externally stressed at the same time, it will crack when the load-carrying ability is sufficiently reduced by the fissures.

6 The crack allows the caustic solutions to penetrate and further corrosion and cracking follow.

Discussion of Proposed Mechanism. Let us examine and critically review each of the major items upon which this postulated mechanism is based. Step 1 relates to the accepted concept of formation of aggressive solutions from dilute boiler water and forms the basis of all modern thought concerning the incidence of intercrystalline cracking. The manner in which concentration occurs is not treated by the author.

Step 2 indicates that steel is attacked by the concentrated caustic solution. However, the author has misinterpreted Stromeyer's experiments to imply that caustic must act indirectly through hydrogen, because it seemed to operate on metal removed from the solution by $^1/_4$ in. of steel. Actually, in all of the embrittlement cracking that has been reported (including Stromeyer's tests) or that the discussers have examined, the failure can be shown to have been started at a metal surface in direct contact with a concentrated solution. This is true both of failures in boiler seams and the failure of specimens in embrittlement detectors. There is therefore no necessity to postulate an agency that can start cracks or fissures from within the metal.

If the postulated agency does tend to produce cracks or fissures from within the metal, it acts in a manner not in accord with known facts. Numerous sections of embrittled boiler seams have been examined microscopically and no such fissures have been found. Cracks that appeared to start and end in the heart of a two-dimensional plane were always traceable in the third dimension to the surface in contact with concentrated solution.

Step 3 demands the reaction of hydrogen in the steel with oxides, carbides, sulphides, phosphides, and nitrides to form corresponding hydrogen compounds. The experimental work cited to support this statement has been reviewed by the discussers. The original papers disclose that Whiteley (130),*

[8] Bureau of Mines, Washington, D. C. Mem. A.S.M.E.
[9] Bureau of Mines, Washington, D. C.

* Numbers in parentheses refer to the Bibliography at the end of the author's paper.

working at 1600 F, checked the results of Le Chatelier and found that hydrogen reduced iron oxide to only a slight depth in solid iron. Baukloh (304) found that while iron sulphide is affected by hydrogen above 750 F, manganese sulphide (the principal sulphur compound present in steel) is stable to 1800 F. Naumann (329) found that under circumstances producing decarburization (4000 psi hydrogen and 650–750 F), hydrogen is not able to remove phosphorus, sulphur, and oxygen. Zapffe and Moore (423), attempting to purify samples of steel with hydrogen at 1800 F found that carbon was readily removed but that oxygen, sulphur, nitrogen, and phosphorus were not removed even after 16-day treatments. This evidence does not support a belief that hydrogen will reduce the impurities in boiler-seam steel to form hydrides, for the temperatures at which these reactions proceed are far above those found in boilers.

The author, in an attempt to show through extrapolated equilibrium data that hydrogen would readily react with inclusions of magnetic oxide at boiler temperatures, made the error of confusing chemical kinetics with chemical equilibrium. What he has shown is that there is slight possibility for oxide to be reduced with the simultaneous formation of water vapor in the temperature range (250–480 F) in which boiler embrittlement occurs.

That the attainment of high pressures of hydrogen in steel at boiler temperatures is most improbable can be shown through calculations based on known diffusion constants. If the boiler steel were being corroded at the very rapid rate of $^1/_{32}$ in. per year, 130 cu in. of hydrogen (measured at standard conditions of temperature and pressure) would be generated per square inch of surface affected. If all of the gas were to enter and be trapped in the steel at 390 F, the rate at which hydrogen was being supplied would be 0.015 cu in. per sq in. of surface per hr. Possibly hydrogen can collect in a rift, cavity, or boundary at this rate. The pressure that would force the gas to leave the cavity by diffusion at the same rate can be calculated from data supplied by many investigators. The experiments of Smithells and Ransley[10] show that molecular hydrogen will diffuse from a cavity at the rate of 0.015 cu in. per sq in. per hr when the pressure is less than 400 psi. The results of experiments by Borelius and Lindblom,[11] by Ham (325), and by Ryder[12] provide figures of the same magnitude. A series of experiments by Naumann[13] also show that a pressure of 500 psi of molecular hydrogen will not only cause the gas to diffuse from a cavity quite rapidly but will cause 0.015 cu in. per sq in. per hr to diffuse through 0.2 in. of standard mild steel. Therefore the hydrogen pressure that may result in the steel of a boiler seam at 390 F cannot be very great.

The fact that boiler material embrittles without evidence of decarburization is shown by Fig. 35 of this discussion. This figure shows a crack in a boiler operated for a period of 9 years at 450 F. There is no evidence of decarburization or hydrogen purification. In contrast, the hydrogen attack that results in intercrystalline cracking is always preceded by decarburization.

Step 4 of the mechanism provides that the hydrides formed during the reduction of inclusions be rejected to the boundaries of the grain because they are insoluble in the grain. If they are not soluble and cannot diffuse in the grain it is difficult to understand why they do not remain in the cavity in which they are supposedly formed. Of course, if they happened to be formed between the grains they would presumably remain there.

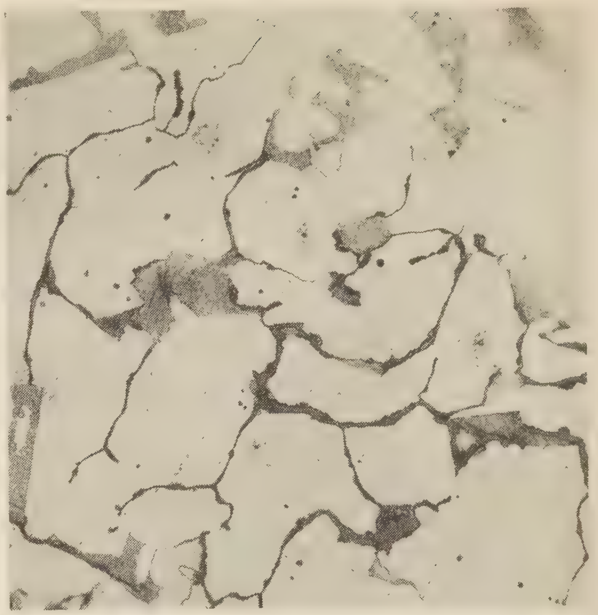

Fig. 35 No Decarburization Shown in Crack in Seam of Boiler Operated at 480 F for 9 Years
(Surface etched with nital; ×300.)

Step 5 of the theory of cracking as developed in the paper makes it necessary to have not only high temperatures but high pressures of hydrogen or other insoluble gaseous products as well. The combined conditions of high temperature and pressure are believed to cause fissures and loss of load-carrying ability in the steel. The evidence to support this belief is drawn from experiments at much higher temperatures than those at which embrittlement has been encountered in boilers. For example, Naumann (329), working with a hydrogen pressure of 13,000 psi, did not attain fissures or cracking during a test of 2000 hr at temperatures below 480 F. Vanick (203) did not find fissuring below 570 F. Desch (396) still obtained intercrystalline cracks when the temperature was lowered to 570 F, but no cracks were found even after a month at 435 F. Inglis and Andrews (262) found that over 2 years of hydrogen service at 3500 psi and 410 F did not greatly affect the strength of a coarse-grained mild-steel tubing, although all the pearlite areas were slightly decarburized. A fine-grained tube of the same composition was not even decarburized by a year of service at 480 F and 3500 psi of hydrogen.

In contrast with the lack of effect of hydrogen at temperatures below 450 F even over very long periods, repeated experiments have shown that embrittlement can be produced in a few hours at 230 F by concentrated caustic solutions boiling under atmospheric pressure. This difference between the action of hydrogen and the action of caustic is supported by the paper; for, in spite of the search of the literature, there is no evidence of hydrogen cracking at the temperatures in which boiler embrittlement occurs, although a great many industries use hydrogen, or obtain the gas as a by-product, in steel containers at temperatures corresponding to those in boiler operation.

Fig. 27 of the paper is advanced as visual proof of the hydrogen theory of cracking. The gray fracture of normal ductile metal comprises the zone farthest removed from the concentrated caustic solution. The corroded zone was obviously affected by penetration of the solution. In between is a bright, clean, brittle

10 "Diffusion of Gases Through Metals," by C. J. Smithells and C. E. Ransley, Proceedings of the Royal Society, London, England, series A, vol. 150, 1935, pp. 172–197.

11 "Diffusion of Hydrogen Through Metals," G. Borelius and S. Lindblom, *Annalen der*, vol. 82, 1927, pp. 201–226.

12 "Relations Between Gases and Steel," by H. M. Ryder, *Electrical Journal*, vol. 17, 1920, pp. 161–165.

13 "Influence of Alloy Elements in Steel Upon Resistance to Hydrogen Under High Pressure," by F. K. Naumann, *Technische Mitteilungen Krupp, Forschungsberichte*, vol. 1, 1938, pp. 223–234.

area. It is inferred that this latter zone is affected by hydrogen, but the corroding liquors had not penetrated to it. Photomicrographs of the three areas are described as proving that hydrogen preceded the corroding agent and did most of the damage. However, no decarburization and no fissuring are evident in any zone although Fig. 31 shows that cracking had penetrated even the gray area. There is no apparent reduction of inclusions in the corroded zone, which indicates either that not enough hydrogen was present to produce any appreciable results on the metal or that the temperature itself was too low to allow it to produce any effect.

If the three distinct areas were not formed by hydrogen, how were they formed? Fig. 36 of this discussion shows a section of boiler plate, which was cut to produce a line of concentrated stress on the surface and was finally broken by impact with a hammer. Again there are three zones in the metal. Obviously none of them is corroded by a caustic solution since neither solu-

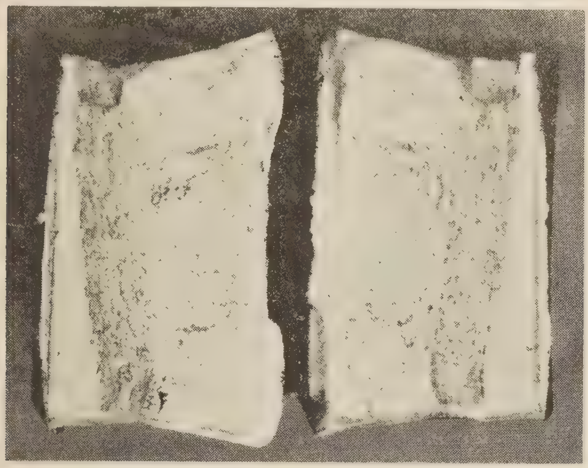

Fig. 36 Boiler Plate, Milled and Cracked by Successive Impacts

tion nor hydrogen was present. The example simply shows that three zones can be produced by impact and that any long discussion of what they mean is apt to lead to false conclusions. The characteristics of the surface are more likely to be determined by the way and the rate at which the steel is fractured than by any illusory internal conditions.

Step 6 completes the author's theory, providing for propagation of the crack through the action of sodium-hydroxide solutions on the newly exposed surfaces. Again the supposition is that a series of fissures are formed through the action of hydrogen on inclusions in the steel and that the fissures are joined to the surface by a crack when the stress exceeds the load-carrying ability of the remaining metal. No such fissure has ever been observed in embrittled boiler seams. It is also interesting to observe that propagation is supposed to take place along grain boundaries, although a piece of steel broken in tension at these temperatures will always break along slip lines in the grains. At the higher temperatures of the ammonia-synthesis and hydrogenation processes from which the author draws his parallel, it is well known that the tensile strength at the boundaries tends to approach that within the grains, permitting the familiar pattern by which the fissures are joined.

This paper points out that certain steels have been developed which will resist the reducing and fissuring action of hydrogen gas. If there is any relationship between hydrogen attack and boiler embrittlement, it would then be expected that these same steels would also be resistant to boiler embrittlement. In the past few months, the discussers have had the opportunity to test steels containing manganese, chromium, and molybdenum of alloy contents which, according to Naumann (346) would greatly increase their resistance to attack by hydrogen. All of these steels were rapidly cracked in the embrittlement detector, and some

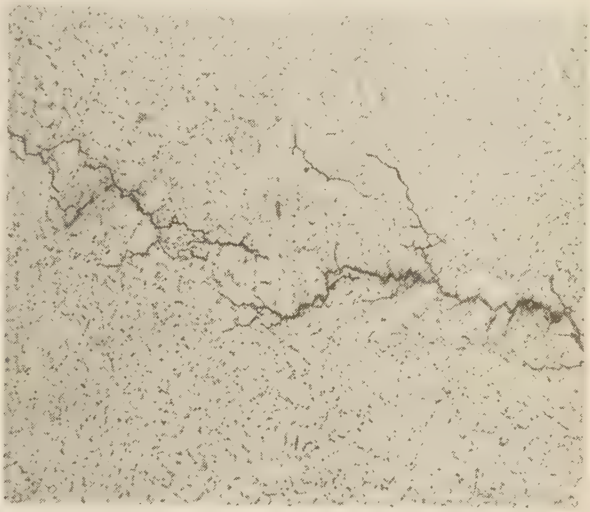

Fig. 37 Intercrystalline Crack in Embrittlement-Detector-Test Specimen of Alloy Steel Containing 5 Per Cent Chromium and 1.5 Per Cent Silicon
(Surface etched with nital; ×300.)

were found to be particularly susceptible to the action of the caustic solutions. For example, Fig. 37 of this discussion shows a crack in a badly shattered specimen of alloy steel containing 5 per cent chromium and 1.5 per cent silicon. These results do not support the belief that hydrogen is a major factor in the cause of boiler embrittlement. At the same time, they indicate that results from testing with hydrogen gas cannot be interpreted in terms of boiler embrittlement without danger of reaching erroneous conclusions.

Work of Other Investigators. In some instances, important references in the paper seem to have been interpreted improperly. Norton's experiments (377), for example, are cited to prove that sodium silicate, which sometimes accelerates boiler embrittlement, also accelerates hydrogen absorption.

Norton painted the outer surface of a steel radio tube with sodium silicate and measured the quantity of hydrogen penetrating to the interior when the tube was heated. He found that silicate-retained water reacted with the steel tube surface at the comparatively high temperature of 390 F. The experiment in no way measured the action of silicate on hydrogen penetration but instead measured its effect in retaining water at a high enough temperature so that a fairly rapid release of hydrogen could take place.

It is also noted that the work at the Bureau of Mines proved that sodium silicate was an accelerator of embrittlement while the work by Desch (396) in England stated that intercrystalline cracking could be produced in the absence of this salt. If the work at the Bureau of Mines is fully examined, it will be found that it does not disagree with the work in England. In the bulletin (398), cited by the author, appear the following statements:[14]

1 Under certain conditions sodium hydroxide alone can produce

[14] Reference (398), p. 45.

intercrystalline cracking. (See last two U-bend tests in Table 1, and detector tests 51, 29, and 45 in Table 9.)

2 Sodium silicate, especially the crystalline metasilicate, may accelerate cracking in some instances or broaden the range of conditions under which it occurs in others.

3 The effect of the silica in accelerating cracking appears to be more pronounced at 225 C and above than at lower temperatures.

4 The effect of the silica in accelerating cracking depends on the concentration of sodium hydroxide in contact with the steel. At high concentrations (especially at the higher temperatures) this latter material may attack so vigorously that general over-all corrosion proceeds as rapidly as grain-boundary corrosion. The addition of sodium silicate can suppress the over-all corrosion but still allow penetration at the boundaries. The influence of this salt in repressing over-all corrosion has already been illustrated.

5 The effect of silica on intercrystalline cracking depends on the applied or internal stress acting during the test. At the higher stresses any influence of the silica may be difficult to detect.

6 The influence of the silica depends on the physical and chemical characteristics of the steel being tested. On steels very susceptible to embrittlement (especially certain low-alloy steels) it is unnecessary to use this compound to produce cracking. On steels difficult to embrittle the silicate often appears to exert an accelerating effect.

7 At the lower boiler temperatures, certain types of sodium silicate may sometimes retard instead of accelerate intercrystalline cracking.

Particular attention should be directed to items 1 and 7 in this list since these show that Desch and the conditions of his test agree with those of the discussers. It is apparent that the action of the silicate is complex and that any attempt to treat it in too simple terms may lead to false conclusions.

The author's paper calls particular attention to the fact that tests with sodium nitrate are irrelevant in relationship to boiler embrittlement. He also feels that this salt ordinarily is not present in boiler water. It is now known that this belief is in error and simply existed because determinations of nitrate seldom are made in boiler-water analyses. Recent work has shown that high concentrations of sodium nitrate are present in a considerable number of boiler waters.

An interesting side light, concerning the comparison of cracking produced in sodium-nitrate solutions and caustic solutions is the development of the somewhat embrittlement-resisting Izett steel. The tests by which this steel was shown to be more resistant to intercrystalline cracking depended on the use of concentrated sodium-nitrate solutions. It is peculiar that this steel also exhibits increased resistance to caustic embrittlement if there is no relationship between action of the nitrate and the caustic solutions as postulated by the author.

The author in discussing the smooth, adherent, black-oxide scale that generally coats embrittled boiler seams states that these must "be distinguished sharply from the highly localized accumulations one must expect from selective corrosion, as required by the theory of Schroeder and his colleagues." The following quotations from the Bureau of Mines bulletin (398) may perhaps help to explain the term "selective corrosion:"

If the solutions indicated in Table 2 produce intercrystalline penetration without extensive over-all corrosion of the crystal faces, it should be possible to observe the progress of this selective activity. For this purpose, a specimen was polished for microscopic examination and bent over a bar in the center by means of bolts at the ends, as shown in figure 15. When the specimen was immersed in a boiling sodium hydroxide-lead oxide solution for 1 or 2 days a deep crack developed, as shown in figure 16. If, however, it was removed from the boiling solution after 2 or 3 hours it was found covered with a thin film of oxide. Under the microscope this film, which was blue to the naked eye, appeared gray, as illustrated in figure 17. The fine slip lines apparent in most of the crystals result, of course, from bending of the specimen to create stress. After this blue or gray film has formed, if the specimen is again immersed in the boiling solution for several hours rather broad white lines appear that outline the approximate position of the grain boundaries. These can be seen in figure 17. With further boiling, short, black lines appear in the broader white ones, which ultimately merge to form distinct cracks, as shown in the figure.

These experiments illustrate that development of the initial intercrystalline fissure is not a sudden or instantaneous process. Its growth can be followed over a considerable period. This type of initial attack would be expected from a selective corrosion process.

The silica, under certain conditions at high temperatures, and the oxidizing agents, in the U-bend tests at lower temperatures, probably function in a similar manner to accelerate the cracking; that is, they protect the crystal faces and prevent over-all corrosion while leaving the grain boundaries open to attack. The films they establish on the crystals may also create electrochemical forces that intensify action at the boundaries.

It is also of interest to note that corrosion in solutions starts at grain boundaries[15] but generally quickly spreads so that the over-all corrosion obscures the selective action.

Course of Embrittlement Investigations in the United States. A careful study of the literature, both with respect to boiler embrittlement and hydrogen embrittlement, has often led to the belief that the actions are related or may be identical. This was true in the studies undertaken by Parr and Straub, for their first paper expressed the belief that hydrogen gas was the primary factor in boiler embrittlement. It was not until they had completed many years of study on the subject that they abandoned this concept.

At the start of their investigation, Partridge and Schroeder made a survey of the literature which covered much of the ground now reviewed by the author. Their conclusions contained the following statement:

A thoroughly logical picture of the way in which the failure of boiler steel may be caused by hydrogen embrittlement may be built up on the basis of the information discussed in this report. When a boiler goes into operation its internal surfaces are immediately covered with a layer of magnetic oxide. This layer under ordinary conditions forms an almost impassable barrier between the boiler water and the steel. As the concentration of hydroxide increases, however, the layer becomes progressively less protective, and the reaction between the water and the steel increases its rate.

This chemical reaction produces hydrogen under condition equivalent to an extremely high gas pressure. The hydrogen diffuses into the steel and causes a definitely brittle condition, probably by setting up internal stresses at grain boundaries and cleavage planes.

Although the steel may be rendered very brittle by the absorption of hydrogen, no serious permanent damage is done until a localized stress at some point reaches a very high value, probably well up toward the ultimate strength of the metal. A piece of normally ductile steel subjected to a high local stress of this sort would yield slightly, automatically reducing the local stress by distributing it over a large section of metal. The embrittled steel, however, instead of gliding along slip planes in the manner of ductile metal, will ultimately crack at the point of high local stress. Although the initial failure may be only the boundary between two grains in the outermost surface layer, it represents a permanent damage to the steel. The high local stress, instead of being reduced in intensity, may then be transmitted to the next layer, with the result that the failure is eventually repeated here. As long as the concentration of hydrogen in the steel is sufficient to cause failure, the process of cracking will continue.

With this belief as a background, experiments were specifically designed to show that hydrogen was one of the primary factors in boiler embrittlement. Each of these experiments gave results in disagreement with preconceived ideas until, after an investigation of about 6 years, the conclusion was inescapable that the theory based on the action of hydrogen gas and developed through a study of the literature was incorrect. In fact, as the experiments were carried further, it was evident that some of the early interpretations of the work in the literature must necessarily be revised to quite different conclusions.

A brief description of one series of such experiments may be of interest. Fig. 38 of this discussion illustrates a piece of equip-

[15] "The Topochemistry of Corrosion and Passivity. III," by E. Pietsch, E. Josephy, B. Gross-Eggebrecht, and W. I. Roman, *Korrosion und Metallschutz*, vol. 8, 1932, pp. 57–68.

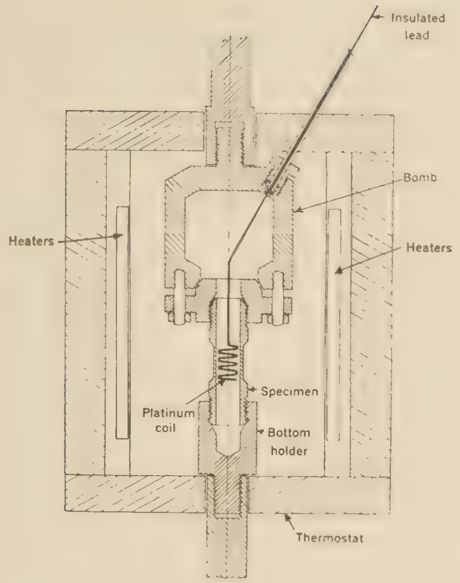

FIG. 38 EQUIPMENT FOR TESTING ACTION OF HYDROGEN AT ROOM OR ELEVATED TEMPERATURE

ment that was used at the Bureau of Mines to study the action of hydrogen gas on stressed steel. Hydrogen was released electrolytically at the surface of the specimen. When the apparatus was tested at room temperature it was found that the gas pressure rose to 1200 psi in 48 hr. The test specimen was then stressed to failure. The very brittle fracture that resulted indicated that the electrode was located satisfactorily with respect to the reduced section of the specimen.

Five specimens were then exposed to hydrogen attack at 250 C (482 F) for periods up to 122 hr. All specimens were stressed at about 50,000 psi during the tests, but no failures resulted. Each specimen was stressed to failure to complete the tests; in no case was any loss of load-carrying ability noted. There was some brittleness in the metal as would be expected from exposure under stress in the blue-brittle temperature region, but the fractures were much less brittle than that produced during the low-temperature test. A caustic solution in the same equipment produced failure in the specimen in 4 to 12 hr. The temperature for the latter tests was 250 C; the load on the test specimen was 50,000 psi; and no hydrogen was generated.

The author may now be in much the same position as were the investigators of the University of Illinois and the Bureau of Mines when they started their studies. Possibly through prolonged experimentation he, too, would revise his ideas and reach the conclusion that hydrogen was not as important as he now thinks it to be. His evident interest in the study of embrittlement, as shown by the large amount of work in the compilation of his bibliography, would certainly justify support for such work as he may wish to undertake. It is, of course, unfortunate that his bibliography does not refer to the important work on selective corrosion of Ulick Evans[16] in England, and Dix,[17] and Mears and Brown[18] at the research laboratory of the Aluminum Company of America.

Since previous investigators have been forced to discard the hydrogen-embrittlement theory, it might be best to undertake further investigation along this line without attempting to justify or prove preconceived ideas developed through literature study. A thoroughgoing investigation which gave equal credence, at least at the beginning, to all of the theories developed to explain boiler embrittlement would be very desirable. Careful experimental study of each of these would allow the investigators to reach conclusions that would be sound. The conclusions would be backed by a set of unprejudiced observations and would carry conviction to the people responsible for boiler operation.

In conclusion, it should be noted that boiler operators in the United States, working in close co-operation with research engineers, have found practical methods of preventing embrittlement cracking of boiler seams. These solutions of the problem have been achieved principally by tests with the embrittlement detector, which directly determines whether a boiler water can or cannot cause cracking. Although theoretical explanations are of considerable interest with relation to complete understanding of causes and effects, practical answers fortunately do not have to await their elucidation. For example, the sodium-nitrate treatment on the Chesapeake & Ohio Railway has effected a decrease in locomotive-boiler cracking, corresponding to a saving of more than 90 per cent of the engine time lost and the material and man-hours required for repair work. Under the present war conditions of transportation, this is indeed fortunate.

THOMAS DE VRIES.[19] There is additional evidence to substantiate the argument of the author that the diffusion of atomic hydrogen into steel is at the basis of the embrittlement phenomena. As early as 1864, Cailletet found that iron absorbed some hydrogen liberated by electrolysis. It has been known for a long time that iron is embrittled in pickling solutions by the hydrogen liberated in the reaction. The work of Bodenstein (153) and others has shown that atomic hydrogen permeates into the steel since the rate of permeation was proportional to the square root of the current density.

Borelius and Lindblom[20] have found a similar result for hydrogen produced by electrolysis. They also studied the permeation of molecular hydrogen into steel at different temperatures and pressures. The rate increased markedly with temperature, approximately as the fifth power of the absolute temperature. The rate also increased with the square root of the pressure thus proving definitely that the hydrogen must be in the atomic state before diffusion can occur.

Post and Ham,[21] investigating the permeation rate-pressure isotherms for hydrogen into nickel, found deviations from the square-root-of-pressure law, which they ascribed to the fact that some of the hydrogen was reacting with impurities in the nickel.

The author refers to Schenck for equilibrium constants for the reaction $Fe_3C + 2H_2 = 3Fe + CH_4$. Since Schenck's data were scattered and did not permit of very definite calculations, reference should be made to Watase,[22] whose data were critically examined by Kelley.[23] From his data for the interval 600–800 deg

[16] (a) "Metallic Corrosion, Passivity and Protection," by U. R. Evans, Edward Arnold & Company, London, England, 1937.

(b) "Velocity of Corrosion From the Electrochemical Standpoint," by U. R. Evans and T. P. Hoar, Proceedings of the Royal Society of London, England, vol. A137, 1932, pp. 343–365.

(c) U. R. Evans' discussion of "Intercrystalline Cracking in Boiler Plates," by C. H. Desch, Trans. North East Coast Institution of Engineers and Shipbuilders, vol. 55, 1938–1939, pp. D35–36.

[17] "Acceleration of the Rate of Corrosion by High Constant Stresses," by E. H. Dix, Jr., *Metals Technology*, T.P. 1204, vol. 7, 1940.

[18] "Causes of Corrosion Currents," by R. B. Mears and R. H. Brown, *Industrial and Engineering Chemistry*, vol. 33, 1941, p. 1001.

[19] Professor of Chemistry, Purdue University, Lafayette, Ind.

[20] "Durchdang von Wasserstoff durch Metalle," by G. Borelius and Sven Lindblom, *Annalen Physik*, vol. 82, 1927, pp. 201–226.

[21] Ref. (347), p. 599.

[22] "Studies on the Equilibrium: $Fe_3C + 2H_2 \rightleftarrows 3Fe + CH_4$," by Takeo Watase, *Journal of the Chemical Society of Japan*, vol. 54, 1933, pp. 110–132.

[23] "Contributions to Data of Theoretical Metallurgy. XIII. Thermodynamic Properties of Metal Carbides and Nitrides," by K. K. Kelley, Bureau of Mines Bulletin 407, 1933, p. 13.

K, the equilibrium constant is given by $\log K = -5937/T + 6.836$, whereas the author used the equation $\log K = -7990/T + 8.75$. Fortunately, the difference in the logarithm of the equilibrium constants (—2.69 and —4.07, respectively, at 350 C) calculated by the two equations is not sufficient to modify the author's argument.

However, it should be realized that the simple equation of Schenck is not suitable for extrapolation to higher temperatures and that this should be considered in any argument concerning decarburization. From the equation given by Kelley it can be calculated that the equilibrium constant, $K = (P_{H_2})^2/P_{CH_4}$, is 1.3×10^{-6} at 375 F (190 C), but 1.15 at 1100 F (600 C). Thus intercrystalline decarburization can definitely occur better at lower temperatures than at very high temperatures. This has been the experience of Potter, Solberg, and Hawkins (330).

R. B. Mears.[24] This paper on boiler embrittlement is a valuable contribution to our knowledge on this subject.

In the paper, several theories of boiler embrittlement and other forms of stress corrosion cracking are described. However, we do not believe that the author has discussed one of the most probable explanations for stress corrosion cracking. As was pointed out by Dix,[25] stress corrosion cracking is often the result of selective attack caused by local potential differences between different portions of the metal or alloy surface. In many cases such potential differences exist between material adjacent to the grain boundaries and the matrix of the grain. In other cases potential differences can exist between slip planes or other metallurgical characteristics of the material and a grain matrix. As a result of these potential differences, the anodic areas suffer electrolytic corrosion and, if the material is under stress, selective attack can cause cracking of the material in a relatively short period of exposure. It has been well established that this is the mechanism of the stress corrosion cracking of the aluminum-base alloys containing copper (Duralumin-type alloys), and it appears to be responsible for the season cracking of brass also.[26]

Furthermore, there is a good indication that the same mechanism applies to the stress corrosion cracking of 18–8 stainless steel.[27] It seems quite possible that the mechanism of boiler embrittlement is also an electrochemical phenomenon of this same type, local differences in potential causing selective attack at certain planes in the metal. Cracking will then occur along these planes.

It has been found in the case of nonferrous alloys that stress corrosion cracking may, in the early stages, follow an intergranular path and then later the metal may fracture transgranularly. Thus, there often exists some confusion of the path of the fracture, since it may start out following grain boundaries and then later on cut through grains themselves. Evidently this same confusion has existed in the case of boiler embrittlement. In any event, whether or not the mechanism of boiler embrittlement is indeed electrochemical, as is the case with many other types of stress corrosion cracking, the electrochemical mechanism of stress corrosion cracking should be mentioned in an extensive review, such as this one by the author.

The stress corrosion cracking of many types of nonferrous materials can be prevented by cathodic protection. The article to be protected is made the cathode, and current is passed to it through the electrolyte from a sacrificial anode. It would seem that a test involving cathodic protection could well be used to examine the hydrogen mechanism of boiler embrittlement. If a boiler steel which cracks readily in the absence of cathodic protection does not crack under the same conditions when it is being cathodically protected, there would appear to be no justification for the hydrogen theory, since during cathodic protection atomic hydrogen is being deposited on the surface of the steel and so should actually accelerate cracking, if the hydrogen theory is correct. We suggest that this experiment should be performed as soon as possible in order to test out the hydrogen mechanism.

The author mentions that the season cracking of brass remains unsolved. We do not believe this statement is correct. Instead there is very conclusive evidence that the season cracking of brass is an electrochemical phenomenon, and actual potential differences between grains and grain boundaries in brass have been measured in solutions which cause season cracking.

The author discusses at some length the results of other workers where occasionally one chemical is found to be an inhibitor of boiler embrittlement under some conditions but a stimulator under others. This type of action is not at all unusual for corrosion inhibitors. In fact, it is the general rule. Most corrosion inhibitors with which this writer is familiar accelerate corrosion under some conditions and retard it under others. In general, inhibitors can localize attack and therefore very often stimulate selective attack which causes intergranular corrosion or stress corrosion cracking. In many cases, very small or very large amounts of an inhibitor will be distinctly beneficial while intermediate amounts will be harmful. An explanation for this behavior can often be obtained from electrochemical measurements, and the mechanism of inhibitor action appears to line up with its effect on the anode and cathode reactions.

In studying the stress corrosion cracking of nonferrous materials, it has been found necessary to investigate the matter on a statistical basis. If a large number of "identical" specimens are exposed to a given corroding environment while under stress, a certain percentage of them only may be found to develop stress corrosion cracking. This percentage is often rather reproducible and may be used to express the probability of stress corrosion cracking of that particular alloy under those particular conditions. Slight changes in the composition of this alloy will affect the percentage of specimens which crack in the test.[28] A change in the nature of the corroding environment will shift the probability of stress-corrosion-cracking figures in one direction or another, so that they may approach either 0 or 100 per cent.

We suggest that in boiler-embrittlement studies it would be desirable to use a statistical picture also, since it seems probable that it will apply in this case as it has in the case of stress corrosion cracking of nonferrous materials.

E. P. Partridge.[29] Much that might have been presented as hypothesis or opinion by a less enthusiastic protagonist appears in this paper in the guise of indubitable fact. While a point-by-point debate would be as lengthy and as barren of result as the medieval argument over how many devils could dance on the point of a needle, the writer cannot forbear a token challenge on two representative passages.

In Part 3, the author refers to the 1941 Report on "Intercrystalline Cracking in Boiler Plate," by the National Physical Laboratory, England. As is natural, he has mentioned the emphasis placed by the British investigators upon hydrogen as the cause of embrittlement, even though the experiments were conducted at 420 and 470 C, far above the highest temperature, about 250 C, at which intergranular cracking has ever been observed in riveted boiler seams, and were characterized by de-

[24] Chief, Chemical Metallurgy Division, Aluminum Research Laboratories, Aluminum Company of America, New Kensington, Pa.

[25] "Acceleration of Rate of Corrosion by High Constant Stresses," by E. H. Dix, Jr., Trans. A.I.M.E., Institute of Metals Division, vol. 137, 1940, pp. 11–40.

[26] "Causes of Corrosion Currents," by R. B. Mears and R. H. Brown, *Industrial and Engineering Chemistry*, vol. 33, 1941, p. 1002.

[27] Ibid., p. 1003.

[28] See for example, ref. 25, Fig. 16, p. 34.

[29] Director of Research, Hall Laboratories, Inc., Pittsburgh, Pa.

carburization of the steel, which has never been found in actual cases. What the author neglected, in his zeal for the hydrogen theory, was what might be called the "minority report" from the National Physical Laboratory, which described the "oxidizing" type of cracking produced by very concentrated caustic solutions at temperatures more nearly representative of boiler conditions. The following statement appears in the British report: "Cracks of the hydrogen type were often found throughout the whole mass of a cold-worked specimen, while those of the oxidizing type existed in patches always in communication with the outer oxide coat." This latter condition more nearly resembles what is observed in actual cracked boiler plate than does the former.

The second item to be challenged is the discussion, also in Part 3 of the paper, relative to the development of high gas pressures in grain boundaries. The author calculates from the steam-iron equilibrium that very high pressures of hydrogen and fantastically high pressures of methane could develop in a closed boiler by reaction of the boiler water with the steel. He then postulates, however, that a grain boundary may be regarded as a "tiny theoretical boiler" in which the same high pressures previously calculated could develop as hydrogen, diffusing through the steel reduced magnetic iron oxide to produce water vapor.

Actually, the operating boiler is not a closed system, and the partial pressure of hydrogen both inside and outside of it is extremely low. Considering the steel for the moment simply as a solid permeable to hydrogen, the pressure of hydrogen at any point within the metal could scarcely exceed the very small pressure of hydrogen within the boiler, something of the order of 0.1 to 0.001 psi. According to the fundamental equilibrium indicated in Fig. 17, the pressure of water vapor developed in the grain boundary would then be almost vanishingly small.

To the extent that molecular H_2 diffuses less readily through the steel than does atomic H at any temperature, the development of higher gas pressures within the grain boundaries is possible. For example, it has been suggested by others that atomic hydrogen diffusing through steel may combine into molecular hydrogen at the discontinuity created by a grain boundary. Even so, the possible localized pressure, based upon data for the diffusion of molecular hydrogen through steel at boiler temperatures is very low in comparison with those suggested in the present paper.

The author should be given credit for a monumental bibliography, for some exceptional photomicrographs, and for his stimulating insistence that we re-examine our previous concepts concerning embrittlement. The importance of this paper really resides, however, in two suggestions which may be subjected to experiment. These are the predictions (*a*) that an inexpensive steel resistant to hydrogen embrittlement will likewise not be subject to cracking in boiler tube ends or riveted seams and (*b*) that inhibitors which will minimize the absorption of hydrogen by ordinary low-carbon steel will at least prolong its service life under embrittling conditions.

Fig. 15 of the paper, representing the work of Naumann, indicates that the addition of no more than 1 per cent of Ti, V, Zr, or Cb to low-carbon steel confers phenomenal stability against damage by hydrogen up to temperatures far beyond any attained in a boiler. If the author's thesis is correct, such low-alloy steels should not crack in the embrittlement detector devised by Schroeder and his associates when tested under conditions which cause rapid failure of low-carbon steel. Experiments in the detector seem to offer an obvious direct test which may forestall endless bickering over the minutiae of mechanisms by which embrittlement is produced.

P. B. Place.[30] The author presents a very interesting and persuasive argument for the role of hydrogen in boiler embrittlement, together with a rather disturbing vision of ultimate failure of all boiler steel. This paper will undoubtedly revive interest in a problem that has been solved several times but which refuses to stay solved.

Although there may be considerable evidence to indicate that a zone of hydrogen attack precedes chemical attack in many cases of intercrystalline failure, practical experience supports the conclusion of other investigators, that initial chemical attack is necessary. The progress toward failure after this initial attack is of more academic than practical interest.

There does not appear to be any appreciable danger due to attack of normal boiler water on the oxide-coated drum and tube surfaces. Boiler embrittlement has been significantly absent in boilers having welded seams and is usually associated with leaky riveted seams and cracks where chemical can concentrate to sufficient strength to attack the metal and its protective oxide film.

It would be pertinent to determine if excessive amounts of hydrogen accumulate in boilers held on bank for long periods, and particularly in boilers that use colloidal iron as an oxygen scavenger. If dangerous amounts of hydrogen are being generated in boilers, passing through drum shells, and reacting with nonmetallic inclusions to form high-pressure pockets in the steel, it seems logical to expect that the outer surfaces of the drum shells of old boilers would show evidence of blisters caused by rupture of pockets close to the outside surface.

The true mechanism of intercrystalline failure in boiler steel must be left in the hands of students of chemistry and metallurgy. In time, a mutually satisfactory explanation will be forthcoming. In the meantime, progress in welded construction, improvement in rolled joints, elimination of riveted seams, and further experience with inhibitors seem likely to reduce the embrittlement hazard to a minimum and eventually eliminate it as a boiler-operating problem.

D. H. Rowland.[31] In the writer's opinion, the author has made some rather positive statements in his paper considering the fact that he has offered very little irrefutable evidence to substantiate his claims.

Since the writer's knowledge of the subject of boiler embrittlement is general rather than specific, he hardly feels qualified to offer a discussion of the paper as a whole. Yet, from a purely metallurgical standpoint, which includes thermodynamic considerations, it would seem that the author has left himself open to criticism in attempting to draw an analogy between what happens to steel in the ammonia-synthesis process and in the ordinary steam boiler, where temperatures and pressures are not comparable.

On the basis of the author's explanations, it is difficult to understand what relationship, if any, exists between boiler embrittlement and hydrogen embrittlement due to pickling or cathodic treatment. It seems likely, and there has been considerable evidence by various experimenters to support the view, that nascent hydrogen may fill voids in steel, and by becoming molecular exert considerable localized pressure, but the role of hydrogen as a decarburizer at boiler temperatures and pressures has not, to my knowledge, been established. On the basis of decarburization experiments with wet hydrogen at elevated temperatures[32] it appears that hydrogen is a rather weak decarburizer at atmospheric pressure. From a thermodynamic standpoint, the decar-

[30] Research and Development Department, Combustion Engineering Company, Inc., New York, N. Y. Mem. A.S.M.E.

[31] Senior Metallurgist, Research Laboratory, Carnegie Illinois Steel Corporation, Pittsburgh, Pa.

[32] "Grain-Size and Its Influence on Surface Decarburization of Steel," by D. H. Rowland and Clair Upthegrove, Trans. A.S.M., vol. 24, 1936, p. 96.

burizing power of hydrogen should and does become more pronounced as temperature and pressure are increased, but in so far as steam boilers are concerned, it is doubtful if there is a sufficient combination of temperature and pressure materially to increase the decarburizing tendency of hydrogen over that at atmospheric pressure. Certainly there is no apparent evidence of decarburization in Figs. 30 and 31 of the paper, notwithstanding the fact that one would expect to see some indication of decarburization where such a condition exists.

The author's statement, "Accordingly, the whole study of boiler embrittlement seems to lie within the single illustration, Fig. 27," is certainly open to criticism. No "story" as complex as boiler embrittlement can be summed up in one photograph, or in a dozen photographs for that matter. If the mechanism of boiler embrittlement were as simple as the author has inferred, it is astonishing why so many boilers could have been in service for years without failure.

On reading this paper, one gets the distinct impression that the author is indirectly advocating the use of certain low-alloy steels for boiler construction on the basis that their carbides are relatively stable in the presence of hydrogen. Such a conclusion, in the absence of considerable supporting evidence, does not appear to be warranted.

Author's Closure

For several years the author has tried by oral argument to point out, with little apparent success, one principal fact regarding previous studies of boiler embrittlement: That, *whether hydrogen is the cause of boiler embrittlement or not, the hydrogen theory has not as yet been tested by valid means.*

Where the paper seems to overemphasize the hydrogen aspect, it does so purposefully for the reason that other theories have been similarly overemphasized when they eliminated the principal competing theory without full cognizance of the record. By this, reference is made specifically to the complete disregard of the many established cases in which hydrogen does cause intercrystalline failure of a type highly resembling boiler embrittlement, and generally to the many other points made in the paper which need no repetition here.

Which theory is ultimately to be proved is of little consequence; the important thing is to prove one or the other. That depends upon a correct diagnosis, which did not stand in the literature. Admittedly, the present paper may likewise contain errors; but if it does nothing other than to cause the correct answer to be developed, and to bring chemists and metallurgists into agreement on certain basic facts, it is well worth the time and effort spent preparing it.

Consequently, Schroeder and Berk's extended criticism of this paper constitutes a valuable contribution, in that respect, to the study of boiler embrittlement. They are among the foremost chemists in this field; and their careful study of this metallurgical approach promises real advancement.

However, some weaknesses still stand in their argument and warrant mention to prevent continued confusion. Because they present their argument stepwise, let us consider it in turn by its steps.

Step 1. There was no need to review in this paper the manner in which concentration occurs, since that matter is satisfactory to everyone as it stands.

Step 2. This paper does not "... postulate an agency that can start cracks or fissures from within the metal." It is plainly stated throughout the paper that hydrogen does not cause observable cracking, but simply renders the metal brittle, preventing a ductile response to stress. The subtle fissures through which the gaseous molecules escape to the surface should not be confused with visible cracking. The crack leading to the surface, which we all see and agree upon, first develops from the operating stresses applied to this brittle metal, and not from chemical leaching of grain-boundary material, which is a secondary, not a primary, reaction, according to the argument.

Consequently, Schroeder and Berk's criticism up to Step 3 is inconsequential.

Step 3. As for Step 3, Schroeder and Berk, who suggest that the author confounded kinetics with equilibrium, persist in overlooking all aspects of equilibrium, especially its relationship to kinetics.

Quoting Lewis and Randall's standard text on "Thermodynamics," we find that "... in all thermodynamics there is no concept more fundamental than this [equilibrium]," and that "... every system which has not reached a state of equilibrium is changing continuously toward such a state with greater or less speed." This "speed" is the rate of the reaction under consideration, and its study constitutes "kinetics."

Although the equilibrium of a system tells us nothing about the "rate" of the individual reactions comprising it, the equilibrium rigorously dictates the *only possible direction* of each reaction, and the *ultimate goal* toward which those reactions strive. Consequently, because we know little of the reaction rates concerned in boiler embrittlement, our first step should be to determine where those reactions are trying to lead us. That information, fortunately, is available and is discussed in the paper.

For example, let us pursue Schroeder and Berk's own calculations to their logical conclusion. First, however, we find them a bit mystifying, since the diffusion equation

$$D = K/d[p^{1/2} \cdot e^{-E_0/(2kT)}]$$

is expressed in terms of volume of gas per area of surface per period of time per "thickness of plate." Their only mention of plate thickness occurs later in remarking on Naumann's experiment. A check of their calculations, however, shows that the figure they obtain applies to steel about $^1/_5$ in. thick, which is somewhat thin for boiler plate. Increasing thickness as shown by d in the equation decreases the gas loss linearly.

Nevertheless, let us accept the figure of 400 psi of H_2 that they give as a maximum value for the infiltrating gas and study its passage through the steel. The work of Ham (325), which they cite, went on to prove quite conclusively our basic assumption that the dissolved H reacts to some extent with the nonmetallics in the steel in the temperature range of 435 down to 260 F—boiler temperatures.

From the corrected methane equilibrium kindly supplied in the foregoing discussion by De Vries

$$\log K = -5937/T + 6.836$$

for the reaction

$$Fe_3C + 2H_2 = 3Fe + CH_4$$

$$K = (H_2)^2/CH_4$$

we find that 400 psi of H_2 in the steel at 390 F will engage carbon kinetically in an effort to reach the equilibrium pressure of *5,625,000,000 psi of CH_4!*

As for its reaction with iron oxide, we cannot expect such a large contribution. The equilibrium

$$Fe_3O_4 + 4H_2 = 3Fe + 4H_2O$$

$$K = H_2/H_2O$$

lies toward lower H_2O values as the temperature is lowered. The contribution from H_2O would not exceed 50 psi.

But let us re-examine this C-H reaction. The figure of nearly

6,000,000,000 psi wastes itself on the ordinary reader. Whereas Schroeder and Berk selected the H_2 pressure of 400 psi as a maximum value and tried to show its inefficacy, let us calculate what the *minimum* H_2 pressure actually need be to create embrittling stresses from methane alone.

Pressures greater than approximately 45,000 psi will deform ordinary boiler steel. If we use the equation supplied by De Vries and insert 45,000 psi for P_{CH_4} and calculate P_{H_2}, we obtain the enlightening result that $P_{H_2} = 1.1$ psi.

This value actually lies so close to the measured range of hydrogen pressures in boilers that it nearly eliminates the necessity for discussing atomic surface conditions, although the author still believes they may differ considerably from the distant atmosphere and will greatly increase that figure.

At the steel surface underneath concentrated alkali, however, any gaseous H_2 showing as bubbles would have a pressure of at least 1 atm, or 15 psi, still disregarding surface phenomena. Because of the quadratic relationship between H_2 and CH_4, the potential CH_4 pressure within the underlying metal would then be over 200 times 45,000 psi. That added factor may be the one which bridges the kinetic inefficiencies of the various reactions and allows embrittlement only at seats of such alkaline concentration. In any event, that figure is a "safety factor" of 200 in favor of the present argument.

As for the lack of visible decarburization in embrittled boiler steel, that point similarly disintegrates as a proof against the gas mechanism. At higher temperatures, actual escape of the gas from the steel permits decarburization to progress until it is visible at the time of failure. But that is an action which can be subsequent to embrittlement, especially at the lower temperatures of boiler operation where gaseous congestion within the metal is less readily relieved. There the gas necessary to cause the embrittlement may be obtained from a lesser quantity of C than is visible as a solid phase change.

For example, when the C as Fe_3C in pearlite reacts with H to form CH_4 and Fe, the density of the solid phase increases from 7.73 to 7.87 g per cc. That allows 0.0162 per cent of the original volume for the gas phase, which represents a compression of 80,000 atm, or 1,150,000 psi at 390 F. Once again we are faced with the fact that fantastic possible pressures of gas trapped within steel are a commonplace.

Obviously, there are vacuities available within the metal other than the one that might result from the phase transformation. However, under the highest magnification of the light microscope, grain boundaries still appear reasonably sound. If we make the large allowance of 0.1 micron, a figure just beyond microscopic resolution, as the thickness of a flat vacuity in a grain boundary, and consider the decarburization of an adjoining lamella of Fe_3C, then we find that 45,000 psi of CH_4 would develop at 390 F when that single cementite lamella was decarburized to a depth of only 0.25 micron. Such a feature would be entirely unrecognizable with the highest possible magnification of the light microscope. Perhaps with the electron microscope, decarburization can be detected. The matter should be investigated.

Step 4. Here the worry about diffusion is groundless. Carbon *is* removed from steel by hydrogen, and the path *is* intergranular. Whether the C diffuses to the grain boundary to react with H there, or whether it reacts within the grain and then diffuses along the internal surfaces depicted in Figs. 4 and 20 of the paper, is academic.

Step 5. The most unfortunate fallacy in all of Schroeder and Berk's discussion concerns their Fig. 36 and is representative of the severe handicap a chemist is under when studying purely metallurgical problems. The fracture they show is simply a typical metal fracture, as they point out in which the concentration of stress set up by the cut caused a typical fracture type. A metallurgist would not have bothered making that test, since its behavior is rudimentary knowledge.

That is why Fig. 33 of the paper was also presented, because these boiler fractures are not ordinary. It is no news to reproduce their Fig. 36, with its commonplace fracture type; but it is news when the same type of hammer blow reveals a ductile zone *between* two brittle zones, as shown in Fig. 33, especially when the concentration of alkali coincidentally was on two sides of this plate, in contrast to the single-sided attack of the plate in Fig. 27 of the paper.

That was also why the trouble was taken to include a stereographic projection and several micrographs, since the character of the metal was as different as day and night from the normal boiler steel shown in their Fig. 36. It would pay each reader well to "fuse" the stereographs in Fig. 28, and see for himself how ridiculous it is to compare that deteriorated metal with the normal steel of their specimen.

Fig. 39 is added here to explain this zoning phenomenon caused

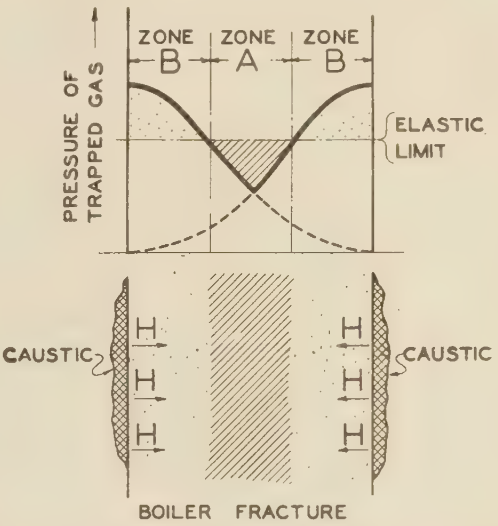

Fig. 39 Schematic Diagram Showing Fractured Section of Boiler Plate (Below) Attacked From Both Sides by Caustic With a Depiction (Above) of the Internal Gas Pressures Set Up by Transfusing Hydrogen

(When gas pressures exceed the elastic limit of the steel, brittle zone B results. Ductile zone A includes the metal which is not so stressed.)

by gaseous congestion. Hydrogen, fed into the steel by the alkali, sets up internal gaseous congestion which is a maximum nearest to the source of the hydrogen and tapers off in pressure as it progresses into the steel (refer back to Fig. 3 of the paper). If the pressure exceeds the elastic limit of the steel near the surface of entrance, embrittling that section (zone B), a boundary will later be reached along the line of diffusion beyond which the gas pressure no longer exceeds the elastic limit of the steel and the ductility remains unimpaired. That boundary marks the beginning of the gray zone A. In that zone some cleansing action might be evident, but the gaseous products are not under sufficient pressure to cause embrittlement.

With unilateral diffusion, the gray zone always extends to the opposite surface. The bilaterally attacked specimen in Fig. 33 of the paper pertains to the diagram in full, showing a central ductile zone; whereas the specimen in Fig. 27 pertains only to one half of the diagram, the ductile zone extending to the far surface.

Step 6. Once again Schroeder and Berk are under a misconception concerning the role of hydrogen in causing cracking.

Since this was corrected in discussing Step 2, no further comments will be made.

As for their tests to check Naumann's work, the only alloying elements they used were the ones shown by Naumann to be least effective. Cr, one of the best of this minority group, was not designated by Naumann to contain the unusual silicon content of 1.5 per cent. As for Mo, what alloying content did they use? A misjudgment in similar early tests with vanadium was pointed out in the paper.

Also, if they tested these steels sufficiently to warrant their conclusions, a table of actual results would be valuable. Until satisfactory and systematic tests have been run on Ti and Cb steels, judgment should be withheld.

In closing the discussion of Schroeder and Berk's critique, several items might be mentioned.

1 The disagreement expressed between their work and that of the British was the opinion of the British (361), not the author's.

2 It is *not* peculiar that Al-killed steel should show improvement both to caustic and to nitrate attack. Each is basically intergranular corrosion; but for caustic, the corrosion may be indirect and gaseous, according to the present argument, thereby involving a mechanism distinct from that in nitrate cracking.

3 It is surprising to find these writers referring to the path they followed in testing the hydrogen theory and suggesting that the present paper parallels their first step taken in 1935, a step which finally led to their unfortunate conclusion in 1941 that intergranular failure cannot be caused by hydrogen. Where would the hydrogenation and ammonia-synthesis industries be today if they had proceeded on the basis of such an erroneous conclusion?

If these earlier writers had attempted to prove that boiler embrittlement is not intercrystalline hydrogen attack, the material in the present paper would then more or less represent one man's viewpoint; but they apparently failed completely to recognize that such a thing as intercrystalline H attack even existed. In addition, they were confused on the whole general behavior of gas-metal systems, drawing the numerous wrong conclusions which are discussed throughout the paper.

The author agrees with their decision expressed long ago that a thoroughgoing and sound investigation of the H mechanism be made, and suggests it at last be undertaken, since the subject has never received that treatment. Whether the hydrogen mechanism be proved unimportant or not, it remains to be so proved; and until then the author will hold the conclusion of the paper that by far the most likely mechanism for boiler embrittlement is the gas mechanism.

On the other hand, the present paper does not exclude the fact that caustic, when it does attack steel, may do so preferentially, especially when the grain boundaries are under great stress from trapped gases. Electrochemical corrosion may be a contributing factor to boiler embrittlement.

De Vries, Place, and Mears are to be thanked for their contributions to the discussion. Mears' assumption that boiler embrittlement is synonymous with stress-corrosion cracking is all right, but the present paper is based on the probability that the mechanism is of another type. The electrochemical theory is of little use in explaining the cracking of metals by other liquid metals and has nothing to do with intercrystalline gaseous attack, whether by O, H, S, or other gases. Consequently, every intergranular failure need not be lumped under an electrochemical subheading. If boiler embrittlement does belong there, the matter has already received ample discussion by others.

Mears' proposal for cathodic protection of the steel is very interesting, although the results of such a test might not be so simply interpreted as he suggests.

In Rowland's and De Vries' discussions, we find the interesting situation wherein one says H is a better decarburizer as the temperature goes up, and the other says it's better as the temperature goes down! A quick glance at the methane equilibrium shows that De Vries is the more correct in so far as the driving power or free energy of the reaction is concerned; but consideration of thermal energy shows that at lower temperatures the reaction may become so sluggish that it is quantitatively unimportant in *removing* carbon. Loose usage of the word "decarburization" both for the reaction and for the removal of CH_4 is to blame.

Correspondingly, Rowland is further confused when he speaks of hydrogen as a ". . . rather weak decarburizer at atmospheric pressure," on the basis of experiments he conducted at temperatures where CH_4 is almost nonexistent. And that ". . . from a thermodynamic standpoint, the decarburizing power of hydrogen should and does become more pronounced as temperature and pressure are increased." The *reaction* or the *removal*, and in what temperature range? The reacting power of H should *not* increase with increasing temperature. The free energy of methane formation decreases with increasing temperature. True, the *removal* is favored by heating; but even then that beneficial effect is soon counteracted by the sharp loss in the free energy of the reaction. In boiler embrittlement, the removal of the reaction product is only of secondary interest.

Rowland's objection to the present picture of boiler embrittlement because it is too simple takes the author so unawares, as does his worry about all the boilers which did not fail, that on answer will be made!

Partridge's criticisms, excepting his delightfully diversionary comments on devils and needle points, duplicate the remarks of others and therefore stand answered in the foregoing discussion.

A Sampling Inspection Plan for Continuous Production

By H. F. DODGE,[1] NEW YORK, N. Y.

The plan of sampling inspection, described for individual parts, subassemblies, finished articles, and the like, is intended primarily for use in process inspection where there must be assurance that the percentage of defective units is kept down to a low prescribed figure. The system is particularly applicable to products manufactured by conveyer or similar straight-line continuous processes. The object of the plan is to establish a limiting value of "average outgoing quality," expressed as a percentage of defective units, which will not be exceeded no matter what quality may be submitted to the inspector.

1 INTRODUCTION

THIS paper presents a plan of sampling inspection for a product consisting of individual units (parts, subassemblies, finished articles, etc.) manufactured in quantity by an essentially continuous process.

The plan, applicable only to characteristics subject to nondestructive inspection on a "Go – No Go" basis, is intended primarily for use in process inspection of parts or final inspection of finished articles within a manufacturing plant, where it is desired to have assurance that the percentage of defective units in accepted product will be held down to some prescribed low figure. It differs from others which have been published[2,3] in that it presumes a *continuous flow of consecutive articles or consecutive lots* of articles offered to the inspector for acceptance in the order of their production. It is accordingly of particular interest for products manufactured by conveyer or other straight-line continuous processes.

In operation, the plan provides a corrective inspection, serving as a partial screen for defective[4] units. Normally, a chosen percentage or fraction f of the units are inspected, but, when a defective unit is disclosed by the inspection it is required that an additional number of units be inspected, the additional number depending upon how many more defective units are found. The result of such inspections is to remove some of the defective units, and the poorer the quality submitted to the inspector, as measured in terms of per cent defective, the greater will be the corrective or screening effect. The object of the plan is the same as that incorporated in some of the sampling tables already published,[5] namely, to establish a limiting value of "average outgoing quality" expressed in per cent defective which will not be exceeded no matter what quality is submitted to the inspector. This limiting value of per cent defective is termed the "average outgoing quality limit (AOQL)."

The theoretical solution treats the case of inspecting a continuous flow of individual units and is based on the distribution of *random-order* spacing of defective units in product whose quality is statistically controlled.[6] Part 3 of the paper extends the application of the method to a continuous flow of individual lots or sublots of articles.

2 INSPECTION OF A FLOW OF INDIVIDUAL UNITS

Inspection of One Characteristic. Consider first the inspection of a flow of individual units, offered consecutively in the order of their production. Assume that inspection is to be made for only one quality characteristic, so that interest will be centered on one kind of defect. Subsequently, consideration will be given to the procedures when inspection is made simultaneously for several kinds of defects.

Procedure A. The procedure is as follows:

(*a*) At the outset, inspect 100 per cent of the units consecutively as produced and continue such inspection until i units in succession are found clear of defects.

(*b*) When i units in succession are found clear of defects, discontinue 100 per cent inspection, and inspect only a fraction f of the units, selecting individual sample units one at a time from the flow of product, in such a manner as to assure an unbiased sample.

(*c*) If a sample unit is found defective, revert immediately to a 100 per cent inspection of succeeding units and continue until again i units in succession are found clear of defects, as in paragraph (*a*).

(*d*) Correct or replace with good units, all defective units found.

Protection Provided by Plan. The inspection plan is defined by the two constants, f and i, which can be altered at will. For given values of f, i, and p (incoming fraction defective), there will result for product of statistically controlled quality a definite average outgoing fraction defective (average outgoing quality, AOQ). For given values of f and i, the AOQ will have a maximum for some particular fraction defective p_1 of incoming quality. As already noted, this maximum is referred to as the average outgoing quality limit (AOQL). For all other values of incoming fraction defective p greater or less than p_1, the AOQ will be less than AOQL. Many combinations of f and i will result in the same AOQL.

The protection offered by the plan discussed here can thus be expressed in terms of the AOQL, in per cent defective.

Theoretical Framework. We are concerned with the spacing between defective units when the individual units are arrayed in the order of their production, as shown in Fig. 1. If the manufacturing process is statistically controlled so that the

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[1] Quality Results Engineer, Bell Telephone Laboratories.

[2] "Single Sampling and Double Sampling Inspection Tables," by H. F. Dodge and H. G. Romig, *Bell System Technical Journal*, vol. 20, 1941, pp. 1–61. An unpublished paper by Prof. Walter Bartky, (developed when he was associated with the Western Electric Company, 1927), provides a continuous multiple sampling plan involving two factors: f, as used here, and i, the number of units in a "compensating sample" required to be inspected for each defective unit found.

[3] "Standardized Inspection," by Lieut. R. J. Saunders, *Army Ordnance*, vol. 24, 1943, pp. 290–292; "Quality Through Inspection," by G. Rupert Gause, *Army Ordnance*, vol. 24, 1943, pp. 117–120.

[4] A unit of product that fails to meet the requirement for a characteristic is classed as nonconforming with respect to that characteristic, and for convenience is referred to as "defective." Thus, a deviation from a specified requirement or from accepted standards of good workmanship is termed a "defect."

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[5] Loc. cit., H. F. Dodge and H. G. Romig.

[6] "Statistical Control," as defined in the literature; see "Statistical Method From the Viewpoint of Quality Control," by W. A. Shewhart, The Graduate School, U. S. Department of Agriculture, 1939.

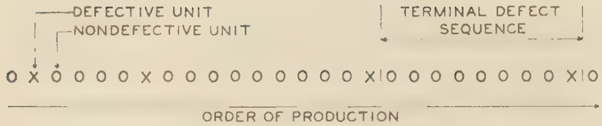

FIG. 1 SPACING OF DEFECTIVE UNITS

probability of producing a defective unit is constant and equal to p, then defective units will have an order spacing of a random character which is expressible in terms of certain probability laws. Product turned out by such a process will be referred to as having a process average fraction defective p. The "event" of particular interest is a "terminal-defect sequence" of $i + 1$ successive units following the observance of a defect, comprising a succession of i nondefective units followed by a defective unit, as shown in Fig. 1. The totality of all possible such sequences, where i varies from 0 to ∞, constitutes the universe of events under consideration.

Each such sequence of $i + 1$ units, comprising i successive nondefective units followed by a defective one, has a definite probability of occurrence, for a process average fraction defective, p. The complete set of such probabilities for all possible sequences, having respectively $i = 0, 1, 2, 3, \ldots \infty$, defines a probability distribution[7] of random-order spacing of defects in uniform product. This is shown in Table 1 in which 0 represents a nondefective unit, X represents a defective one, p is the fraction defective, and $q = 1 - p$.

[7] "Due Nuovi Criteri di Controllo Sull 'andamento Casuale di Una Successione di Valori," by V. Romanovsky, *Giornale dell'Instituto Italiano degli Attuari*, 1932, discusses this probability distribution of spacing of events, referring to the spacing as the "length of a partial series." Our term "terminal-defect sequence" has the same significance as his term "partial series." See also "Note on Theoretical and Observed Distributions of Repetitive Occurrences," by P. S. Olmstead, *Annals of Mathematical Statistics*, vol. 11, 1940, pp. 363–366; "The Distribution Theory of Runs," by A. M. Mood, *Annals of Mathematical Statistics*, vol. 11, 1940, pp. 367–392.

TABLE 1 PROBABILITIES OF OCCURRENCE OF DEFECT SPACING

Sequence	Defect spacing (No. of units in sequence)	No. of nondefective units before finding the next defect	Probability of occurrence	No. of term in the power series
X	1	0	p	1st
$0X$	2	1	pq	2nd
$00X$	3	2	pq^2	3rd
$000X$	4	3	pq^3	4th
$0000X$	5	4	pq^4	5th
.	.	.	.	.
$000\ldots.0X$	$i + 1$	i	pq^i	$(i + 1)$st
.	.	.	.	.

These probabilities are the successive terms in the infinite power series

$$\left.\begin{array}{l} p + pq + pq^2 + pq^3 + \ldots \\ \text{or} \quad p(1 + q + q^2 + q^3 + \ldots). \end{array}\right\} \ldots\ldots\ldots\ldots [1]$$

The sum of this series is $p\left(\dfrac{1}{1-q}\right) = 1$; i.e., the total probability for all possible sequences is unity (as it should be).

The sum of the first $i + 1$ terms of the series is the probability of occurrence of a "terminal-defect sequence" (defect spacing) of $i + 1$ units or less. The sum of the first i terms is the probability, P_1, of failing to find the next i units clear of defects, which is

$$P_1 = \sum_{j=}^{j=i-1} pq^j = 1 - q^i \ldots\ldots\ldots\ldots [2]$$

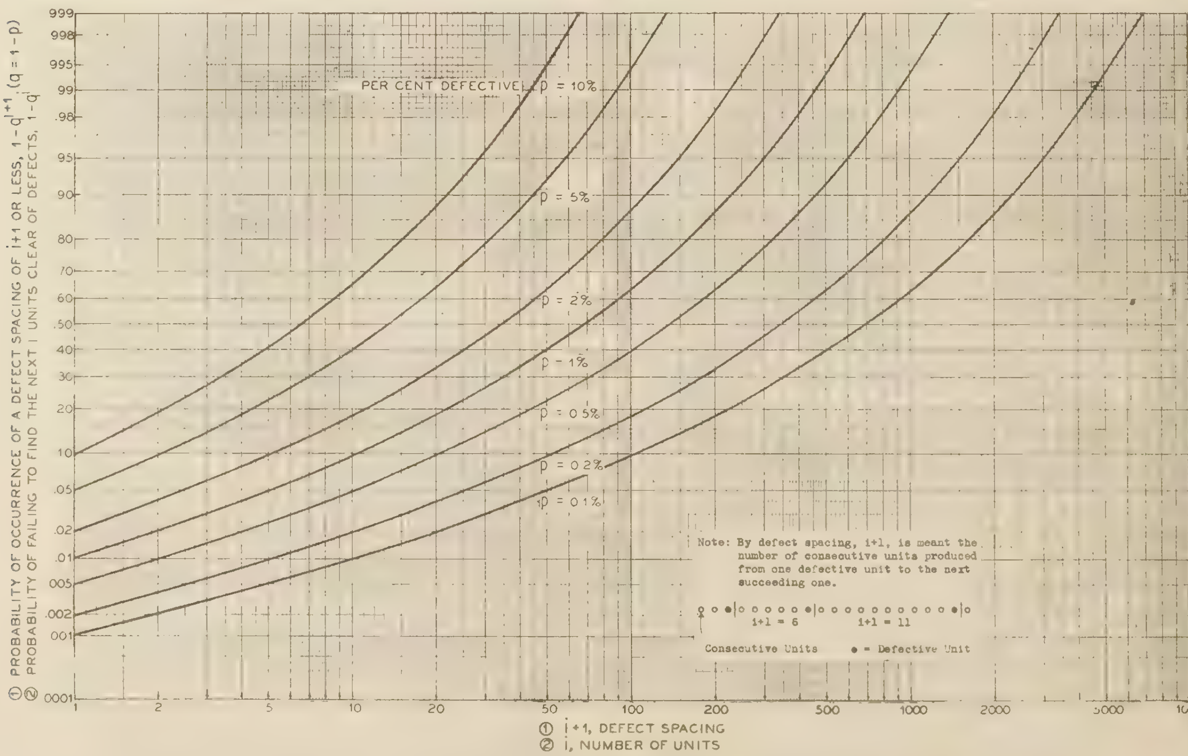

FIG. 2 CURVES DEFINING DISTRIBUTION OF RANDOM-ORDER SPACING OF DEFECTS IN UNIFORM PRODUCT

In turn, the sum of all terms beyond the ith term is the probability of finding 0 defects in the next i units, which is

$$Q_1 = 1 - P_1 = q^i \quad [3]$$

These results and the power series [1] enter into subsequent portions of the discussion. The curves of Fig. 2 give values of $1 - q^i$.

Average Outgoing Quality. Suppose a plan is selected, choosing specific values of f and i. For given values of i and p, there will be an expected average number of units, u, inspected following the finding of a defect. Likewise, for given values of f and p, there will be an expected average number of units, v, that will be passed under the sampling procedure before a defect is found. The latter average number includes the sampling units actually inspected as well as the uninspected units produced between successive sample units.

The average fraction of the total product units inspected in the long run is

$$F = \frac{u + fv}{u + v} \quad [4]$$

It is now assumed for purposes of solution that the inspection operation itself never overlooks a defect and that all defective units found during the inspection of f and i will be corrected or replaced by good units.[8]

As a result of the screening effect of the inspection, the average outgoing quality, AOQ, designated p_A, is related as follows to the incoming quality p

$$p_A = p(1 - F) = p\left(1 - \frac{u + fv}{u + v}\right) \quad [5]$$

Determination of u. The average number of units, u, inspected on a 100 per cent inspection basis, following the finding of a defect, is a function of i and p, and may be determined from a consideration of two power series, one limited and the other infinite.

Once the 100 per cent inspection starts, there are several things that can happen before i units are found clear of defects. The first i may be found clear; or 1, 2, 3, or more defects may be found before finally a run of i units is found clear.

One of the quantities to be determined is the *average number of units* inspected in a "failure sequence;" that is, one terminating in a defect and comprising i or less units. This average number, designated as h, is the average of the distribution made up of the first i terms of the power series [1]. The average is

$$h = \frac{p}{1 - q^i}(1 + 2q + 3q^2 + 4q^3 + \ldots + iq^{i-1}) \quad [6]$$

where the denominator is the sum of the probabilities for the first i terms. This may be evaluated as follows

$$h = \frac{p}{1 - q^i}\frac{d}{dq}(1 + q + q^2 + q^3 + \ldots + q^i)$$

$$h = \frac{p}{1 - q^i}\frac{d}{dq}\left[\frac{1 - q^{i+1}}{1 - q}\right]$$

$$= \frac{1}{p(1 - q^i)}[1 - q^i(1 + pi)] \quad [7]$$

Note that if pi is small compared with unity, h is approximately $1/p$.

The next step is to determine the *average number of failure sequences* that will be encountered before finding i units clear of defects. This average number, designated as G, may be found from the probability distribution of all possible numbers of failure sequences, expressed by the infinite series

$$Q_1(1 + P_1 + P_1^2 + P_1^3 + \ldots) \quad [8]$$

where P_1 is given by Equation [2], $Q_1 = 1 - P_1$, as given by Equation [3], and the successive terms are the probabilities of occurrence of 0, 1, 2, 3, etc., failure sequences before finding i units clear of defects. The average number of failure sequences, G, is given by the sum of the infinite series

$$G = Q_1(0 + 1P_1 + 2P_1^2 + 3P_1^3 + \ldots)$$
$$= Q_1P_1(1 + 2P_1 + 3P_1^2 + 4P_1^3 + \ldots) \quad [9]$$

Summing the series, we have

$$G = Q_1P_1\frac{1}{(1 - P_1)^2} = \frac{P_1}{Q_1} = \frac{1 - q^i}{q^i} \quad [10]$$

Now u, the average number of pieces inspected following the finding of a defect, is made up of a number of failure sequences followed by a run of i units clear of defects. Using the average values of G and h just found, we have

$$u = Gh + i = \frac{1 - q^i}{pq^i} \quad [11]$$

Determination of v. The average number of units, v, that will be passed in a period of sampling inspection will be $1/f$ times the average number of individual sample units inspected in such periods. Here again the solution will depend upon the random-order spacing of defects in uniform product. Whether the individual units selected during the sampling inspection procedure are selected by a random-spacing device, or by any other means which will prevent known bias in the sample, we may assume that defects will be found to occur in accordance with the distribution of random-order spacing, defined by the terms of the power series [1]. The average number of sample units inspected in a period of sampling inspection will thus be the average defect spacing for a product having fraction defective, p, which is given by the infinite series.

$$H = p(1 + 2q + 3q^2 + 4q^3 + \ldots) \quad [12]$$

Summing the series, we have

$$H = \frac{p}{(1 - q)^2} = \frac{1}{p} \quad [13]$$

and the value of v is found to be

$$v = \frac{H}{f} = \frac{1}{fp} \quad [14]$$

Determination of f and i for a Given Value of AOQL. From the considerations previously given, the average fraction of the product inspected, F, and the value of average outgoing quality, p_A, can be determined for any given values of p, f, and i. Substituting in Equation [5], the values of u and v given in Equations [11] and [14], we have

[8] The assumption that the inspection operation is perfect cannot be made without reservation. Machine inspection devices have their margins of error. Also, inspection fatigue prevents 100 per cent manual and visual inspections from insuring perfection, particularly if such inspections continue over a considerable period of time. But the efficiency of the latter inspections is generally higher when an interest incentive is provided as is usually the case in sampling inspection plans where the extent of such inspections hinges on their findings.

The solution given assumes correction or replacement of defective units. Where it is expedient to reject such units and not replace them Equations [19] to [22], inclusive, should be modified by replacing i by $(i - 1)$.

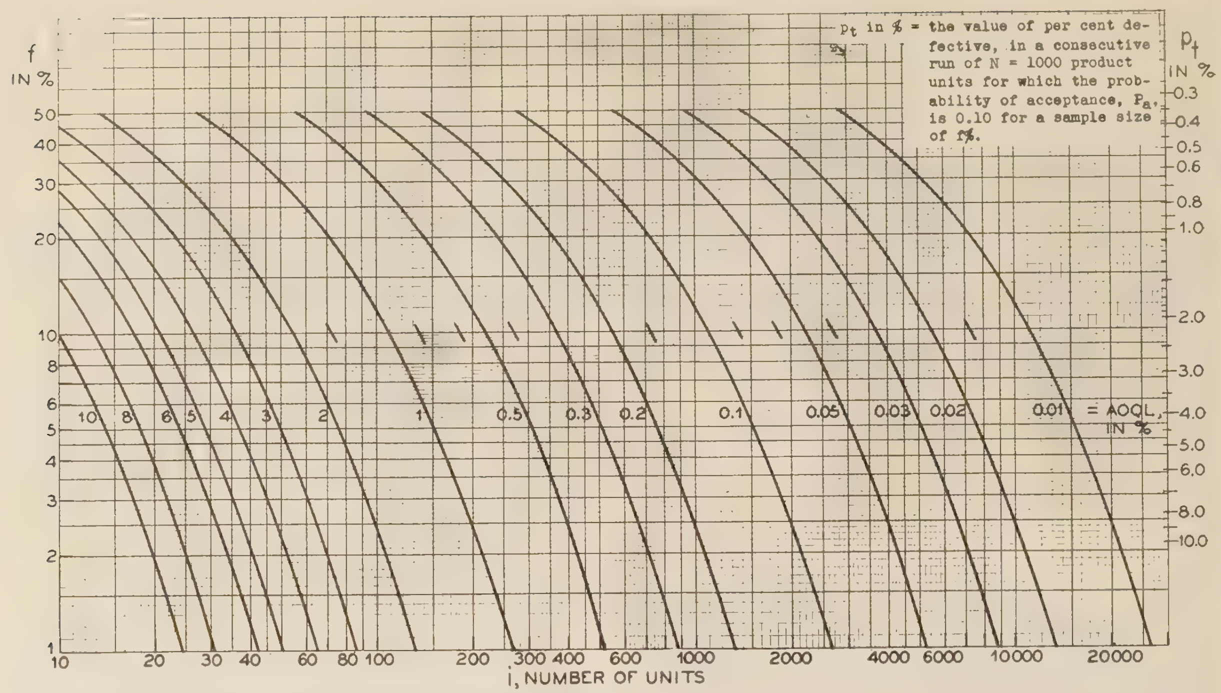

Fig. 3 Curves for Determining Values of f and i for a Given Value of AOQL

$$p_A = p\left[1 - \frac{f}{f + (1-f)(1-p)^i}\right] \qquad [15]$$

The average outgoing quality limit, AOQL, (p_L) is the maximum value of p_A that will result for any given values of f and i, considering all possible values of p in the submitted product. The value of p for which this maximum value of p_A occurs is designated by p_1, hence

$$p_L = p_1\left[1 - \frac{f}{f + (1-f)(1-p)^i}\right] \qquad [16]$$

The value of p_1 for which $p_A = p_L$ is determined by differentiating Equation [15] with respect to p, equating to 0, and solving for p, that is

$$\frac{dp_A}{dp} = 1 - \frac{f^2 + f(1-f)(1-p)^i + pfi(1-f)(1-p)^{i-1}}{[f + (1-f)(1-p)^i]^2} \qquad [17]$$

Simplifying, and using the designation p_1 for the maximizing value of p, gives

$$(i+1)p_1 - 1 = \frac{1-f}{f}(1-p_1)^{i+1}$$

or

$$(1-p_1)^i = \frac{f[(i+1)p_1 - 1]}{(1-f)(1-p_1)} \qquad [18]$$

Substituting in Equation [16] this value of $(1-p_1)^i$, we have

$$p_L = \frac{(i+1)p_1 - 1}{i} \qquad [19]$$

hence

$$p_1 = \frac{1 + ip_L}{i+1} \qquad [20]$$

From Equations [18] and [19], we have

$$p_L = \frac{1-f}{fi}(1-p_1)^{i+1} \qquad [21]$$

hence

$$f = \frac{(1-p_1)^{i+1}}{ip_L + (1-p_1)^{i+1}} \qquad [22]$$

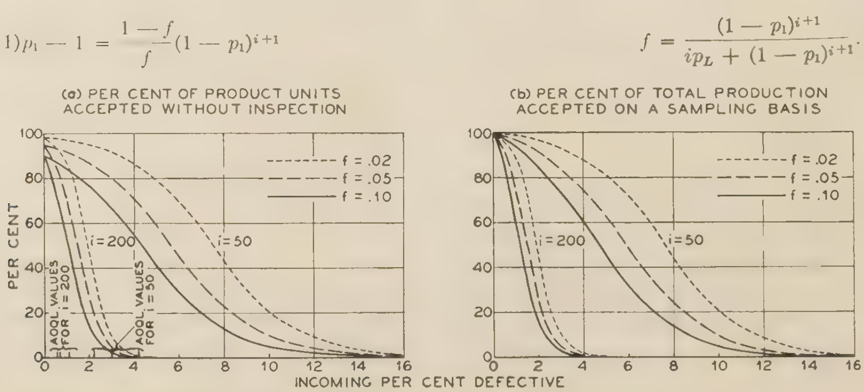

Fig. 4 Curves Showing Effect of f and i on Operating Characteristics of Plan

The curves given in Fig. 3 were calculated by choosing values of i for given values of AOQL (p_L) and calculating p_1 from Equation [20], and f from Equation [22]. Thus, for a given AOQL value, an i value may be found for a chosen f value and vice versa. It will be noted that for a given value of f, i varies inversely with the AOQL value, to a close degree of approximation.

Operating Characteristics of Plan. Figs. 4(a) and 4(b) give a picture of the operating characteristics of the general plan as f and i are varied. They indicate, for example, that for a moderate range of f values the factor i has a stronger influence than f in determining the discrimination that the method affords between high and low levels of incoming per cent defective. For the values of f and i shown, Fig. 4(b) indicates just what level of incoming per cent defective would force a correction of the manufacturing process, if the percentage of total production that would be accepted on a sampling basis falls below a critical value; often, a value of the order of 80 to 90 per cent.

Fig. 5 gives a comparison of the characteristics of several plans having the same AOQL value, 1 per cent. It indicates, for example, that when the normal level of incoming per cent defective is well below the AOQL, the AOQL value can be assured with less

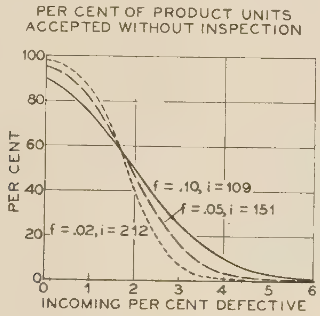

Fig. 5 Characteristics of Three Plans Having the Same AOQL of 1 Per Cent

inspection by choosing f small and i large. But since, for a given AOQL value, the average amount of inspection approaches a minimum as f approaches 0, factors other than the minimum amount of inspection have a more important influence upon the choice of the most advantageous combination of f and i values for a given set of circumstances. For example, when the inspector is located at the end of the production line, it may be desirable to use a value of i not greater than some small multiple of the number of product units on the line at any one time. Or again, the value of f is often influenced by the normal work loads of the inspector and the operators on the line. Protection against "spotty" quality, such as may arise from temporary irregularities in workmanship or materials, should receive special consideration in connection with the choice of f.

Protection Against Spotty Quality. The p_t scale, at the right in Fig. 3 provides a guide concerning the protection afforded against spotty quality in a continuous run of product. The value of p_t is the per cent defective in a run of 1000 consecutive product units, for which the probability of acceptance by sample is 0.10 for a percentage sample equal to the corresponding f value shown on the chart.

This scale indicates that the protection against spotty quality falls off very rapidly with f and that the protection, considering runs of product of 1000 consecutive units each, becomes quite poor if f is less than 2 per cent.

Effect of Selecting Group Samples. The foregoing development assumes selection of individual sample units one at a time from the flow of product and immediate examination of a unit to determine whether or not it is defective. Deviations from this procedure will in general result in giving values of AOQL higher than those shown in Fig. 3.

For example, the actual AOQL may be higher than the theoretical value (a) if the inspector delays looking at the individual units immediately when they are withdrawn from the line, or (b) if he selects a group of units at one time from the production line. The effect of either of these two deviations, both constituting a delay, may be quite large if i is small, or if large group samples are taken.

Although the modification of the theoretical AOQL value resulting from the selection of group samples has not been thoroughly explored, this should not be excessive, (a) if group samples of $n = 10$ or less are drawn from the line, and (b) if $i = 50$ or more, provided there is no delay in examining the group samples drawn from the line.

It should be noted, however, that the effect of these delay factors on the AOQL may be compensated for in part if, when a defect is found, the 100 per cent inspection includes some of the units that have already passed the inspection point.

Where appreciable delays are unavoidable, an alternative is to withhold from acceptance a stipulated number of units pending the examination of the sample units that have been selected to represent this quantity of product. Such a procedure provides in effect a lot acceptance plan, the treatment for which is covered in Part 3.

Administration of Inspection Operations. The inspection plan is most effective in practice if it is administrated in such a way as to provide an incentive to clear up causes of trouble promptly. Such an incentive may be had by imposing a penalty on the operating or manufacturing department when defects are encountered. Normally, no such penalty is imposed if both the sampling inspection and the 100 per cent inspection are performed by the same person or group of persons and the two costs merged; the inspector then merely serves as an agency for screening defects when quality goes bad. It is accordingly recommended that the sampling inspection and the 100 per cent inspection operations be treated as two separate functions.

With this in mind, the inspection work can be performed by two different inspectors, designated inspector C and inspector M. Inspector C may be considered as the *consumer's* representative in that his work is performed as a function independent of the manufacturing group. The term "consumer" is used in the general sense of the recipient of the product after the inspection has been completed. Inspector M is responsible to the manufacturing department and the cost of his work is borne by that department. His work must, however, be subject to the surveillance and approval of inspector C.

The following method of administering the inspection plan can then be used:

1 Inspector C inspects the required fraction f. So long as no defects are found, product is considered acceptable and is passed.

2 When inspector C finds a defect, he proceeds as follows:

(a) Continues inspecting the fraction f.

(b) Places some identification on the succeeding flow of product to indicate nonacceptance (or diverts it from the regular production line if the design of the line permits), such designation to apply until clearance is obtained in accordance with paragraph (3).

(c) Calls inspector M to inspect the succeeding flow of product in accordance with paragraph (3).

3 Inspector M (one or more inspectors as needed) inspects all succeeding units, except those inspected by inspector C in the fraction f, until the required number of units, i, are found clear of defects. Inspector M reports immediately to inspector C all

defects found in the course of his 100 per cent inspection and notifies him when a run of i units has been found clear of defects.

4 When notified that a run of i units has been found clear of defects, inspector C, if satisfied with the work of inspector M, releases inspector M.

5 To facilitate speedy correction of causes of trouble, inspector C, on finding a defect, should promptly notify the production foreman or other designated authority and furnish the latter with detailed information regarding the character of the defect found.

It will be noted that the foregoing procedure requires calling inspector M whenever inspector C finds a defect. To avoid taking such action on the occurrence of a single defect, the procedure can be modified so that inspector M is called into the picture only when two defects in succession are observed by inspector C. Where this feature is desired, paragraph (2) may be modified to read as follows:

2 When inspector C finds a defect, he proceeds as follows:

(a) Immediately inspects all succeeding units up to a total of i units, and if no defects are found therein, he again limits his inspection to the fraction f. If, on the other hand, during the course of inspecting the next i units, inspector C finds a second defect, he immediately discontinues his 100 per cent inspection.

(b) He then places some identification on the succeeding flow of product . . . etc.

While this procedure carries the disadvantage of placing a varying work load on inspector C, it is often preferred since a single defect tends to be regarded as an isolated occurrence, whereas two defects in quick succession (like a first and second offense) are normally accepted as sufficient evidence to justify special action.

Inspection for Several Kinds of Defects Simultaneously. The procedure just given may be applied directly to an inspection covering two or more kinds of defects, provided that the chosen AOQL value applies to all defects collectively and each unit inspected is always inspected for all of the defects under consideration.

It is sometimes desired, however, when a defect of one kind is observed, to confine the 100 per cent inspection to this one kind of defect alone. This requires a modification of the general procedure and the establishment of a separate AOQL for each kind of defect. A similar modification is required, for example, where the inspection is to cover several kinds of defects, but where the defects are grouped into two or more classes, according to their seriousness, and the defects in each class treated collectively.

The following paragraphs outline for illustrative purposes a procedure for use where the defects under consideration are to be classified into two groups, "major" and "minor," and where all major defects are to be treated collectively and all minor defects likewise. By analogy, the procedure to be followed when each kind of defect is to be treated separately will be obvious. In any event, the fraction f is made the same for all classes or all kinds of defects.

Procedure:

Several kinds of defects are grouped into two classes with respect to seriousness; designated "major" and "minor."

All defects of the same class (major or minor) are treated collectively.

Preliminary:

1 Establish an over-all AOQL value for major defects and an over-all AOQL value for minor defects. Select a suitable value for f, applicable to both major and minor defects. From Fig. 3 determine a value of i for major defects, designated i_A, and a value of i for minor defects, designated i_B.

2 At the outset, inspect 100 per cent of the units consecutively for both major and minor defects until i_{max} units in succession are found clear of defects ($i_{max} = i_A$ or i_B, whichever is the larger).

Routine:

3 When i_{max} units in succession are found clear of defects, discontinue 100 per cent inspection and inspect only a fraction f of the units *for both major and minor defects*, selecting individual sample units one at a time from the flow of product.

4 If a major (or minor) defect is observed during sampling inspection, inspect 100 per cent of the succeeding units *only for defects of the class in question* until i_A (or i_B) units in succession are found clear of defects of this class.

(a) During the 100 per cent inspection referred to in paragraph 4, inspect a portion f *for both major and minor defects.*

(b) If during the 100 per cent inspection for a particular class of defect (major or minor), a defect of the other class is observed on an individual unit of product, start 100 per cent inspection for defects of the new class *only* if the new defect is observed on one of the f units that has been inspected for both major and minor defects, and continue such 100 per cent inspection for defects of the new class until i (as determined in paragraph 1 for the new class) units in succession are found clear of defects of the new class. Do not take such action, however, if the new defect happens to be observed on one of the non-f units.

5 When the proper number of successive units are found clear of defects as in paragraphs 4 or 4(b) reinstate sampling inspection as in paragraph (3).

From the foregoing, it may be appreciated that difficulties of administration are introduced in treating a large number of classes of defects or a large number of individual defects separately. How best to group defects together for collective treatment can generally be determined from the nature of the inspection operations, whether visual or gaging, and the expectancy of defects as determined from the quality history. Items involving visual inspection can often be treated collectively to advantage.

As is generally true, the layout of an inspection plan depends to a considerable extent upon the nature of inspection operations to be performed. Simplicity of administration is always to be desired. From the standpoint of minimizing over-all inspection costs, it is often preferable, where several quality characteristics are to be inspected, to break down the inspection work into two or more separate inspection steps, each covering a relatively small number of characteristics.

3 INSPECTION OF A FLOW OF INDIVIDUAL LOTS OR SUBLOTS

Purposes of Inspection. A manufacturer's inspection of his own product serves two purposes:[9]

1 "Process control" providing a basis for action with regard to the production process, with a view to better future product.

2 "Product acceptance" providing a basis for action with regard to the product already at hand.

The plan outlined in Part 2 has both of these purposes in mind, but the provision for selecting sample units continuously from the production line places special emphasis on control. It aids, for example, in the prompt detection of defects and location of causes of trouble in the manufacturing process.

The problem of acceptance of product is often eased, though at some sacrifice to the control aspects of the inspection work, if the product is submitted to the inspector in lots or sublots and a sample taken from each.

[9] See A.S.A. War Standard, Z1.3, "Control Chart Method of Controlling Quality During Production," American Standards Association, New York, N. Y., 1942, pp. 5–6.

Inspection Procedure for Sublots. With minor modifications, the plan and procedure of Part 2 can be extended to the case where material is offered as a *flow of consecutive sublots* of articles. In the inspection of parts, for example, the material may be offered in panloads or trays, each containing a collection of parts produced under essentially the same conditions. Or again, the product from a common source for a given short period of time, such as a $^1/_2$ hr, 1 hr, etc., may often be treated as a sublot and offered to the inspector as such for his acceptance. In what follows, however, it is essential that such sublots be kept in the *order of their production.*

The theoretical development given in Part 2 makes use of random-order spacing of defects in a statistically controlled product, with the specific provision that the units inspected be selected in the order of their production. In applying the general plan to the inspection of a *flow of consecutive sublots,* we no longer have individual units available in the order of their production. But we can use the same theoretical framework if we consider the random spacing of defects as their spacing in the chain of inspected units arranged in the *order of their inspection.* The probability distribution of the spacing of defects in inspected units will be the same regardless of the manner of selecting the units to be inspected, so long as we hold to the concept of statistical control in our solution.

The "i units in succession to be found clear of defects," discussed in Part 2, will now be defined as "i consecutively inspected units." During sampling inspection, a group sample of units will be selected from each sublot, and the fraction f will relate to the ratio of the number of units in the sample to the total number of units in the sublot. The fraction f will be held constant for all sublots. Furthermore, when it is required under the general plan to find i inspected units in succession clear of defects, the 100 per cent inspection must be allowed to extend into immediately succeeding sublots if i units in succession are not found clear in the current sublot.

Procedure B. The procedure is as follows:

1 At the outset, start inspecting 100 per cent of the units in a sublot and continue such inspection until i inspected units in succession are found clear of defects. Extend the 100 per cent inspection, if necessary, into one or more succeeding sublots in the order of their production.

2 When i inspected units in succession are found clear of defects, discontinue 100 per cent inspection and inspect only a fraction f of the units from each of the sublots, selecting the sample units in such a way as to fairly represent the sublot.

3 If a sample unit is found defective, start a 100 per cent inspection of the remainder of the sublot, and continue the 100 per cent inspection until again i inspected units in succession are found clear of defects, as in paragraph (1), extending such inspection into succeeding sublots, if necessary.

4 In the event the 100 per cent inspection extends into one or more succeeding sublots, if the number of units inspected in the last of such succeeding sublots exceeds a fraction f of the number of units in the sublot, accept this last sublot without further inspection. If on the other hand, the number of units inspected in this last sublot is less than the fraction f, inspect additional units from this same sublot to make up a sample equal to a fraction f of the number of units in the sublot.

5 Correct or replace with good units all defective units found.

As was the case in Part 2, the inspection plan is defined by two constants, f and i, and the protection offered is expressed in terms of AOQL. This sublot inspection plan differs from those already published in that the screening action is not confined to a single sublot but may extend over a succession of sublots, the entire production being regarded as a train of sublots that are linked together for purposes of inspection in the order of their production.

4 REMARKS

It will have been noted that the plan here outlined should be regarded as a "special-purpose" plan applicable under the conditions which have been enumerated: where production is practically continuous, where inspection is to be made during production or immediately thereafter and is to serve not only as a screening *acceptance* agency if necessary, but as an aid to *process control* by disclosing promptly any substandard quality conditions in the product. It is believed that the general plan provides a structure, which with possible variations in procedure to serve particular circumstances, may be found useful in designing additional sampling inspection techniques.

Physical Properties of a Structural Plastic Material

By CECIL W. ARMSTRONG[1]

THE aviation industry is vitally interested in high-strength low-density materials which show promise of utilization in airplane structures. This interest exists because of the following reasons:

1 The need for the further reduction in weight of all types of airplanes in order to obtain better performance characteristics.

2 The need for increasing the rigidity of all thin-gage sections to reduce local failures resulting from vibration and to maintain contours under all flight conditions.

3 The need for reducing the number of man-hours required for the tooling, fabrication, and final assembly of airframes.

4 The need for improvement in the over-all smoothness of all aerodynamic surfaces.

It is generally agreed among aeronautical engineers that the development of a low-density structural plastic material, capable of being molded in large assemblies with inexpensive molds, would help to satisfy the needs listed. The word "structural" necessarily implies that the plastic material will possess certain minimum mechanical properties. For structural use in airplanes, the strength-weight ratios and other mechanical properties must compare favorably with corresponding values of the commonly used aluminum alloys.

Many new resins have recently been developed and without doubt countless others will emerge from the numerous chemical laboratories. These resins are being copolymerized and interpolymerized. They are being cast, molded, and laminated with a large variety of organic and inorganic fibrous fillers, woven fabrics, macerated fabrics, and the like. It has become exceedingly difficult for the engineer to choose from among these thousands of combinations any one material for a specific application. The need exists for reliable engineering data on plastic materials so that an intelligent selection of a material for a specific application can be accomplished. Considerable progress is being made in this direction by the joint efforts of the Army, Navy, and the plastics industry, but much remains to be done.

Among the new resins which have become commercially available (military requirements first, of course), there are several thermohardening resins which are unique in that no external pressure is required during the process of molding a plastic part. These new resins may be used in combination with high-strength filler materials to produce low-density high-strength laminated plastics which soon may fulfill some of the needs which have been enumerated.

Methods of Laminating "No-Pressure" Resins

To produce a laminated-plastic article from one of these "no-pressure" resins, one or more layers of the filler material may be wrapped around or draped over a mold. The filler material may be saturated with the liquid resin either before or after placing in contact with the mold. The laminate then is cured (hardened) by application of heat. Curing temperatures seldom exceed 240 F. The curing period may require from thirty minutes to eight hours, depending upon the requirements of the particular resin, the thickness of the laminate, and the heat conductivity of the mold. If a smooth glossy surface is desired, nonporous cover sheets, such as Cellophane, may be used.

Parts having intricate shapes and double contours may necessitate the use of some pressure (0.1 to 10 psi) to keep the impregnated-fabric filler material in intimate contact with all surfaces of the mold and to prevent free resin from accumulating in localized spots. For application of these pressures, the use of thin transparent films of inexpensive thermoplastic materials has been found to be more suitable than the use of expensive synthetic-rubber bags. Many novel and ingenious techniques already have been developed for the fabrication of complete assemblies, which if made of metal would require the forming and assembly of several contoured parts. Because of the fact that pressures are low, molds may be simple, light in weight, and inexpensive. The primary requirement of molds for fabrication of parts from these new resins is that they be made to correct dimensions and contours.

Physical Test Program

A preliminary study of the limited amount of available data, supplemented by various laboratory tests, indicated that the MR-1A resin,[2] when laminated with certain of the Fiberglas fabrics, possessed better mechanical properties than any of the commercially available "no-pressure" resins. Accordingly, a modest test program was undertaken by the Lockheed Structures Research Laboratory. Results which are reported in this paper are typical of data thus far obtained.

Materials Tested and Methods of Testing

Data presented herewith were obtained from MR-1A resin laminated with Fiberglas-fabric fillers. A brief description of the Fiberglas-filler materials which were used in the preparation of flat sheet laminates, from which test specimens were machined, is given in Table 1.

TABLE 1 FIBERGLAS CONTINUOUS-FILAMENT CLOTHS USED FOR MR-1A LAMINATED TEST SPECIMENS[a]

CLOTH DESIGNATION	NOMINAL THICKNESS IN	WT-OUNCES PER SQ YD	WARP		FILLING	
			ENDS PER INCH	APPROX STRENGTH LBS/IN WIDTH	PICKS PER INCH	APPROX STRENGTH LBS/IN. WIDTH
OC-63	.0115	9.12	46	875	12 (Cotton)	3
ECC-11-127	.007	6.36	42	412	32	334
ECC-11-148	012	10 95	30	625	19	459
ECC-11-161	.015	14 11	28	839	16	562

[a] Data supplied by Owens-Corning Fiberglas Corporation.

Tension- and bend-test specimens were prepared and tested in accordance with the General Federal Specification[3] for Organic

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Contributed jointly by the Rubber and Plastic Groups and the Aviation Division and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

[2] Manufactured by Marco Chemicals, Inc., Sewaren, N. J., laminated and distributed by Swedlow Aeroplastics Corporation, Glendale, Calif.

[3] Section IV (Part 5) of Federal Standard Stock Catalog, Superintendent of Documents, Washington, D. C.

TABLE 2 PHYSICAL TEST RESULTS MR-1A FIBERGLAS LAMINATES

LAMINATE DESIGNATION	223	224	309	236	221	241
FILLER, FIBERGLAS NO.	OC-63	OC-63	OC-63	ECC-11-127	ECC-11-148	ECC-11-161
NUMBER OF LAMINAE	12	12	8	18	10	8
STACK ARRANGEMENT	PARALLEL	CROSSED 90°	CROSSED 90°	CROSSED 90°	PARALLEL	CROSSED 90°
NOMINAL THICKNESS, INCHES	.104	.107	.107	.128	.124	.121
SPECIFIC GRAVITY	1.96	1.93	1.69	1.81	1.80	1.79
TENSION						
DIRECTION OF LOAD	WITH WARP	—	—	—	WITH FILL	—
ULTIMATE LOAD, LBS.	6,390	3,140	2,095	2,840	2,070	2,210
SPECIMEN WIDTH, IN.	.604	.502	.502	.502	.482	.505
ULT LOAD PER INCH OF WIDTH PER LAMINA	880	520	520	315	430	550
TANGENT PROPORTIONAL LIMIT P.S.I.	50,000	32,000	21,000	26,000	13,000	13,500
.01% OFFSET PROPORTIONAL LIMIT	58,000	41,000	25,000	27,500	15,000	15,500
.2% OFFSET YIELD STRESS	—	58,000	—	38,700	30,800	24,300
ULTIMATE TENSILE STRESS	105,000	58,500	39,000	44,200	34,600	36,200
YOUNG'S MODULUS OF ELASTICITY P.S.I.	5,960,000	2,800,000	1,900,000	2,100,000	1,500,000	1,900,000
ELONGATION AT FAILURE (% OVER 2 INCH LENGTH)	1.8	2.3	2.1	2.6	2.6	2.9
COMPRESSION (EDGEWISE)						
DIRECTION OF LOAD	WITH WARP	—	—	—	WITH FILL	—
ULTIMATE LOAD, LBS.	1,340	1,380	1,580	1,393	1,000	1,200
SPECIMEN WIDTH, IN.	.496	.500	.495	.500	.500	.500
ULT LOAD PER INCH OF WIDTH PER LAMINA	225	230	400	155	200	300
TANGENT PROPORTIONAL LIMIT P.S.I.	26,100	25,950	21,500	14,000	11,000	10,000
.01% OFFSET PROPORTIONAL LIMIT	—	—	23,700	15,000	14,000	12,000
ULTIMATE COMPRESSIVE STRESS	26,100	26,000	29,000	22,500	16,900	19,600
YOUNG'S MODULUS OF ELASTICITY P.S.I.	5,500,000	3,000,000	2,260,000	2,800,000	2,200,000	2,300,000
DEFORMATION (% OVER 2 INCH LENGTH)	.47	.86	1.38	.86	.75	.93
MANNER OF FAILURE	DIAGONAL SHEAR	DIAGONAL SHEAR	DIAGONAL SHEAR	DIAGONAL SHEAR	DIAGONAL SHEAR	DIAGONAL SHEAR
BENDING (FLATWISE)						
LENGTH, WIDTH, THICKNESS, INCHES	5×1.02×.102	5×.745×.106	5×.749×.109	5×.751×.128	5×.751×.126	5×.746×.116
SPAN, INCHES	3	3	3	3	3	3
ULTIMATE LOAD AT CENTER OF SPAN, LBS.	138	90	128	105	94	78
TANGENT PROPORTIONAL LIMIT P.S.I.	55,000	41,000	45,000	17,000	19,000	19,000
MODULUS OF RUPTURE, P.S.I.	59,300	48,200	64,800	38,400	35,500	34,900
MODULUS OF ELASTICITY, P.S.I.	5,950,000	3,000,000	2,100,000	2,300,000	1,630,000	2,080,000
BEARING						
HOLE DIAMETER	—	.124	.124	.124	.124	.124
NOMINAL BEARING STRENGTH (4% HOLE DEFORMATION)	—	28,000	23,500	20,000	29,500	23,300
ULTIMATE BEARING STRENGTH, P.S.I.	—	32,100	31,300	38,500	34,800	34,600

TABLE 3 SPECIFIC STRENGTH VALUES MR-1A FIBERGLAS LAMINATES

MATERIAL	(1) SPECIFIC TENSILE STRENGTH P.S.I.	(2) SPECIFIC TENSILE MODULUS P.S.I.	(3) SPECIFIC BUCKLING STABILITY P.S.I.	(4) SPECIFIC MODULUS OF RUPTURE P.S.I.	(5) SPECIFIC ULT. BEARING STRENGTH P.S.I.
MR-1A FIBERGLAS-223	53,500	3,040,000	790,000	15,400	—
MR-1A FIBERGLAS-224	30,300	1,450,000	416,000	12,900	16,600
MR-1A FIBERGLAS-309	23,000	1,120,000	435,000	22,600	18,500
MR-1A FIBERGLAS-236	24,400	1,160,000	388,000	11,700	21,300
MR-1A FIBERGLAS-221	19,200	833,000	280,000	10,900	19,300
MR-1A FIBERGLAS-241	20,200	1,060,000	362,000	10,900	19,300
24S-T ALCLAD ALUMINUM ALLOY	20,200	3,800,000	495,000	7,300	29,600
24S-RT ALCLAD ALUMINUM ALLOY	22,400	3,800,000	495,000	8,100	32,400

(1) Specific Tensile Strength = Ult. tensile strength divided by specific gravity.
(2) Specific Tensile Modulus = Tension modulus divided by specific gravity.
(3) Specific Buckling Stability = Flexural modulus divided by (specific gravity)^3
(4) Specific Modulus of Rupture = Modulus of rupture divided by (specific gravity)^2
(5) Specific Ult. Bearing Strength = Ult. bearing strength divided by specific gravity.

Plastics, L-P-406, dated December 9, 1942. Bearing-stress-deformation data, Fig. 4, were obtained by use of a fixture similar to one developed at the University of Kansas.[4] Ultimate bearing-stress values were obtained from test specimens 2.50 in. wide, with 0.124-in-diam holes centrally located at an edge distance of 0.375 in. The small compression-test specimens were supported and tested in a jig, Fig. 5, designed by L. F. Bonza[5] and made in the Lockheed Structures Research Laboratory. Similar compression jigs have been widely used for some time.

Test Results

Typical test results are given in Table 2 and in Figs. 1 to 4.

[4] "Bearing Strength of Plastics and Plywood," by James Bond, Trans. A.S.M.E., vol. 65, 1943, pp. 9–14.

[5] Jun. Research Engineer, Lockheed Aircraft Corp., Burbank, Calif.

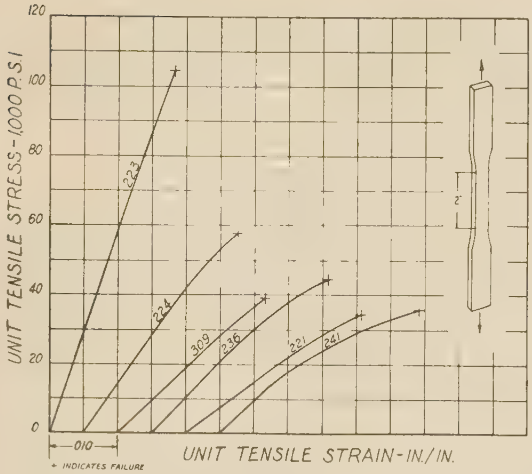

Fig. 1 Tensile Stress-Strain Curves of MR-1A Fiberglas Laminates

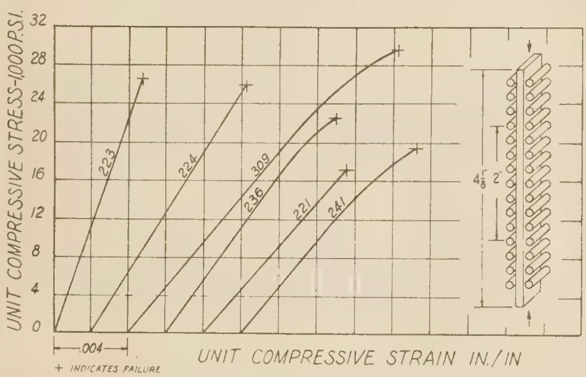

Fig. 2 Compressive Stress-Strain Curves of MR-1A Fiberglas Laminates

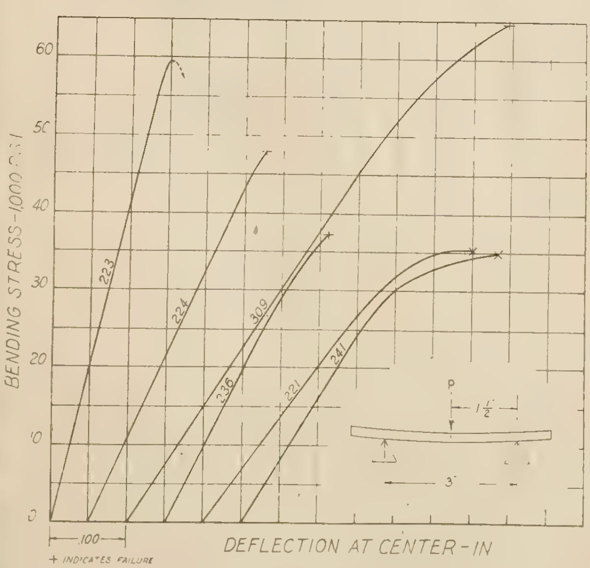

Fig. 3 Flexural Stress-Strain Curves of MR-1A Fiberglas Laminates

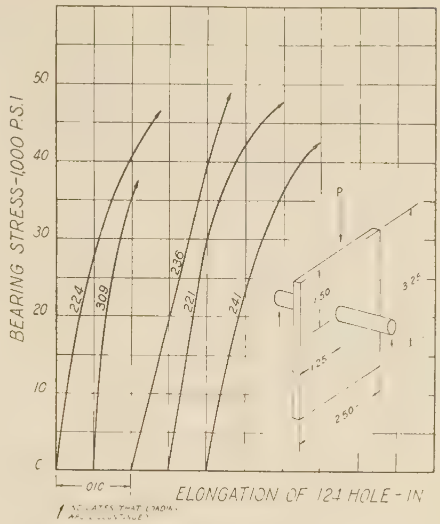

Fig. 4 Bearing-Stress-Deformation Curves for MR-1A Fiberglas Laminates

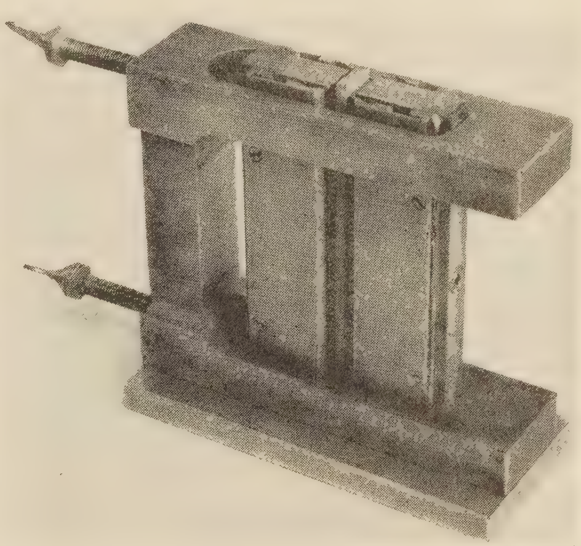

Fig. 5 Jig for Testing Specimens in Compression

Certain of these values, adjusted to an equivalent-weight base, are compared with corresponding values of two aluminum alloys in Table 3. The specific-strength properties of the aluminum alloys were computed from previously published values.[6]

Attention is directed to the fact that these laminates are anisotropic, resulting from parallel stacking or cross-stacking (alternate laminae oriented at right angles) of unidirectional materials, such as OC-63 Fiberglas or bidirectional materials, such as are listed in Table 1. Results which are shown in Table 2 were obtained from specimens cut parallel to one of the two principal directions of the laminates.

Preliminary tensile tests made from specimens cut at 45 deg with the principal directions of the cross-stacked MR-1A Fiberglas laminates exhibited ultimate tensile-strength and tensile-modulus values thirty to forty per cent lower than corresponding "with-grain" values.

[6] "Strength of Aircraft Elements," U. S. Army-Navy-Civil Committee on Aircraft Requirements, Publication ANC-5, Dec., 1942.

Tensile tests made at room temperature from unfilled cast MR-1A resin (specific gravity = 1.2) indicated an average ultimate tensile strength of 6200 psi and an average tensile modulus of 370,000 psi.

The following data on other properties of the MR-1A Fiberglas laminates were supplied by the Swedlow Aeroplastics Corporation. Methods of testing were stated to have been in accordance with A.S.T.M. specifications:

Impact strength, notched Izod, ft-lb per in. of notch...	25 – 100
Ultimate shear strength, psi........................	14000–33000
Ultimate compressive strength, flatwise, psi.........	41000–75000
Moisture absorption, per cent.......................	0.2 – 0.4
Gasoline absorption, per cent.......................	0.01 – 0.15
Burning rate, in. per min...........................	0 – 0.4
Hardness, Rockwell M............................	71 – 107
Hardness, Barcol impressor.........................	40 – 65
Heat distortion, deg F.............................	300 – 350
Coefficient of linear expansion, 10^{-5} per deg F.......	2.7 – 5.2

Fatigue- and creep-test data and other physical properties which are not listed have not as yet been determined.

Discussion of Results Obtained

It is interesting to compare test results on a basis of pounds load per inch of width per lamina with corresponding values of the Fiberglas cloth. Note that for the unidirectional laminate (see Table 2, laminate No. 223) the ultimate tensile load per inch of width per lamina is 880 lb; also from Table 1, that the nominal breaking strength of the OC-63 Fiberglas is 875 lb. For cross-stacked laminates of the same material (No. 224 and No. 309) corresponding values of 520 lb were obtained; however, since only one half of the laminae were oriented parallel to the direction of the load, these values could be expressed as 1040 lb per in. of width for each lamina oriented in the direction of the applied load. Similar comparisons may be made with laminates made from the woven fabrics.

Attention is directed to the fact that both the thickness and density of the laminate are dependent upon the relative proportion of resin and filler materials. In tension, most of the load is carried by the filler material. Two laminates, each containing the same number of laminae but varying in resin content might carry equal tensile loads, but because of the greater thickness would have different ultimate unit-stress and modulus values.

Examination of the data will show that the resin content (low resin content results in high specific-gravity values) for optimum tensile properties will not result in optimum bend and compression properties. This fact may be demonstrated by comparing data obtained on laminate No. 224 with corresponding data obtained on laminate No. 309.

Costs of the Processed Materials

The present high prices of Fiberglas cloths impose serious limitations on the use of these materials for simple structural applications. Many high-strength organic and inorganic fibrous materials which recently have been developed, or ones known to be in the developmental stage, may offset this limitation.

For contoured structural shapes which generally begin with flat metal sheets and which require the forming of several parts with expensive dies by drop hammers or hydropress, followed by manual assembly of these parts, the use of a suitable plastic material can save cost and valuable man-hours of production time. Comparative totals of man-hours required for the tooling and fabrication of actual contoured metal parts and corresponding man-hours required for the tooling of the same "no-pressure" plastic parts should be very enlightening.

Conclusions

The MR-1A Fiberglas laminates possess physical properties which enable this material to be used to good advantage in many airplane structural applications. The ability to mold large contoured assemblies with substantially no pressure will, without doubt, result in substantial savings in cost and in man-hours of time required for the tooling and fabrication of many assemblies now made of metal.

Known disadvantages include the anisotropic properties of fabric laminates, the length of time required to complete the curing of each part, the low values of the modulus of elasticity, the low bearing strength, and the present incompleteness of important physical data.

Acknowledgment

The author is indebted to Mr. L. F. Bonza for his assistance in the conduct of certain of the physical tests and in the compilation of test data which are included in this paper.

Isothermal Pressure Drop for Two-Phase Two-Component Flow in a Horizontal Pipe

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The static pressure drop accompanying the isothermal two-phase two-component flow of air and eight various liquids, including benzene, water, and S.A.E. 40 oil, was measured for flow conditions varying from all air to all liquid in a 1-in. glass pipe and a 1/2-in. galvanized-iron pipe. The flow patterns existing at the various flow rates were studied visually and photographically. A macroscopic analysis of the flow phenomenon is presented, which allows the prediction of two-phase two-component static pressure drop under certain flow conditions. The application of the proposed method to the prediction of pressure drop along a pipe in which a fluid is evaporating is tentatively outlined.

Nomenclature

THE following nomenclature is used in the paper:

A_c = cross-sectional area of pipe of diameter D_p, sq ft
A_l = cross-sectional area of liquid flow, sq ft
A_g = cross-sectional area of gas flow, sq ft
D_H = hydraulic diameter, ft
D_g = hydraulic diameter for gas flow, ft
D_l = hydraulic diameter for liquid flow, ft
D_p = inside diameter of pipe, ft
f_c = unit thermal convective conductance, Btu/hr sq ft deg F
F_1, F_2, F_3, F_4 = functions utilized in Appendix 1
g = gravitational force per unit mass

$$= 32.2 \frac{\text{lb}}{\left(\dfrac{\text{lb sec}^2}{\text{ft}}\right)}$$

L = length of pipe, ft
m = exponent of Reynolds modulus in the Blasius expression for friction factor for the gas phase
n = exponent of Reynolds modulus in the Blasius expression for friction factor for the liquid phase
p = average static pressure in pipe, psi
P = average static pressure in pipe, psf
r = latent heat of vaporization of liquid, Btu per lb
t = average fluid temperature, deg F
T = average fluid temperature, deg R
v_g = velocity of gas, fps
v_l = velocity of liquid, fps
W_g = weight rate of gas, lb per sec
W_l = weight rate of liquid, lb per sec
W_T = total weight rate (vapor + liquid), lb per sec
Z = co-ordinate measured along pipe axis, ft
γ_g = weight density of gas, lb per cu ft
γ_l = weight density of liquid, lb per cu ft
ΔL = finite difference of length, ft
ΔP = finite difference of static pressure in length ΔL psf
$\left(\frac{\Delta P}{\Delta L}\right)_g$ = pressure drop per unit length due to gas flowing at a rate W_g, with density γ_g in a pipe of diameter D_p (lb per sq ft) per ft
$\left(\frac{\Delta P}{\Delta L}\right)_{T-P}$ = pressure drop per unit length for two-phase flow, (lb per sq ft) per ft
Δt = difference in temperature between tube wall and fluid being evaporated, deg F
μ_g = absolute viscosity of gas, lb sec per sq ft
μ_l = absolute viscosity of liquid, lb sec per sq ft

Dimensionless Moduli

α = flow-type modulus for liquid $= \dfrac{A_l}{\frac{\pi}{4} D_l^2}$

β = flow-type modulus for gas $= \dfrac{A_g}{\frac{\pi}{4} D_g^2}$

$\chi_{tt} = \left(\dfrac{\mu_l}{\mu_g}\right)^{0.111}\left(\dfrac{\gamma_g}{\gamma_l}\right)^{0.555}\left(\dfrac{W_l}{W_g}\right)$ = two-phase flow modulus for mechanism in which flow of both liquid and gas is turbulent

$\chi_{vt} = \dfrac{C_l}{C_g}(Re_g)^{-0.8}\dfrac{\gamma_g}{\gamma_l}\cdot\dfrac{\mu_l}{\mu_g}\cdot\dfrac{W_l}{W_g}$ = two-phase flow modulus for mechanism in which flow of liquid is viscous and that of gas is turbulent

$\chi_{vv} = \dfrac{\mu_l}{\mu_g}\cdot\dfrac{\gamma_g}{\gamma_l}\cdot\dfrac{W_l}{W_g}$ = two-phase flow modulus for mechanism in which flow of both phases is viscous

ζ_g = friction factor for gas (see Equation [2])
ζ_l = friction factor for the liquid (see Equation [1])
Φ_{tt} = function of χ_{tt} utilized in calculation of two-phase pressure drop
Φ_{vt} = function of χ_{vt} utilized in calculation of two-phase pressure drop
C_g = constant in empirical equation for friction factor for gas. Note that for turbulent flow C_g = 0.184 for smooth pipes
C_l = constant in empirical equations for friction factor for liquid. Note that C_l = 64 for viscous flow
Re = Reynolds modulus $= \dfrac{4}{\pi}\cdot\dfrac{W}{D\mu g} = \dfrac{vD\gamma}{\mu g}$
$Re_g = \dfrac{4}{\pi}\cdot\dfrac{W_g}{D_p\mu_g g}$ = Reynolds modulus for gas phase

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TABLE 1 RANGE OF VARIABLES

	Fluids: Liquid	Fluids: Gas	Liquid viscosity,[a] lb-sec/ft²	Liquid, specific gravity[a]	Liquid surface tension,[a] lb per ft	Average pressure p, psia	Average temperature, t, deg F	Liquid rate: Minimum lb per sec	Liquid rate: Maximum lb per sec	Air rate: Minimum lb per sec	Air rate: Maximum lb per sec
Taylor tests, glass pipe 1.017 in. ID	Water	Air	2.34×10^{-5}	1.00	4.89×10^{-3}	18	60	0.015	4.68	0.00074	0.063
	Water + Nekal	Air	1.81×10^{-5}	1.00	3.12×10^{-3}	18	80	0.050	4.66	0.00063	0.069
	Kerosene	Air	4.35×10^{-5}	0.830	2.13×10^{-3}	18	78	0.029	3.70	0.00060	0.068
	Diesel fuel	Air	10.35×10^{-5}	0.867	2.03×10^{-3}	18	80	0.030	3.83	0.00088	0.071
	Blend no. 1 of Diesel fuel + oil B	Air	87.5×10^{-5}	0.901	2.16×10^{-3}	18	83	0.050	2.90	0.00087	0.068
	Blend No. 2 of Diesel fuel + oil B	Air	257×10^{-5}	0.913	2.23×10^{-3}	18	85	0.040	1.64	0.00091	0.060
	Oil B (S.A.E. 40)	Air	558×10^{-5}	0.912	2.31×10^{-3}	18	85	0.050	0.810	0.00089	0.054
Thomsen tests, 1/2-in. standard galvanized pipe	Benzene	Air	1.29×10^{-5}	0.876	1.93×10^{-3}	52	75	0.0141	0.0758	0.0052	0.0361
	Water	Air	2.05×10^{-5}	1.00	4.96×10^{-3}	52	70	0.0194	0.0832	0.0109	0.0304
	Water + Kemenol	Air	2.05×10^{-5}	1.00	3.30×10^{-3}	52	70	0.0226	0.0874	0.0081	0.0320
	Kerosene	Air	3.38×10^{-5}	0.910	2.08×10^{-3}	52	70	0.0172	0.0773	0.0051	0.0273

[a] Properties at temperature of test t, deg F.

based on W_g and pipe diameter D_p

$$Re_l = \frac{4}{\pi} \cdot \frac{W_l}{D_p \mu_l g} = \text{Reynolds modulus for liquid phase}$$

based on W_l and pipe diameter, D_p

Subscripts

c = cross-sectional, convective
g = gas
l = liquid
H = hydraulic
p = pipe
$T\text{-}P$ = two phase
tt = turbulent liquid, turbulent gas
vt = viscous liquid, turbulent gas
vv = viscous liquid, viscous gas

Introduction

Many flow systems in engineering design involve the simultaneous transport in pipes of a gaseous (or vapor) phase and a liquid phase. Examples of such systems are numerous; among them are water-steam mixtures in boiler tubes, petroleum liquids and their vapors in tube stills, liquid-vapor mixtures in refrigeration systems, oil-laden compressed air, condensate-return lines, gasoline-engine manifolds, and partial condensers in fractionation equipment. Air flow in cooling towers and steam flow in the low-pressure stages of turbines are more complex examples of this type of hydrodynamic problem. A considerable amount of experimental work in the field of two-phase flow has been reported in the literature but no satisfactory analyses of the data have been accomplished. This paper presents further data on the subject of static pressure drop in two-phase flow and outlines a macroscopic analysis of the test data, which allows prediction of the pressure drop under certain flow conditions.

Previous experimental work had indicated that the variables affecting the pressure drop per unit length for two-phase flow were the pipe diameter, the rate of liquid flow, the rate of gas (or vapor) flow, the liquid viscosity, and perhaps the surface tension of the liquid. A preliminary analytical analysis indicated that the liquid density and the gas density and viscosity were also pertinent variables.

To note the effect of these variables, a series of tests was performed in which the pressure drop along a 1-in. glass tube (1)[6] and a 1/2-in. galvanized-iron pipe (2), resulting from the two-phase two-component flow of air and various liquids, was measured. The range of variables covered by the tests is shown in Table 1. The data on the 1-in. glass tube include the greater range of variables, and the major portion of the discussion will be concerned with these results. The tests on the 1/2-in. pipe were found to substantiate the data obtained on the glass pipe.

The test arrangement for the 1-in. glass tube is shown in Fig. 1. The liquid and the air were mixed several feet before the test

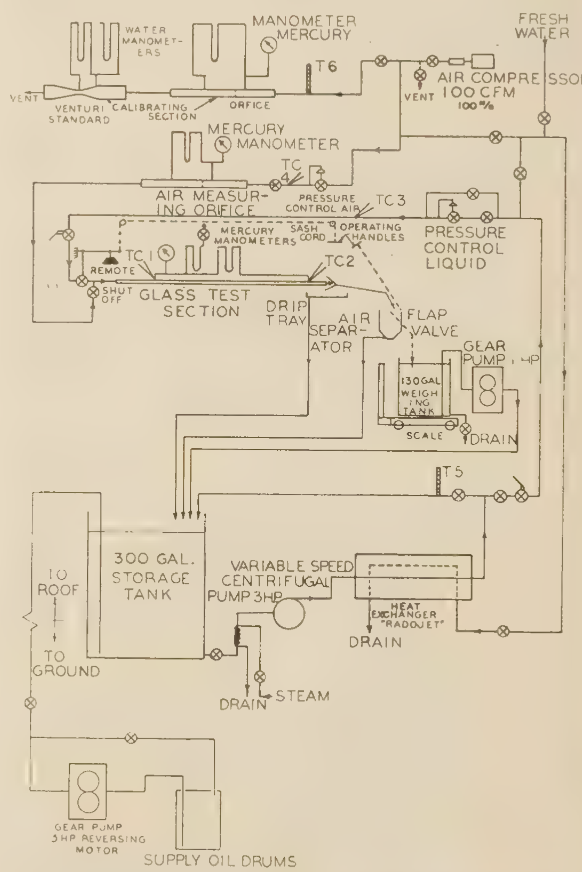

Fig. 1 Schematic Diagram of Experimental Equipment

[6] Numbers in parentheses apply to references at the end of the paper. A comprehensive Bibliography is also included.

section proper. The method used for mixing the two phases was chosen as a result of considerable experimentation. Trials were made with both air injected into the liquid and liquid injected into the air. These different modes of injection were tried to minimize "slugging," which is the name given a stream of alternate plugs of liquid and air. Such plugs were from a few inches to several feet long, depending upon the rates of flow. The arrangement necessary for minimum "slugging" required the introduction of the air into the upper surface of the liquid and perpendicular to it. Furthermore, any bends, valves, or other obstructions between the injection point and the measuring section must be avoided.

After being mixed, the fluids passed through a glass test section in which the flow pattern could be observed and the static pressure drop measured across the taps[7] shown in Fig. 1. The mixture of liquid and air exhausted into a separator, the liquid returning to the reservoir and the air exhausting to the atmosphere. Air rates were measured by means of calibrated orifices, and liquid rates were determined by gravimetric means. The average temperature and static pressure in the pipe were also recorded. The static pressure drops due to the flow of the two pure components, i.e., all air or all liquid, were carefully measured at intervals throughout the two-phase test runs.

The equipment utilized for the tests on the smaller pipe was practically identical to that just described.[7]

Data and Results

The experimental results obtained on the 1-in. glass tube are shown in graphical form in Figs. 2 to 8,[8] inclusive. Inspection of the data indicates the effect of the variables mentioned previously. Figs. 2 to 7, inclusive, reveal the pressure drop as a function of the gas and liquid rates for air and water, kerosene, Diesel fuel oil, two blends of Diesel fuel oil and S.A.E. 40 oil, and S.A.E. 40 oil. The data for water with a surface-tension-reducing additive are shown in Fig. 8 and indicate that the change of surface tension had no apparent effect on the pressure drop, although it influenced the flow pattern appreciably, as will be discussed later. The decrease in the surface tension of the water due to the additive was, however, only 36 per cent and it is possible that a greater change would influence the results.

The following general trends are evident in Figs. 2 to 7, inclusive:

1 The static pressure drop for two-phase flow is always greater than the pressure drop for each phase flowing alone at the weight rates which obtain in the two-phase flow.

2 For a given gas rate, adding liquid increases the pressure drop, the pressure drop becoming greater as more liquid is added.

3 When the air flow approaches zero for any constant liquid rate, the static pressure drop due to the flow of the pure liquid is approached. Thus the lines of constant liquid rate become horizontal as the gas rate is decreased.

4 As the gas flow is increased, for any constant liquid rate, the static pressure drop increases and approaches the 100 per cent gas line as an asymptote.

5 The flow for both the air and the liquid may be turbulent. Figs. 2, 3, 4, and 8 indicate this type of flow mechanism, for both the 100 per cent liquid and 100 per cent air data reveal a variation of pressure drop with the 1.8 power of the rate of flow.

6 The flow may be turbulent for the air and viscous for the liquid. Curves 5, 6, and 7 reveal this type of flow, for the pressure drop due to the liquid alone is seen to vary directly with the weight rate, while the pressure drop for the air alone varies with the 1.8 power of the air rate. The case in which both the air and the liquid flow was viscous was not observed in these tests.

7 The greater the viscosity of the liquid, the greater is its effect on the pressure drops for two-phase flow. This effect is particularly evident when the liquid is flowing viscously, Figs. 5, 6, and 7.

Simultaneously with the static pressure drop measurements, visual and photographic observations of the flow mechanism were made. These are of considerable interest because they reveal the complexity of the microscopic phenomena occurring during two-phase flow. As would be expected, two distinct types of flow phenomena were observed, one when the flow of both fluids was turbulent, the other when the flow of the liquid was viscous and the flow of the gas was turbulent.

The results of the observations are given in Table 2 and the flow types are illustrated in Fig. 9. In this figure, flow type 1 indicates air alone, and flow type 9, liquid alone. Type 2 is a frothy mixture of liquid and air and did not exist when the liquid flow was viscous. In type 3 the fluid moved along the bottom of the pipe, with the air above, waves of definite wavelength moving at the gas-liquid interface.[9] In the case of viscous flow of the liquid phase, the wavelengths were somewhat greater than in the case of turbulent flow of the liquid. Type 4 followed type 2 at medium air rates as the liquid rate was reduced, and consisted of definite liquid waves on both the bottom and top of the pipe traveling at different velocities. As the liquid rate was decreased further, the upper waves disappeared and type 6 appeared. In this interesting flow type, the crests of the waves point, as shown, in the direction opposite to the direction of fluid flow. When the air rate was very low and the liquid rate was reduced, type 2 flow was followed by type 7 in which small air bubbles collected in the upper half of the pipe. As the liquid rate was further reduced (at low air rates) type 5 appeared in which long bubbles, occupying almost one half the pipe cross-sectional area, moved along the pipe. At very low liquid rates and high air rates, "annular" flow occurred (type 8). The liquid flowed in the form of an annulus, even though the pipe was horizontal, and the liquid surface was covered with small capillary waves. When the surface tension of the water was lowered by adding Nekal BX, much foaming was observed, type 2 being prominent in the results.

It is then evident that the microscopic behavior of the two-phase flow system is extremely complex. Since, however, the static pressure drops for the water and the water-Nekal blend were almost identical, although the detailed flow patterns were quite different, it appears that it should be possible to analyze the problem from a macroscopic point of view and obtain results which are of utility from an engineering standpoint, even though the details of the flow mechanism are not clarified. This method of attack is of course the one employed whenever the "friction factor" is utilized, i.e., the "friction factor" allows prediction of static pressure drops in turbulent flow without the need for consideration of the complex details of turbulence.

Analysis of Results

The analysis of the static pressure drop accompanying two-phase flow results from the application of two basic postulates. The first postulate is that the static pressure drop for the liquid

[7] The static pressure taps for the 1-in. glass pipe were 21.6 ft apart and those for the $^1/_2$-in. galvanized-iron pipe 50 ft apart. The pressure in many cases fluctuated severely due to the unavoidable "slugging" of the fluid. During these runs the differential manometer deflection was recorded at the estimated mid-point of the oscillation. The static pressure manometer was "throttled" to reduce oscillations.

[8] The pressure drops shown in Figs. 2 to 8, inclusive, have been corrected to atmospheric pressure conditions by the approximate method outlined in Appendix 2. Points for zero gas flow (100 per cent liquid) are plotted on the 0.0001 lb/sec gas rate abscissa for convenience.

[9] Such waves form at the interface of two fluids moving with different velocities (4).

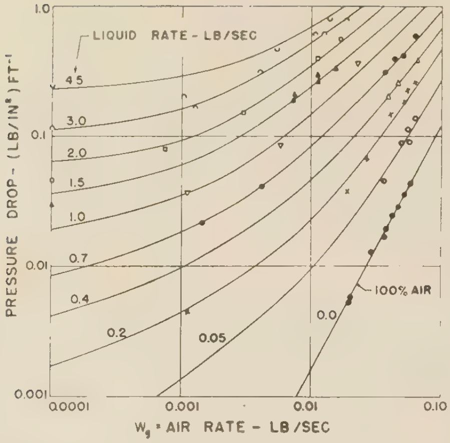

Fig. 2 Pressure Drop for Flow of Air and Water in 1-In. Glass Pipe
(Viscosity of liquid = 2.34×10^{-5}, lb sec/ft².)

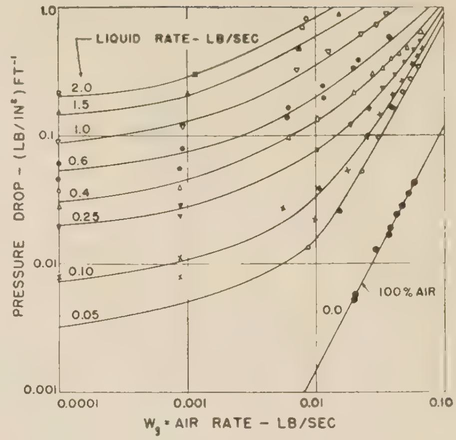

Fig. 5 Pressure Drop for Flow of Air and Blend No. 1 of Diesel Oil and Oil B in 1-In. Glass Pipe
(Viscosity of liquid = 87.5×10^{-5}, lb sec/ft².)

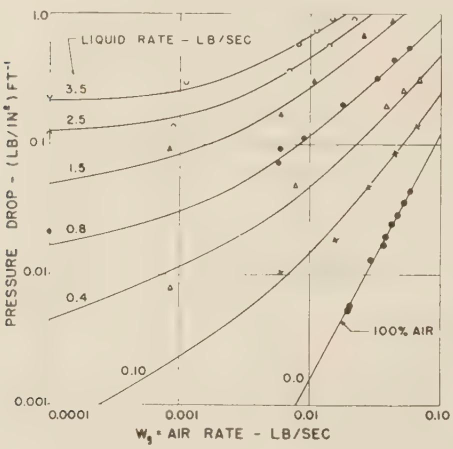

Fig. 3 Pressure Drop for Flow of Air and Kerosene in 1-In. Glass Pipe
(Viscosity of liquid = 4.35×10^{-5}, lb sec/ft².)

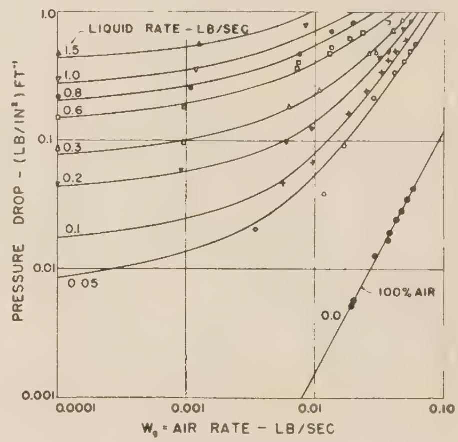

Fig. 6 Pressure Drop for Flow of Air and Blend No. 2 of Diesel Oil and Oil B in 1-In. Glass Pipe
(Viscosity of liquid = 257×10^{-5}, lb sec/ft².)

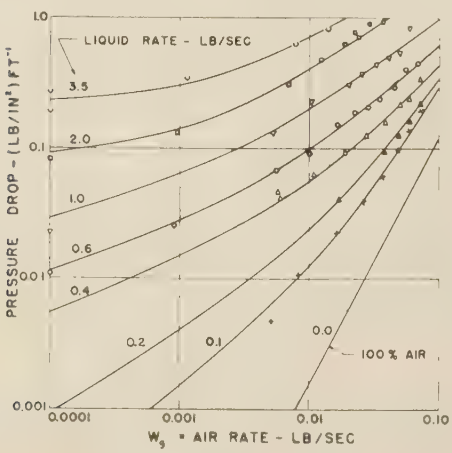

Fig. 4 Pressure Drop for Flow of Air and Diesel Fuel Oil in 1-In. Glass Pipe
(Viscosity of liquid = 10.35×10^{-5}, lb sec/ft².)

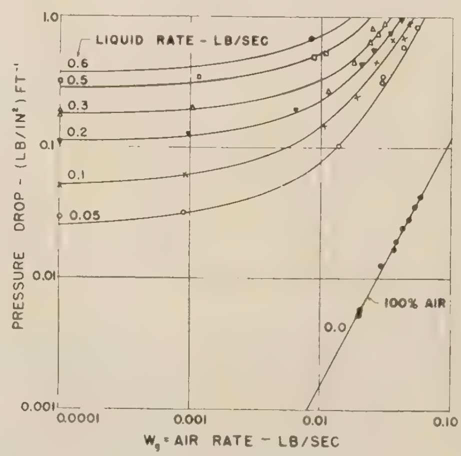

Fig. 7 Pressure Drop for Flow of Air and Oil B (S.A.E. 40) in 1-In. Glass Pipe
(Viscosity of liquid = 588×10^{-5}, lb sec/ft².)

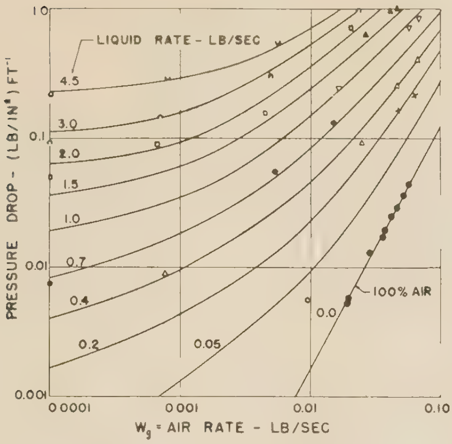

Fig. 8 Pressure Drop for Flow of Air and Water Plus Nekal-BX in 1-In. Glass Pipe

(The lines for constant liquid rates are the same as those shown in Fig. 2.)

TABLE 2 OBSERVATIONS OF FLOW MECHANISMS[a]

	Low air rate		Medium air rate		High air rate	
	Liq. turb.	Liq. visc.	Liq. turb.	Liq. visc.	Liq. turb.	Liq. visc.
Very low liquid rate..	3	3	3	3	8	8
Low liquid rate......	5	3	6	3	8	8
Medium liquid rate...	7	7	4	4	4	4
High liquid rate......	2	–	2	–	2	–

[a] See Fig. 9 for sketch of flow types.

phase must equal the static pressure drop for the gaseous phase regardless of the details of the flow, as long as appreciable static pressure differences do not exist along any pipe diameter. The second postulate is that the volume occupied by the liquid plus the volume occupied by the gas at any instant must equal the total volume of the pipe. The expression of these two propositions in algebraic form leads to the following equations for the prediction of two-phase-flow static pressure drop:

The static pressure drop due to the liquid flow may be written by the usual Weisbach equation as

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \zeta_l \frac{\gamma_l}{D_l} \cdot \frac{v_l^2}{2g} \qquad [1]$$

where D_l is the unknown "hydraulic diameter" of the liquid flow pattern. Similarly, for the gas flow

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \zeta_g \frac{\gamma_g}{D_g} \cdot \frac{v_g^2}{2g} \qquad [2]$$

The pressure drop in two-phase flow is greater than that for the flow of either single phase alone for various reasons, among which are the irreversible work done by the gas on the liquid and the fact that the presence of the second fluid reduces the cross-sectional area of flow for the first fluid. Thus, during two-phase flow the hydraulic diameters D_l and D_g will always be less than the pipe diameter D_p, and as noted in Equations [1] and [2], this reduction of hydraulic diameter will increase the pressure drop greatly. It should be observed that for a given weight rate of fluid the pressure drop varies inversely as the fourth or fifth power of the hydraulic diameter, depending upon the type of flow.

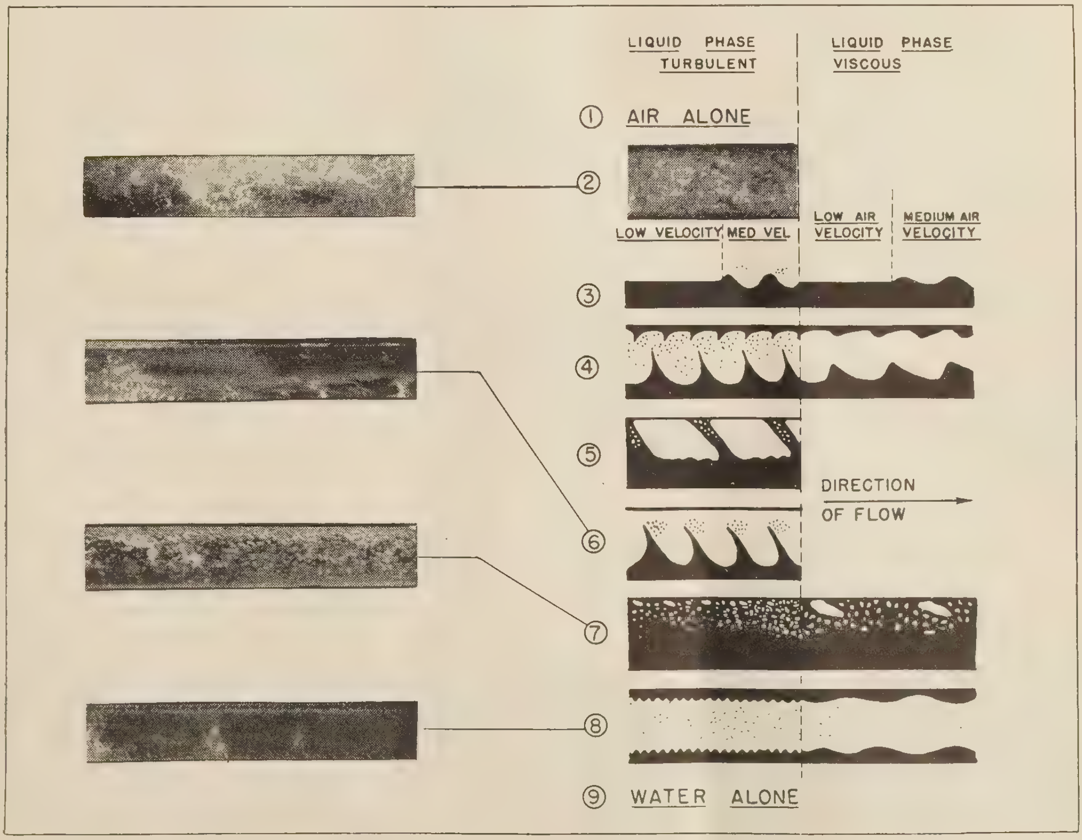

Fig. 9 Flow Types Occurring in 1-In. Glass Pipe

The hydraulic diameter of the flow pattern is related to the cross-sectional area through which the fluid is flowing at any instant. For a cylindrical flow pattern the simple relation

$$\frac{\pi}{4} D^2_H = A_c \qquad [3]$$

applies. For a more complex cross-sectional area, such as that possessed by the fluids in the pipe, the following relations may be written

$$A_l = \alpha\left(\frac{\pi}{4} D^2_l\right) \qquad [4]$$

$$A_g = \beta\left(\frac{\pi}{4} D^2_g\right) \qquad [5]$$

where α and β are, in effect, the ratio of the actual cross-sectional area of flow to the area of a circle of diameter D_l and D_g, respectively. A consideration of the air flow indicates that as a first approximation, the cross-sectional shape of the air flow is approximately circular at all times, so that β can be considered as 1. On the other hand, at high rates of gas flow and low rates of liquid flow, the liquid can assume a thin annular shape, and thus, for this case, α is much greater than unity. As a reasonable approximation, therefore, α will be left as an unknown in Equation [4] while β is taken as unity.[10]

Equations [4] and [5] may be utilized to evaluate the respective velocities v_l and v_g; thus

$$v_l = \frac{W_l}{\alpha\left(\frac{\pi}{4} D_l^2\right)\gamma_l} \qquad [6]$$

$$v_g = \frac{W_g}{\left(\frac{\pi}{4} D_g^2\right)\gamma_g} \qquad [7]$$

The velocities, v_l and v_g, calculated from Equations [6] and [7], are the mean absolute velocities of the two fluids. The velocities v_l and v_g in Equations [1] and [2] involve the "relative" velocities between the fluids, and thus substitution of the magnitudes of the velocities from Equations [6] and [7] into Equations [1] and [2] is not exact.[11] This difficulty is not as serious as it first appears, for the ratio α is evaluated from experimental data and thus includes the effect of relative motion as well as the fluid geometry. A more precise analysis of the problem of two-phase pressure drop should include a consideration of the matter of relative motion in greater detail.

The friction factors ζ_l and ζ_g may be expressed in the generalized Blasius form

$$\zeta_l = \frac{\left(\frac{\pi}{4}\right)^n C_l}{\left(\frac{W_l}{\alpha D_l \mu_l g}\right)^n} \qquad [8]$$

and similarly

$$\zeta_g = \frac{\left(\frac{\pi}{4}\right)^m C_g}{\left(\frac{W_g}{D_g \mu_g g}\right)^m} \qquad [9]$$

For viscous flow, the constants C_g, C_l are equal to 64 and the exponents n and m are both unity. For turbulent flow C_l and C_g and n and m vary with the Reynolds modulus, but an average magnitude (3) for C_g and $C_l = 0.184$, and n and $m = 0.20$ may be utilized for smooth tubes for a range of the Reynolds modulus from 5000 to 200,000.

The proper choice of C_l, C_g and n, m depends upon the type of flow occurring in the pipe. Four distinct flow mechanisms are possible, as follows:

1 The flow of both the liquid and gas may be turbulent ($C_l = C_g = 0.184$ and $n = m = 0.2$).

2 The flow of the gas may be turbulent, and the flow of the liquid viscous ($C_l = 64$, $C_g = 0.184$, $n = 1$, $m = 0.2$).

3 The flow of both the liquid and gas may be viscous ($C_l = C_g = 64$ and $n = m = 1$).

4 The flow of the gas may be viscous and that of the liquid turbulent ($C_l = 0.184$, $C_g = 64$, $m = 1$, $n = 0.2$).

Of these four possible cases, the fourth is very improbable and thus has not been analyzed in detail, although the analysis would be almost identical to that of the second flow mechanism. The third case was not observed experimentally but has been analyzed since this mechanism may arise in practice (viz., flow in porous media). As discussed previously, both the first and second flow combinations were observed experimentally and are analyzed in the following sections of the paper.

Flow Mechanism 1. Liquid Turbulent, Gas Turbulent

In flow mechanism 1, in which both fluids are in turbulent motion, the magnitudes of C_l and C_g are equal to 0.184 for smooth pipes, and the exponents m and n are both equal to 0.2. Substitution of these magnitudes in Equations [8] and [9], and a further substitution of Equations [8], [9], [6], [7] into Equations [1] and [2] results in the equation

$$\left(\frac{D_l}{D_g}\right)^2 = \left(\frac{\gamma_g}{\gamma_l}\right)^{\frac{2}{5-n}} \left(\frac{\mu_l}{\mu_g}\right)^{\frac{2n}{5-n}} \left(\frac{W_l}{W_g}\right)^{\frac{2(2-n)}{5-n}} (\alpha)^{\frac{2(n-2)}{5-n}} \qquad [10]$$

It is noted that, since the constants C_l and C_g are equal for both fluid phases, they cancel in the derivation of Equation [10], while the exponent n remains. In Equation [10], the three unknowns are D_l, D_g, and α. Equation [10] is the result of the first basic postulate previously discussed, i.e., the equality of the pressure drops for the two phase of flow. Substitution of $n = 0.2$ yields

$$\left(\frac{D_l}{D_g}\right)^2 = \left(\frac{\gamma_g}{\gamma_l}\right)^{0.416} \left(\frac{\mu_l}{\mu_g}\right)^{0.083} \left(\frac{W_l}{W_g}\right)^{0.75} \alpha^{-0.75} \qquad [11]$$

Application of the second basic postulate, equating the sum of the liquid and gas volumes to the pipe volume yields

$$\alpha\left(\frac{\pi}{4} D_l^2 L\right) + \left(\frac{\pi}{4} D_g^2 L\right) = \frac{\pi}{4} D_p^2 L$$

$$\alpha D_l^2 + D_g^2 = D_p^2 \qquad [12]$$

In Equation [12], the variables D_g and D_l are unknown. One of the unknown "hydraulic diameters," D_l, may be eliminated between Equations [11] and [12]. The resulting expression for D_g is

$$D_g = \frac{D_p}{\sqrt{1 + \alpha^{1/4}\left(\frac{\gamma_g}{\gamma}\right)^{0.416}\left(\frac{\mu_l}{\mu}\right)^{0.083}\left(\frac{W_l}{W}\right)^{0.75}}} \qquad [13]$$

[10] A more precise analysis can be performed in which both α and β are left as unknowns. The modulus β can be assumed equal to unity and α estimated. By rearranging the equations, β can then be solved for approximately, utilizing the approximate magnitudes of α. A repetition of this trial-and-error procedure will finally yield magnitudes of both α and β. The present analysis, in effect, includes any variations of β in the modulus α. This may be one explanation of the scattering of the experimental points observed in Figs. 10 and 14.

[11] Noted by V. Skoglund in a verbal discussion of this paper at the 1943 Semi-Annual Meeting in Los Angeles, Calif.

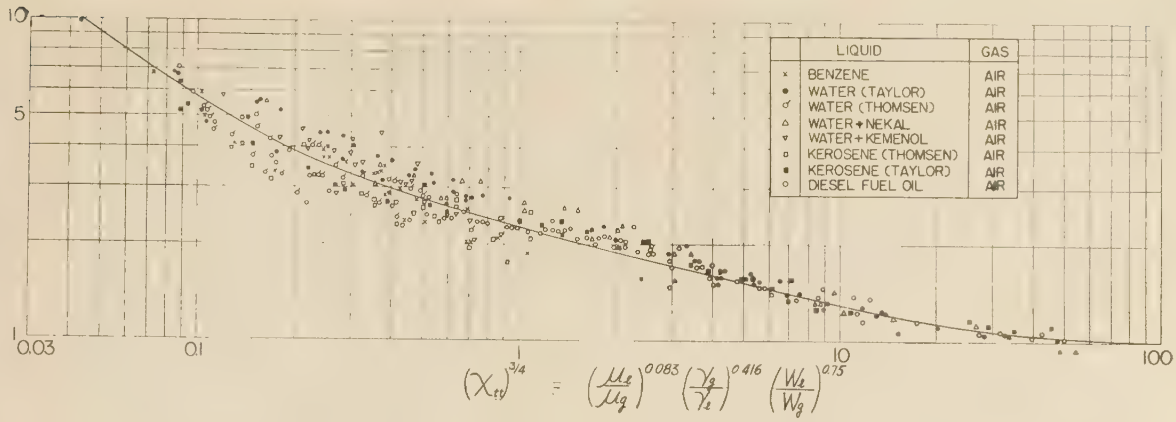

Fig. 10 Flow Type Modulus $\alpha^{1/4}$ as a Function of the Two-Phase Flow Modulus $(\chi_{tt})^{3/4}$. Flow Mechanism in Which Both Liquid and Gas Are in Turbulent Motion

It may be shown by reference to Equation [2] that the static pressure drop for the gas phase during the two-phase flow is:

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \frac{\left(\frac{4}{\pi}\right)^{2-m} C_g}{\left(\frac{W_g}{D_g \mu_g g}\right)^m} \cdot \frac{1}{D_g} \cdot \frac{W_g^2}{2g D_g^4 \gamma_g} \qquad [14]$$

$$= \left[\frac{1}{2}\left(\frac{4}{\pi}\right)^{2-m} \frac{C_g \mu_g^m}{g^{1-m}\gamma_g} \cdot \frac{W_g^{2-m}}{D_p^{5-m}}\right]\left(\frac{D_p}{D_g}\right)^{5-m} \qquad [14]$$

$$= \left(\frac{\Delta P}{\Delta L}\right)_g \left(\frac{D_p}{D_g}\right)^{5-m} \qquad [15]$$

For the case under consideration $m = 0.2$, thus

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \left(\frac{\Delta P}{\Delta L}\right)_g \left(\frac{D_p}{D_g}\right)^{4.8} \qquad [16]$$

The term in brackets in Equation [14] may be demonstrated to be the pressure drop for the gas phase alone flowing with a weight rate W_g and density γ_g, in a pipe of diameter D_p. Thus, the pressure drop for two-phase flow is obtained by multiplying the static pressure drop for the gas phase alone (corresponding to W_g, γ_g, and D_p) by the ratio $(D_p/D_g)^{5-m}$. When both fluids are in turbulent motion, this ratio can be obtained from Equation [13] and substituted in Equation [16].

The final equation for the static pressure drop per unit length, when both the fluids are flowing in the turbulent regime, is then

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \left(\frac{\Delta P}{\Delta L}\right)_g \left[1 + \alpha^{1/4}\left(\frac{\mu_l}{\mu_g}\right)^{0.083}\left(\frac{\gamma_g}{\gamma_l}\right)^{0.416}\left(\frac{W_l}{W_g}\right)^{0.75}\right]^{2.4} \qquad [17]$$

Inspection of Equations [17] and [14] reveals that, as W_g approaches zero, the equation reduces to the expression for the pressure drop of the liquid flowing alone with a weight rate W_l through a pipe of diameter D_p.

In order to establish the cases for which Equation [17] is applicable, the Reynolds modulus for each fluid is calculated utilizing the rates W_g and W_l, the fluid viscosities, and the pipe diameter D_p. If the magnitude of both of these Reynolds moduli is greater than 2000, Equation [17] is applicable. An example of the method of calculation is shown in Appendix 1.

Evaluation of α for Flow Mechanism 1. In Equation [17] the function α is still unknown. This function, as mentioned earlier, represents the ratio of the actual cross-sectional area of liquid flow to the area calculated by the product $\frac{\pi}{4} D_l^2$ and should equal unity when the liquid completely fills the pipe and be greater than unity when the liquid flows along the pipe in a thin annulus. The function α was calculated from experimental data by substituting known magnitudes of the variables into Equation [17] and then solving for $\alpha^{1/4}$. The magnitudes of $\alpha^{1/4}$ varied from unity when the pipe was filled with liquid to about 10 when the liquid was flowing in an annulus.

In order to generalize the test results, a parameter was sought which would allow the correlation of the magnitudes of $\alpha^{1/4}$ for the various fluids and pipe diameters for which data were available. Many dimensionless groups were tried, but the best correlation of the data was obtained when the multiplier of $\alpha^{1/4}$ in Equation [17] was utilized. This parameter represents the ratio of shearing forces in the two fluid phases, and thus is a logical basis for the correlation of the magnitudes of α.

For simplicity in calculation, the dimensionless group

$$\chi_{tt} = \left(\frac{\mu_l}{\mu_g}\right)^{0.111}\left(\frac{\gamma_g}{\gamma_l}\right)^{0.555}\left(\frac{W_l}{W_g}\right) \qquad [18]$$

was defined. The multiplier of $\alpha^{1/4}$ in Equation [17] is thus $(\chi_{tt})^{3/4}$. The subscript tt refers to the turbulent flow of the liquid and gas.

In Fig. 10 is shown the plot of $\alpha^{1/4}$ as a function of $(\chi_{tt})^{3/4}$. The scattering of the experimental points is rather severe at low magnitudes of the parameter χ_{tt}, but when it is observed that five different liquids, and data from two experimenters are plotted, the correlation is, on the whole, satisfactory. The scattering of the points is in part due to the use of a constant exponent of 0.20 in the empirical equation for the friction factor; the actual exponent varies with the Reynolds modulus. Further, due to the "slugging" flow of the liquid during certain runs, the pressure drop varied periodically with time, and the recorded mean pressure drop may not represent a true time average. Finally, the fact that $\alpha^{1/4}$ includes the effect of relative motion between the fluids, as well as the flow geometry, may cause some scattering. There seems to be some slight evidence that a third parameter may enter the correlation since the points for water are on the average somewhat higher than the points for the hydrocarbon liquids. The variation of surface tension obtained in these tests seems to be of little importance in the evaluation of $\alpha^{1/4}$, since the points for water, and the water-Kemenol, and water-Nekal mixtures are almost coincident.

From the point of view of ease in calculation, an interesting

point is to be noted; since $\alpha^{1/4}$ is a function of χ_{tt}, inspection of Equation [17] reveals that

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \left(\frac{\Delta P}{\Delta L}\right)_g [1 + \alpha^{1/4}\chi_{tt}{}^{3/4}]^{2.4}$$

$$= \left(\frac{\Delta P}{\Delta L}\right)_g [1 + f(\chi_{tt})(\chi_{tt})^{3/4}]^{2.4} \quad \text{[19]}$$

Thus the pressure drop in two-phase flow depends only upon the product of the pressure drop of the gaseous component flowing alone, and a function of χ_{tt}. The latter function may be readily evaluated. A curve was faired through the experimental points

TABLE 3 MAGNITUDES OF THE FUNCTION Φ_{tt}

$\sqrt{\chi_{tt}}$	Φ_{tt}
0	1.00
0.10	1.50
0.20	1.68
0.40	2.13
0.70	3.03
1.00	4.08
2.00	8.30
4.00	19.6
7.00	42.3
10.0	71.0
20.0	222
40.0	770
46.2	1000
For $\sqrt{\chi_{tt}} > 46$	$\Phi_{tt} = (\sqrt{\chi_{tt}})^{1.8}$

of Fig. 10. For each magnitude of χ_{tt} a value of $\alpha^{1/4}$ was established. Then

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \left(\frac{\Delta P}{\Delta L}\right)_g [1 + \alpha^{1/4}\chi_{tt}{}^{3/4}]^{2.4} = \Phi_{tt}{}^2 \left(\frac{\Delta P}{\Delta L}\right)_g \quad \text{[20]}$$

where

$$\Phi_{tt} = [1 + \alpha^{1/4}\chi_{tt}{}^{3/4}]^{1.2}$$

Table 3 presents magnitudes of Φ_{tt} versus $\sqrt{\chi_{tt}}$. The square-root form was utilized in order to reduce the range of the variables to facilitate plotting. Fig. 11 presents Φ_{tt} versus $\sqrt{\chi_{tt}}$ and may be utilized directly for the calculation of two-phase pressure drop.

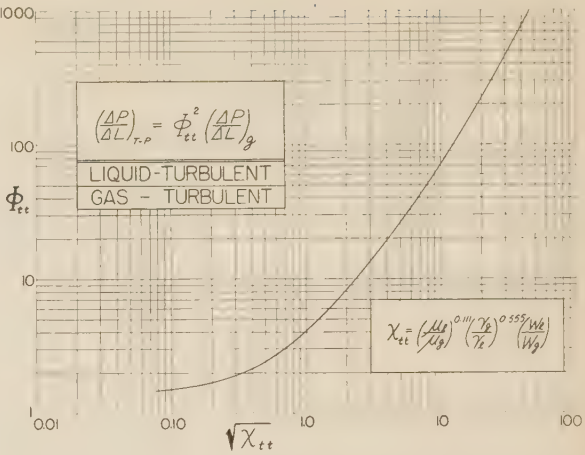

FIG. 11 MODULUS Φ_{tt} AS A FUNCTION OF TWO-PHASE FLOW MODULUS $\sqrt{\chi_{tt}}$ FOR FLOW MECHANISM IN WHICH BOTH LIQUID AND GAS ARE IN TURBULENT MOTION. THE MODULUS Φ_{tt} MAY BE UTILIZED DIRECTLY TO CALCULATE THE PRESSURE DROP OCCURRING IN TWO-PHASE FLOW

Fig. 11 reveals that, as χ_{tt} approaches zero, Φ_{tt} approaches unity very slowly due to the rapid increase of $\alpha^{1/4}$ at low magnitudes of χ_{tt}. It follows that, even if the fluid just wets the pipe wall (low values of χ_{tt}) the two-phase pressure drop is increased appreciably over that due to the gaseous components alone. For $\sqrt{\chi_{tt}}$ equal to 0.10 which corresponds to a reasonable liquid flow, the two-phase pressure drop is 2.25 times that of the gas alone. At high magnitudes of $\sqrt{\chi_{tt}}$ the curve of Φ_{tt} approaches an asymptote with a slope of 1.8.

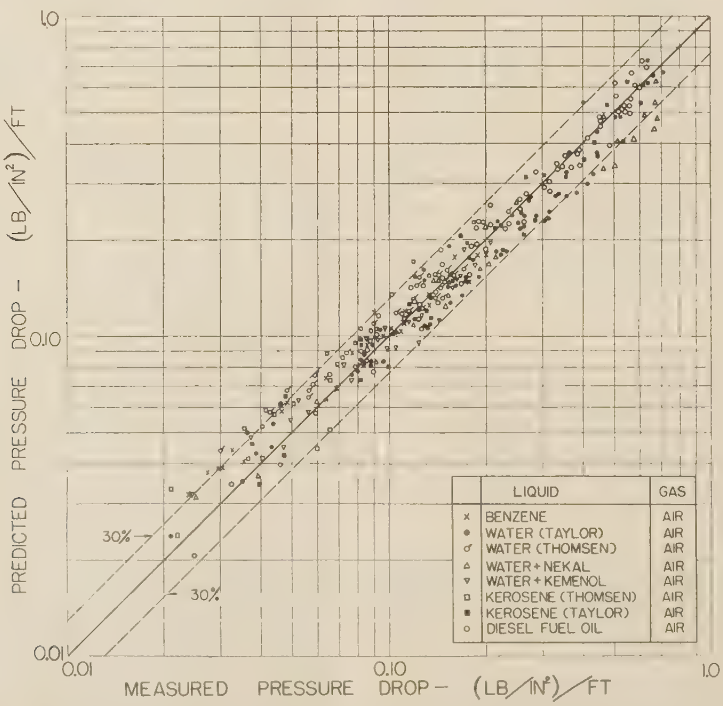

FIG. 12 COMPARISON OF PREDICTED AND MEASURED PRESSURE DROP FOR TWO-PHASE TWO-COMPONENT FLOW. FLOW MECHANISM IN WHICH BOTH LIQUID AND GAS ARE IN TURBULENT MOTION. PREDICTION BY MEANS OF FIG. 11

Fig. 11 was utilized to predict the pressure drops observed in the laboratory by calculating χ_{tt} based on experimental values of μ_g, μ_l, γ_g, γ_l, W_g, and W_l. The results of these calculations are shown in Fig. 12. The prediction is seen to be well within ± 30 per cent for a large majority of the points. Thus, within the range of the experimental variables which are listed in Table 1, Equation [17], or its equivalent Fig. 11, may be utilized to predict the pressure drop accompanying two-phase two-component flow when both phases are in turbulent motion. Extrapolation of Equation [17] to magnitudes of the variables greatly different from those utilized in the tests should be made with caution until the general applicability of the parameter χ_{tt} is tested by further research.

In order to determine how well Equation [17] could predict the curve shapes obtained experimentally, Fig. 11 was utilized to predict the pressure drop along a 1-in. glass pipe in which air and water were flowing at an average pressure of 14.7 psia and a temperature of 66 F. The resulting predicted curves of pressure drop versus air rate for several constant liquid rates are shown in Fig. 13. The agreement between these curves and the experimental points shown in Fig. 2 is satisfactory. Lack of agreement at some points is no doubt due in part to the rather arbitrary method utilized to correct the experimental data shown in Fig. 2 to standard conditions (see Appendix 2).

FLOW MECHANISM 2. LIQUID VISCOUS, GAS TURBULENT

The equation for the two-phase pressure drop for the second flow mechanism is derived by substituting $m = 0.2$ and $n = 1$ in Equations [8] and [9]. Further substitution of Equations [8], [9], [6], [7] into Equation [1] and [2] yields

$$\frac{D_l^4}{D_g^{4.8}} = \left(\frac{\pi}{4}\right)^{0.8} g^{0.8}\,\frac{C_l}{C_g}\cdot\frac{\gamma_g}{\gamma_l}\cdot\frac{\mu_l}{\mu_g^{0.2}}\cdot\frac{W_l}{W_g^{1.8}}\cdot\frac{1}{\alpha}\ldots\ldots[21]$$

or

$$\left(\frac{D_l}{D_g}\right)^4 = \left(\frac{\frac{\pi}{4}\,g\,\mu_g\,D_p}{W_g}\right)^{0.8}\frac{C_l}{C_g}\cdot\frac{\gamma_g}{\gamma_l}\cdot\frac{\mu_l}{\mu_g}\cdot\frac{W_l}{W_g}\left(\frac{D_g}{D_p}\right)^{0.8}\frac{1}{\alpha}\ldots[22]$$

$$\left(\frac{D_l}{D_g}\right)^2 = (Re_g)^{-0.4}\sqrt{\frac{C_l}{C_g}\cdot\frac{\gamma_g}{\gamma_l}\cdot\frac{\mu_l}{\mu_g}\cdot\frac{W_l}{W_g}}\left(\frac{D_g}{D_p}\right)^{0.4}\alpha^{-1/2}\ldots[23]$$

Both C_l and C_g appear in Equation [23] in contrast to Equation [10], in which neither constant appeared. For smooth pipes, magnitudes of $C_l = 64$ and $C_g = 0.184$ may be utilized as a good approximation.

It is convenient at this point to define a variable χ_{vt} which will be utilized to correlate the magnitudes of $\alpha^{1/2}$ for the second flow mechanism.

$$\chi_{vt} = (Re_g)^{-0.8}\left(\frac{C_l}{C_g}\cdot\frac{\gamma_g}{\gamma_l}\cdot\frac{\mu_l}{\mu_g}\cdot\frac{W_l}{W_g}\right)\ldots\ldots\ldots[24]$$

Then Equation [23] may be written as

$$\left(\frac{D_l}{D_g}\right)^2 = \sqrt{\chi_{vt}}\left(\frac{D_g}{D_p}\right)^{0.4}\alpha^{-1/2}\ldots\ldots\ldots\ldots[25]$$

The second fundamental relation, equating the sum of the gas and liquid volumes to the pipe volume yields, as before

$$\alpha D_l^2 + D_g^2 = D_p^2\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots[26]$$

Eliminating D_l between Equations [25] and [26] results in the equation

$$\left(\frac{D_p}{D_g}\right) = \left[\left(\frac{D_p}{D_g}\right)^{0.4} + \alpha^{1/2}\sqrt{\chi_{vt}}\right]^{1/2.4}\ldots\ldots\ldots[27]$$

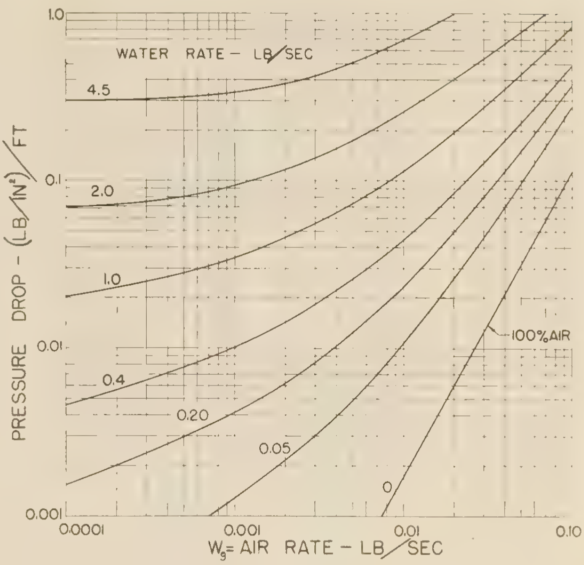

FIG. 13 PREDICTED PRESSURE DROP IN TWO-PHASE FLOW AS A FUNCTION OF FLUID RATES. AIR-WATER FLOW IN A 1-IN. GLASS PIPE, p = 14.7 PSIA, t = 66 F. BOTH FLUIDS IN TURBULENT MOTION. PREDICTION BY MEANS OF FIG. 11

(Compare with experimental curves in Fig. 2.)

Equation [15] is again applicable; thus

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \left(\frac{\Delta P}{\Delta L}\right)_g\left(\frac{D_p}{D_g}\right)^{5-m}\ldots\ldots\ldots\ldots[28]$$

The exponent $m = 0.2$, since the gas is flowing with turbulent motion. Substituting Equation [27] into Equation [28] yields

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \left(\frac{\Delta P}{\Delta L}\right)_g\left[\left(\frac{D_p}{D_g}\right)^{0.4} + \alpha^{1/2}\sqrt{\chi_{vt}}\right]^2\ldots[29]$$

When the term $\left(\frac{D_p}{D_g}\right)^{0.4}$ is important in Equation [29], χ_{vt} is small. This occurs when W_g is large and W_l small, i.e., when the pipe is filled with gas. Thus, it is permissible, with little error, to replace $\left(\frac{D_p}{D_g}\right)^{0.4}$ by unity. The equation for two-phase pressure drop, for the case when the liquid is flowing viscously and the gas is in turbulent motion, then becomes

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \left(\frac{\Delta P}{\Delta L}\right)_g\left[1 + \alpha^{1/2}(Re_g)^{-0.4}\sqrt{\frac{C_l}{C_g}\cdot\frac{\gamma_g}{\gamma_l}\cdot\frac{\mu_l}{\mu_g}\frac{W_l}{W_g}}\right]^2\ldots[30]$$

In order to establish the applicability of Equation [30] the Reynolds modulus for each fluid phase must be calculated, utilizing the rates W_g and W_l, the fluid viscosities, and the pipe diameter. If the Reynolds modulus for the liquid (Re_l) is less than 2000, and if the Reynolds modulus for the gas (Re_g) is greater than 2000, Equation [30] is applicable. An example of the method of calculation is shown in Appendix 1.

Evaluation of α for Flow Mechanism 2. Equation [30] was utilized to evaluate $\alpha^{1/2}$ from experimental data. The function $\alpha^{1/2}$ was found to vary from 1 at high liquid rates and low gas rates to about 5 at low liquid rates and high gas rates. The magnitudes of $\alpha^{1/2}$ for the three fluids employed in the test were correlated by means of χ_{vt}, as defined by Equation [24]. The

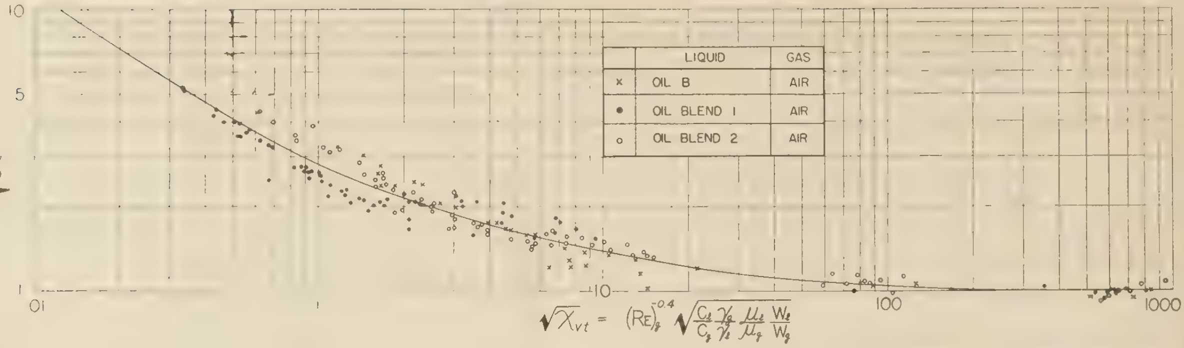

Fig. 14 Flow Type Modulus $\alpha^{1/2}$ as a Function of Two-Phase Flow Modulus $\sqrt{\chi_{vt}}$. Flow Mechanism in Which the Liquid Flows Viscously and the Gas Is in Turbulent Motion

TABLE 4 MAGNITUDES OF THE FUNCTION Φ_{vt}

$\sqrt{\chi_{vt}}$	Φ_{vt}
0	1.00
0.07	2.00
0.10	2.14
0.20	2.46
0.40	2.92
0.70	3.42
1.00	3.85
2.00	5.30
4.00	7.87
7.00	11.3
10.0	14.8
20.0	25.4
40.0	46.0
70.0	75.8
100	105
200	203
400	400
1000	1000
For $\sqrt{\chi_{vt}} > 1000$	$\Phi_{vt} = \sqrt{\chi_{vt}}$

resulting plot of $\alpha^{1/2}$ versus $\sqrt{\chi_{vt}}$ is shown in Fig. 14. The scattering of the data is about the same as in Fig. 10, evidence of a third parameter again being present, but the trend is not conclusive. A single curve was faired through the experimental points, and Φ_{vt} defined by

$$\Phi_{vt} = [1 + \alpha^{1/2}\sqrt{\chi_{vt}}] \ldots\ldots\ldots\ldots\ldots\ldots [31]$$

was calculated. The resulting curve is shown in Fig. 15 and the data are tabulated in Table 4.

Fig. 15 reveals that, again, due to the rapid rise of $\alpha^{1/2}$ at low magnitudes of χ_{vt}, the curve of Φ_{vt} approaches unity slowly. Thus, even at very low liquid rates and high gas rates (walls just wetted) the two-phase pressure drop is increased greatly over that due to the gas phase alone. A small amount of viscous oil flowing along the walls of a pipe ($\sqrt{\chi_{vt}} = 0.10$) may quadruple the pressure drop. The curve of Φ_{vt} versus $\sqrt{\chi_{vt}}$ reaches an asymptote with a slope of unity at high magnitudes of $\sqrt{\chi_{vt}}$.

A plot of the predicted pressure drops versus the measured pressure drops is shown in Fig. 16. The prediction was accomplished by calculating χ_{vt} for the experimental conditions, obtaining Φ_{vt}, and calculating

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \Phi_{vt}^2\left(\frac{\Delta P}{\Delta L}\right)_g \ldots\ldots\ldots\ldots\ldots [32]$$

The shape of the experimental curves, shown in Fig. 7, are well predicted by Equation [30]. Fig. 15 was used to calculate the pressure drop along a 1-in. glass pipe in which air and oil B were flowing at an average pressure of 14.7 psia and a temperature of 66 F. The predicted curves of pressure drop versus air rate at constant oil rates are shown in Fig. 17 and agree very well with the experimental data of Fig. 7.

It may be concluded that Equation [30], or its equivalent Fig. 15, can be utilized to predict pressure drop in two-phase flow when the gas is flowing with turbulent motion and the liquid with viscous motion. Extrapolation to magnitudes of the variables very much different from those covered by the tests are to be made with caution until the general applicability of the parameter χ_{vt} is tested by further research.

Flow Mechanism 3. Liquid Viscous, Gas Viscous

Although the type of two-phase flow in which both phases are flowing with viscous motion was not observed in the tests on the 1-in. and 1/2-in. pipe, the analysis of this phenomenon will be presented. Flow of two phases in porous media is of this viscous character, and perhaps an application of the equations which have been developed to this type of problem is possible.

It will be readily seen that Equation [10] is directly applicable to this problem, with the exponent $n = 1$. This substitution yields

$$\left(\frac{D_p}{D_g}\right)^2 = \left(\frac{\gamma_g}{\gamma_l}\right)^{1/2}\left(\frac{\mu_l}{\mu_g}\right)^{1/2}\left(\frac{W_l}{W_g}\right)^{1/2}\frac{1}{\alpha^{1/2}} \ldots\ldots [33]$$

Substitution of Equation [33] into Equations [12] and [15] results in the equation for pressure drop in two-phase flow when both phases are flowing with viscous motion

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = \left(\frac{\Delta P}{\Delta L}\right)_g\left[1 + \alpha^{1/2}\left(\frac{\mu_l}{\mu_g}\right)^{1/2}\left(\frac{\gamma_g}{\gamma_l}\right)^{1/2}\left(\frac{W_l}{W_g}\right)^{1/2}\right]^2 \ldots [34]$$

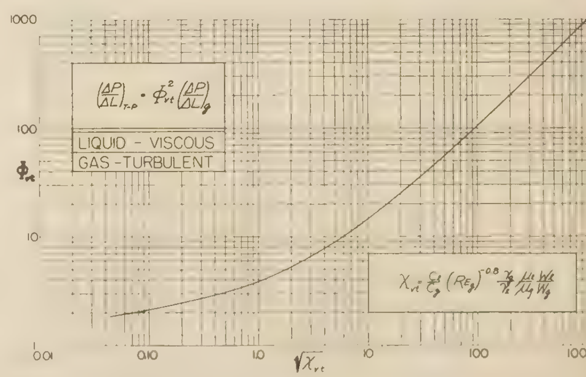

Fig. 15 Modulus Φ_{vt} as a Function of Two-Phase-Flow Modulus $\sqrt{\chi_{vt}}$, for Flow Mechanism in Which the Liquid Flows Viscously and the Gas Is in Turbulent Motion. The Modulus Φ_{vt} May Be Utilized Directly to Calculate the Pressure Drop Occurring in Two-Phase Flow

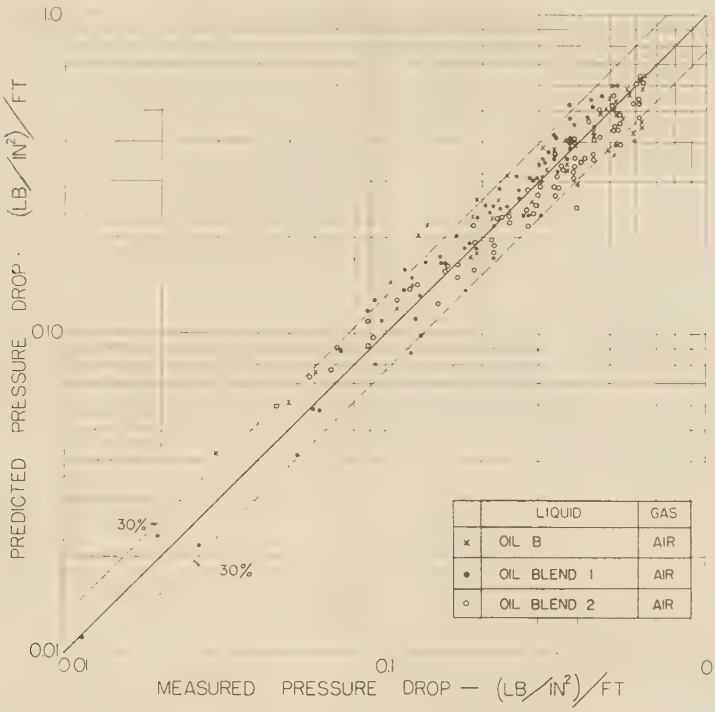

Fig. 16 Comparison of Predicted and Measured Drop for Two-Phase, Two-Component Flow. The Liquid Flows Viscously and the Gas Is in Turbulent Motion. Prediction by Means of Fig. 15

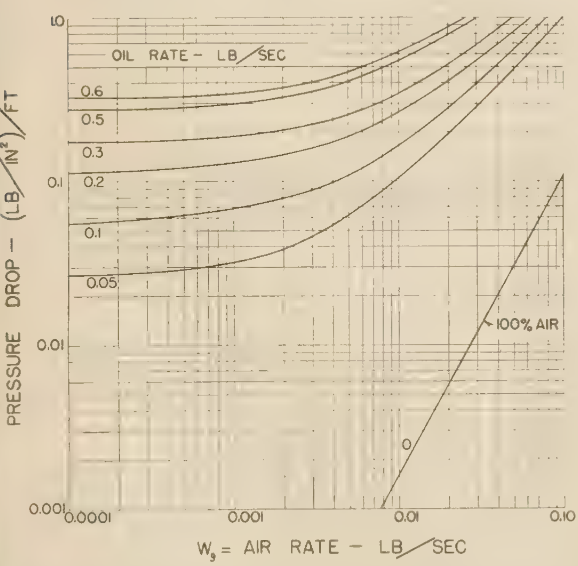

Fig. 17 Predicted Pressure Drop in Two-Phase Flow as a Function of Fluid Rates. Air-Oil B Flow in a 1-In. Glass Pipe p = 14.7 Psia, t = 66 F. Oil Flowing Viscously, Air in Turbulent Motion. Prediction by Means of Fig. 15
(Compare with experimental curves in Fig. 7.)

Again $\left(\frac{\Delta P}{\Delta L}\right)_g$ is the pressure drop which would exist if the gas alone were flowing through the pipe of diameter D_p with weight rate W_g and density γ_g. In the case of flow through porous media $\left(\frac{\Delta P}{\Delta L}\right)_g$ would probably be the pressure drop through a given medium for the gas phase flowing alone with a rate W_g.

The function $\alpha^{1/2}$ will probably be correlated by means of the parameter $\chi_{vv} = \left(\frac{\mu_l}{\mu_g} \cdot \frac{\gamma_g}{\gamma_l} \cdot \frac{W_l}{W_g}\right)$.

Flow Mechanism 4. Gas Viscous, Liquid Turbulent

The fourth flow mechanism, in which the gas is flowing viscously and the liquid has turbulent motion, is very unlikely to exist in practice. The solution of this problem would, however, follow in detail the solution for the second flow mechanism, by merely interchanging the exponents of the gas and liquid phases.

Conclusions

1 There are four flow mechanisms possible in two-phase flow, as follows:

(a) Both gas and liquid may flow in turbulent motion.
(b) Gas may flow turbulently, liquid viscously.
(c) Both gas and liquid may flow viscously.
(d) Gas may flow viscously, liquid turbulently.

2 The first two mechanisms have been noted experimentally and have been analyzed macroscopically in this paper. The third type has been analyzed but has not been observed by the authors, although this mechanism undoubtedly exists in practice.

3 Equation [17], or Fig. 11, allows the prediction of two-phase pressure drop for the flow mechanism in which both phases flow with turbulent motion. Equation [17] predicts the experimental pressure drop with a maximum error of about ± 30 per cent.

4 Equation [30], or Fig. 15, allows the prediction of two-phase pressure drop for the flow mechanism in which the liquid is flowing viscously and the gas turbulently. Equation [30] predicts the experimental pressure drop with a maximum error of about ± 30 per cent.

5 Extrapolation of the equations to variables with magni-

tudes which are much different from those tested must be made with caution until the general applicability of the parameters χ_{tt} and χ_{vt} has been tested by further research.

6 Equation [34] is suggested as the equation to be utilized to predict two-phase pressure drop for the flow mechanism in which both phases are flowing viscously. The relation between α and χ_{vv} is still to be established.

Acknowledgments

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Appendix 1

In order to illustrate the application of the method outlined in the paper, three examples will be presented. The first is an example of the prediction of the pressure drop in a 2-in. horizontal pipe in which both air and water are flowing at known rates. The second is an example of the calculation of the pressure drop along a 2-in. pipe in which air and an oil are flowing simultaneously. The third example presents a tentative outline for the application of the proposed method to the prediction of the pressure drop along a horizontal tube in which a liquid is being evaporated.

Example 1

It is desired to estimate the pressure drop in 100 ft of 2-in. pipe, in which air is flowing at a rate of 0.50 lb per sec, and water is flowing at a rate of 6 lb per sec. The temperature of the fluids is 77 F and the average pressure in the line is 100 psia.

(*a*) *Determination of properties and type of flow:*

W_g = 0.50 lb per sec
γ_g = 0.502 lb per cu ft at 100 psia and 77 F
μ_g = 0.388×10^{-6} lb-sec per sq ft at 100 psia and 77 F

Reynolds modulus for gas considered as a single phase (Re_g) = 295,000

W_l = 6.0 lb per sec
γ_l = 62.2 lb per cu ft at 77 F
μ_l = 1.87×10^{-5} lb-sec per sq ft at 77 F

Reynolds modulus for liquid considered as a single phase (Re_l) = 73,500

Equation [17] may be utilized since both phases are flowing turbulently.

(*b*) *Calculation of pressure drop due to air alone* = $(\Delta P/\Delta L)_g$:

The usual calculation reveals that the pressure drop resulting from the flow of 0.50 lb per sec of air at 77 F, 100 psia, in a 2-in. pipe is 1.03 psi per 100 ft of pipe. (For the flow of the gas in commercial pipe at Re_g = 295,000 the friction factor (3) is equal to 0.018).

(*c*) *Calculation of* χ_{tt}:

$$\chi_{tt} = \left(\frac{\mu_l}{\mu_g}\right)^{0.111} \left(\frac{\gamma_g}{\gamma_l}\right)^{0.555} \left(\frac{W_l}{W_g}\right) = 1.28$$

(*d*) *Calculation of* Φ_{tt}:

$$\sqrt{\chi_{tt}} = 1.13$$

thus

$$\Phi_{tt} = 4.6 \text{ (Fig. 11)}$$

(*e*) *Calculation of* $(\Delta P/\Delta L)_{T-P}$:

$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = (4.6)^2\,(1.03) = 21.8$ psi per 100 ft of pipe, which is about 21 times the pressure drop for the air alone and 14 times the pressure drop for the water alone. (For the flow of the liquid in commercial pipe at Re_l = 73,500, the friction factor, reference 3, is equal to 0.0232.)

Example 2

It is desired to estimate the pressure drop in 100 ft of 2-in. pipe in which air is flowing at a rate of 0.50 lb per sec, and an oil (whose properties will be given) is flowing with a rate of 0.30 lb per sec. The temperature of the fluids is 77 F and the average line pressure is 100 psia.

(*a*) *Determination of properties and type of flow:*

W_g = 0.50 lb per sec
γ_g = 0.502 lb per cu ft (at 100 psia and 77 F)
μ_g = 0.388×10^{-6} lb-sec per sq ft (at 100 psia and 77 F)

Reynolds modulus for gas considered as a single phase (Re_g) = 295,000

W_l = 0.30 lb per sec
γ_l = 55 lb per cu ft (77 F)
μ_l = 600×10^{-5} lb-sec per sq ft (77 F)

Reynolds modulus for oil considered as a single phase filling the pipe (Re_l) = 11.5.

Since the liquid is in viscous motion and the gas in turbulent motion, Equation [30] must be utilized.

(*b*) *Calculation of* χ_{vt}:

$$\chi_{vt} = \frac{C_l}{C_g}\,(Re_g)^{-0.8}\,\frac{\mu_l}{\mu_g}\cdot\frac{\gamma_g}{\gamma_l}\cdot\frac{W_l}{W_g}$$

Take C_l = 64 (see text for discussion)
C_g = 0.216 (for commercial pipe (3), the constant C_g is greater than 0.184 which applies only to smooth pipes)

Then χ_{vt} = 1.07

(*c*) *Evaluation of* Φ_{vt}:

$$\sqrt{\chi_{vt}} = 1.03$$

Thus

$$\Phi_{vt} = 3.9 \text{ (Fig. 15)}$$

(*d*) *Calculation of* $(\Delta P/\Delta L)_{T-P}$:

$$\left(\frac{\Delta P}{\Delta L}\right)_{T-P} = (3.9)^2\,(1.03) = 15.7 \text{ psi per 100 ft}$$

Thus, the pressure drop for the two-phase case is about 15 times the pressure drop of the air alone and 15 times the pressure drop due to the oil flowing alone. Comparison of examples 1 and 2 reveals that 0.30 lb per sec of the oil had almost as much effect on the pressure drop as 6 lb per sec of water.

Example 3

It is desired to predict the pressure as a function of distance along a horizontal tube in which evaporation is occurring. If it is assumed that at each section of the tube, the steady-state results discussed in the paper are applicable, the method of procedure is as follows:

(*a*) *Determination of amount of vapor present at any point in tube:*

From a heat balance on an element of tube of length dZ

$$f_c(\pi D_p dZ)\Delta t = 3600r\, dW_g \ldots\ldots [1a]$$

Thus

$$W_g = \int_0^Z \frac{\pi f_c D_p \Delta t}{3600r}\, dZ = F_1(Z) \ldots\ldots [1b]$$

Where $F_1(Z)$ represents the pounds of vapor per second flowing at any point Z feet from the entrance to the heating section.

(*b*) *Ratio of W_l/W_g:*

Since the total fluid rate is constant at any point along the tube

$$W_T = W_l + W_g$$

Thus

$$\frac{W_l}{W_g} = \left(\frac{W_T}{F_1(Z)} - 1\right) \ldots\ldots [1c]$$

(*c*) *Pressure gradient of any point Z due to vapor flow alone:*

$$\left(\frac{dP}{dZ}\right)_g = \frac{1}{2}\left(\frac{\pi}{4}\right)^{-1.8} \frac{C_g \mu_g^{0.2} W_g^{1.8}}{g^{0.8}\gamma_g D_p^{4.8}} \ldots\ldots [1d]$$

$$= F_2(Z)W_g^{1.8} = F_2(Z)[F_1(Z)]^{1.8} = F_3(Z) \ldots\ldots [1e]$$

(*d*) *Pressure gradient with two-phase flow:*

From Equation [17] (assuming turbulent flow of both phases)

$$\left(\frac{dP}{dZ}\right)_{T-P} = F_3(Z)\left[1 + \alpha^{1/4}\left(\frac{\mu_l}{\mu_g}\right)^{0.083}\left(\frac{\gamma_g}{\gamma_l}\right)^{0.416}\left(\frac{W_T}{F_1(Z)} - 1\right)^{0.75}\right]^{2.4}$$

$$= F_3(Z)\left[1 + F_4(Z)\left(\frac{W_T}{F_1(Z)} - 1\right)^{0.75}\right]^{2.4} \ldots\ldots [1f]$$

(*e*) *Pressure at any point:*

Integrating Equation [1*f*]

$$P = \int_0^Z F_3(Z)\left[1 + F_4(Z)\left(\frac{W_T}{F_1(Z)} - 1\right)^{0.75}\right]^{2.4} dZ \ldots [1g]$$

The graphical evaluation of Equation [1*g*] is tedious, but not difficult.

Appendix 2

The measured pressure drops for two-phase flow were corrected approximately to standard conditions by the following equation

$$\left(\frac{\Delta P}{\Delta L}\right)_{\text{corr}} = \left(\frac{\Delta P}{\Delta L}\right)_{\text{meas}}\left[1 + \frac{\left(\frac{\Delta P}{\Delta L}\right)_{\text{meas}} - \left(\frac{\Delta P}{\Delta L}\right)_l}{\left(\frac{\Delta P}{\Delta L}\right)_{\text{meas}}}\left(\frac{\gamma_m}{\gamma_{\text{std}}} - 1\right)\right]$$

where

$\left(\frac{\Delta P}{\Delta L}\right)_{\text{meas}}$ = measured two-phase pressure drop for W_l lb per sec liquid, and W_g lb per sec gas

$\left(\frac{\Delta P}{\Delta L}\right)_{\text{corr}}$ = two-phase pressure drop corrected to "standard" conditions

$\left(\frac{\Delta P}{\Delta L}\right)_l$ = pressure drop due to W_l lb per sec of liquid flowing alone

γ_m = density of air at conditions of test

γ_{std} = "Standard" air density (66 F and 1 atm)

The points shown on Figs. 2 to 8, inclusive, were corrected to standard conditions by this approximate method.

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The Differential Shrinkage of Wood

By W. L. GREENHILL,[1] SOUTH MELBOURNE, AUSTRALIA

This paper was prepared originally as discussion of a paper by A. J. Stamm and W. K. Loughborough (1),[2] but because of war conditions was not received until some months after publication of that paper and discussion. The results of shrinkage measurements made with the aid of the microscope on wood samples, consisting of thin radial sections cut entirely from rays, are of such interest that the Wood Industries Division considered their publication advisable. A discussion of the subject paper is also presented by A. J. Stamm and W. K. Loughborough.

The fact that wood shrinks less in the radial than in the tangential direction is normally attributed to the restraining effect of the medullary rays. However, Stamm and Loughborough (1), mainly from the results of investigations by Ritter and Mitchell (2), state: "There can be no restraining action on the shrinking and swelling of wood in the radial direction by the ray cells." The work referred to consisted of a study of single cells of basswood holocellulose by polarized light, when it was found that the ray cells behaved differently from the fibers. The ray cells showed the same coloration as the fibers when their axes were perpendicular to the fibers, whereas when parallel one type of cell showed blue and the other yellow. It was concluded from these observations and from actual shrinkage measurements on isolated ray cells that rays have a large axial and small transverse shrinkage.

This explanation has already been challenged by Barkas (3), who points out that, in the examination of cells with polarized light, the preponderance of thickness determines the color in the microscope, whereas the strength of the fibrils in the different directions determines the swelling. In other words, the spirally wound fibrils in the primary walls of the ray cells may consist of a very thin layer which does not show in the microscope but which may exert considerable restraint on the lateral expansion of the secondary fibrils.

The use of shrinkage measurements on isolated fibers would in itself appear of doubtful value when applying the results to groups of fibers. To overcome this objection, shrinkage measurements have recently been made by the author on thin radial sections cut entirely from rays. In size these were up to $^1/_2$ in. square and measurements were made with a microscope. Normal tangential and radial shrinkage values were determined for the same wood samples from thin cross sections. The results obtained are given in Table 1.

TABLE 1 SHRINKAGE MEASUREMENTS ON THIN RADIAL SECTIONS

(Soaked to oven-dry; per cent soaked dimensions)

Species	Tangential[a]	Radial[a]	Longitudinal[a]
Casuarina luehmanni:			
Normal cross section	11.9	3.3	0.14
Rays	...	1.2	4.8
Grevillea robusta:			
Normal cross section	9.5	3.7	0.08
Rays	...	1.2	5.7
Xylomelum pyriforme:			
Normal cross section	8.0	2.0	0.17
Rays	...	0.7	1.8
Quercus sp:			
Normal cross section	9.2	4.9	0.32
Rays	...	3.2	3.5

[a] Actual directions in tree in all cases.

These results completely substantiate those of Clarke (4), and justify his conclusion that the axial shrinkage of the rays is less than the radial shrinkage of the wood as a whole, but that their transverse shrinkage is much greater than the longitudinal shrinkage of the wood as a whole. The evidence available thus appears to throw considerable doubt on Ritter's and Mitchell's conclusions. On the other hand, although the shrinkage of the rays axially is almost certainly less than the radial shrinkage of the wood as a whole, and therefore that the rays exert a restraining effect is indisputable, there is also some indication that, even without the restraining effect of the rays, radial shrinkage would still be less than tangential. To demonstrate this experimentally, shrinkage measurements have been made on small samples of three *Casuarina* species, some samples containing a normal proportion of the prominent rays in these species and some being prepared free from all but uniseriate rays. The results obtained, which are given in Table 2 show that by eliminating the large rays there is a definite increase in the radial shrinkage, but on the other hand an even greater increase would have been expected had the rays been solely responsible for the difference between the normal tangential and radial shrinkages.

TABLE 2 SHRINKAGE MEASUREMENT ON CASUARINA SPECIES

(Soaked to oven-dry; per cent soaked dimensions)

Species	Tangential	Radial
Casuarina cunninghamii:		
Normal	10.6	2.6
Free from large rays	10.8	3.9
Casuarina torulosa:		
Normal	9.0	2.0
Free from large rays	9.1	5.0
Casuarina luehmanni:		
Normal	11.9	3.3
Free from large rays	12.0	4.3

Possible factors in addition to the ray cells which may be responsible for the differential shrinkage are as follows:

1 The presence of a greater number of pits on the radial than the tangential faces of the fibers, resulting in the fibrils on the radial faces being less perfectly oriented than on the tangential faces. This explanation, according to Stamm and Loughborough (1) is due to Ritter and Mitchell (2), although nothing to this effect can be found in the reference given.

2 The effect of bands of wood of varying density, such as early and late wood or wide bands of parenchyma adjacent to bands of normal wood fibers. This explanation is suggested by Frey-Wyssling (5).

Collapse

An explanation of the excessive and irregular shrinkage known as collapse was first given by Tiemann (6). His "liquid-tension" theory has been generally accepted and satisfactorily explains the phenomenon and its effects, although some criticism of the theory has been made from time to time. The alternative "stress" theory propounded by Stamm and Loughborough certainly cannot be accepted without question as the main cause of collapse, but on the other hand it is quite probable that drying stresses assist in producing collapse. Some of the evidence which

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[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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favors the liquid-tension theory and which appears to have been overlooked by Stamm and Loughborough is as follows:

1 The fact that the presence of air prevents the occurrence of collapse, according to the tension theory, can be used to demonstrate the validity of this theory. Using two matched samples of green timber of a species subject to collapse, such as mountain ash (*Eucalyptus regnans*), carefully air-dry one of the samples so that the "free" water is removed from the cells. This sample should then be thoroughly soaked until its moisture content is at least as high as that of the green sample which, incidentally, should be kept green meanwhile. Both samples should now be dried at a temperature of about 140 F and a relative humidity of 80 per cent, conditions which favor collapse. It will be found that collapse occurs to a much greater extent in the green than in the resoaked piece. The reason for this is that in the soaking after drying, the entrapped air in the cell cavities is removed completely only with great difficulty so that very few of the cells become liable to collapse. In the original green condition a considerable number of the cells may, quite conceivably, be completely filled with water. Furthermore, it is found that any trees of such species as mountain ash, with an unusually high green-moisture content are particularly liable to collapse.

2 The statement by Stamm and Loughborough (1) that collapse is not randomly distributed throughout the wood but is located in the central zone under compression shows a lack of knowledge of even the appearance of timbers which collapse. Indeed one of the most usual features of collapse is its extremely irregular nature. The effect of the collapse of a cell is not, of course, shown in an isolated cell, as it is not possible without disruption of the fibers, for one cell to collapse without distorting those cells surrounding it. However, the most logical explanation of the variation in the degree of collapse in different portions of a board is surely the random distribution of areas of collapse-producing cells.

3 By cutting successive cross sections of a piece of wood of a species such as mountain ash, starting with sections about $^1/_4$ in. thick and gradually reducing the thickness it will be found that on drying the sections the tangential and radial shrinkages gradually decrease with decrease in thickness. However, this does not continue indefinitely and for thicknesses below about 0.05 in., the shrinkage is constant. Tests of this nature, made on various species, have been reported previously (7), and it has been shown that the thickness at which the shrinkage becomes constant bears a fairly constant ratio (1.6–1.8) to the fiber length. At such a thickness the great majority of the fibers would be cut at one end, evaporation of the free water could then proceed from this end, and internal tensions would not be possible.

A similar effect suggested by Tiemann can be observed by cutting a wedge-shaped sample of green wood tapering from a fine edge to 5 or 6 times a fiber length in the grain direction. On drying such a sample under conditions producing collapse, it will be found that the thin end shrinks uniformly, but where the thickness has increased to about $1^1/_2$ fiber lengths, the shrinkage becomes much greater and irregular in nature. No convincing explanation of this behavior seems possible from a consideration merely of simple drying stresses.

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Discussion

A. J. Stamm[3] and W. K. Loughborough.[4] Ritter and Mitchell (2),[5] as a result of their optical and swelling measurements, concluded that "they (the ray cells) at least have a less restraining effect on radial swelling of wood than was previously believed." The author's data lead to the same conclusion. The matter of controversy thus rests on the degree to which ray cells can restrain radial swelling.

The author's data for the swelling of sections made up entirely of rays, show a swelling in the radial direction of the tree ranging from one third to two thirds of the swelling of the wood as a whole in the same direction. Ritter's and Mitchell's data (2), on the other hand, show a smaller difference between the swelling of individual ray cells and the wood as a whole in the same direction. These differences may be due to species differences, differences between measurements on single ray-cell fibers and ray-cell bundles, or differences in technique. In any case, the author's data indicate that the ray cells swell quite differently in their longitudinal direction (radial direction in the tree) from the longitudinal swelling of tracheids (10 times as much). This difference is explained by Ritter's and Mitchell's optical measurements.

Ritter and Mitchell recognize the fact that thickness of section in their optical measurements does affect the color to some extent. They have, however, taken this into consideration in drawing their conclusions. Their explanation of the retardation of radial swelling of wood by the pits did not, as the author states, appear in the article referred to. It appears elsewhere.[6]

The effect of pits on the swelling and shrinking will explain the differences obtained by the author between the radial swelling of normal wood and wood free from large rays. It is known that tracheids tend to end chiefly at junctions with ray cells and that the pits are more highly concentrated near the ends of tracheids. The tracheids of sections relatively free from ray cells will thus tend to have less pits on their surfaces and, as a result, will show a smaller swelling restraint along their radial walls.

Hence the writers feel that, on the basis of both the author's data and those of Ritter and Mitchell, the old concept of the ray cells furnishing the major restraint to radial swelling is not substantiated.

Collapse. Here again the controversy rests on a matter of differences in the extent to which two different mechanisms are effective in causing a certain phenomenon, rather than a complete difference in the mechanism. We believe that collapse in some cases is due to liquid tension, but in the case of North American

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[5] Numbers in parentheses refer to the Bibliography at the end of the paper.

[6] "Structural Factors Contributing to the Directional Shrinking and Swelling of Wood," by G. J. Ritter and R. L. Mitchell, unpublished report of the Forest Products Laboratory, U. S Department of Agriculture, March, 1941.

woods, with which we are familiar, we feel that collapse is more frequently due to stresses than to a liquid tension. We include in our definition of collapse all cases where the shrinkage is abnormally high in contrast to some who consider that collapse has taken place only when groups of fiber cavities have completely caved in. Part of the difference of opinion as to the importance of the two forms of collapse rests on these two different definitions.

The author cites an experiment in which wood subject to collapse is carefully air-dried to remove free water and let air into the structure. Free water, however, cannot be removed from fibers which meet the requirements for liquid-tension collapse without causing the fibers to collapse. If air is admitted without collapse occurring, it is due to the fact that some air existed in the fiber cavities. In either case, stresses are set up during the initial drying. Decreased shrinkage after resoaking and again drying could be due either to a decrease in liquid-tension collapse, as maintained by the author, or to a decrease in stress collapse, as described in the writer's paper (1).

Experiments have been made at the Forest Products Laboratory in which sodium carbonate is allowed to diffuse into green blocks of wood, followed by the diffusion of acid into the structure to liberate carbon dioxide. This pretreatment did not reduce collapse when the block was dried under collapsing conditions. On the basis of the liquid-tension theory, the presence of gas in the cells should have eliminated collapse.

The writers have also made measurements on thin cross sections of wood. Under favorable drying conditions, it is true that collapse is decreased, but it is not eliminated as it would be if it were entirely due to liquid tension. The fact that the amount of collapse occurring in such sections can be varied appreciably under different drying conditions, makes us feel that it is largely due to stresses even in such sections. The existence of stresses in these thin sections can be observed by the way free water is first removed around the edges, giving a circular wet zone on the surface of each of the faces.

Collapse of the readily visible extreme type, we appreciate, may be somewhat randomly distributed, due to conditions for liquid-tension collapse, or due to groups of cell walls which have abnormally low compressive strengths. From our experience, however, the general decrease in fiber-cavity sizes seems to be chiefly concentrated in compression zones.

The writers wish to emphasize that the chief object of the latter part of their paper (1) was to point out that all collapse is not due to liquid tension, as so many believe, and that it can be logically explained on the basis of stresses. It is believed that the stress theory is more generally applicable than the liquid-tension theory.

Advanced-Training Plane, the Fairchild AT-21

Problems of Construction and Alternate Substitutions in Wood Aircraft

By J. M. Stevens,[1] Hagerstown, Md.

Some of the problems coincident with the greatly expanded use of wood and plywood in aircraft construction are discussed in this paper. A greater use of certain approved alternate wood species in place of spruce and other critical aircraft materials is advocated. Several specific examples of what can be accomplished in the way of simplifying and standardizing the construction of flat and molded plywood constructions are presented.

With the advent of the present world conflict the demands made upon wood and plywood for use in aircraft manufacture have increased tremendously. As a result, these materials, first considered as noncritical, can now be considered only as less critical. In some instances they cannot be obtained at all in the quality and quantity required for our strategic offensive effort.

The magnitude of the demands that are being made upon the aircraft industry can be more readily understood when we recall President Roosevelt's speech made in the spring of 1940, in which he announced that the United States was going to produce 50,000 airplanes a year. This figure was subsequently boosted to 125,000 a year. It was soon apparent that our aluminum supply was not adequate to meet these requirements, so immediate plans were made for the expansion of our aluminum-producing facilities. In the meantime, manufacturers of light aircraft were requested to use nonstrategic materials wherever possible. As a result wood, in either laminated or plywood form rapidly gained a position of paramount importance in aircraft manufacture.

Wood Applied Successfully to Training Planes

One of the outstanding examples of the successful use of wood in aircraft construction is in the primary-training field, a typical example of which is the Fairchild designed PT-19.

The wings are constructed of two built-up spars with thin face-plating, and the ribs are constructed of square spruce capstrips reinforced by three-ply mahogany plywood gussets at all corners and joints. The outer wing covering is made of two sections of $^3/_{32}$-in. mahogany plywood with poplar core. This three-ply plywood is preformed to conform with the shape of the leading edge, and the bottom is flat, the two sections being joined at the trailing edge.

At present the rear cowling deck, formerly made of metal, is constructed of light veneers bonded by the Duramold process. Covers for the gas tanks on the underside of the center section formerly of aluminum alloy are also now made of wood. The

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flaps which on the first production models were made of aluminum alloy, are now constructed of plywood.

These PT-19's, have withstood hundreds of thousands of hours of grueling flight-training work under many climatic conditions. In most cases hangar facilities are not sufficient to house all these aircraft every night, and a large percentage must be left to the mercy of the elements. Even under such extreme conditions, very encouraging service reports have been obtained.

The Fairchild AT-21, a two-engined trainer announced recently, is another example of how wood is being used successfully to meet the expanding training requirements of the AAF. The Beech AT-10 is similar in size while the Curtiss C-76 cargo plane, contract now canceled, was to have been many times larger.

It can be readily seen that as the size of wooden airplanes is increased the quantities of wood required to construct them is also proportionately increased. For example, 220 fbm of spruce lumber are required to construct the PT-19 airplane, whereas 1260 fbm are required to construct the AT-21 airplane. On the plywood side of the bill of material, a comparison shows 850 sq ft of plywood used for the PT-19, 3000 sq ft for the AT-21, while an estimated 12,000 sq ft would have been required for the C-76 cargo. When volume is taken into consideration, these plywood differences may become even greater. In 1942, the aircraft industry used 6,000,000,000 fbm of aircraft plywood. In view of the trend to more and larger types of wood aircraft, it can be expected that in 1943 the production of aircraft-grade wood and plywood will have been doubled in order to meet production schedules.

Coincident with this rapid expansion in number and size of wooden airplanes has been the problem of procuring material in sufficient quantity that would meet AN specifications. Supposedly inexhaustible supplies of aircraft woods simply were not available. The chief contributing factor to this situation is that only the very best quality of wood can be used in aircraft construction. This limits the quantity suitable for aircraft use to a very small percentage of actual production. In addition, there is a shortage of precision milling equipment that can cut to the varied and peculiar demands of the aircraft industry; service demands plus the lure of better paying defense jobs have created a shortage of skilled woods labor; there are not sufficient lumber kilns and veneer driers to maintain rapid delivery schedules. Aircraft grades are practically always cut to special order and rarely, if ever, are carried in stock by producing sawmills. This is due to the highly specialized nature of such grades and the wide ranges of sizes and lengths required by various aircraft manufacturers. Also, there has been a lack of specific information on potential sources of supply for these aircraft grades of lumber and veneer.

Wood Fabrication Problems

In 1941, the production of Douglas fir lumber was 8,000,000,000 ft or a weekly average production of 150,000,000 ft; 1942 production was the same. In contrast, the 1941 production of Sitka spruce lumber was only 1,300,000,000 or a weekly average of 25,000,000 ft which by 1942, had dropped to 7,000,000 ft, a decrease of over 350 per cent. Further, according to the West Coast Lumberman's Association, not over 1 per cent of fir is going into aircraft production at the present time, but probably another 3 or 4 per cent now going into pontoon lumber, ship decking, and so on, would meet aircraft specifications. The Forest Product's Laboratory estimates that only 2 per cent of fir lumber and 5 to 6 per cent of spruce will meet aircraft specifications.

TABLE 1 FIR AND SPRUCE AVAILABLE FOR AIRCRAFT PRODUCTION

	Average weekly production, fbm	Aircraft grade per cent	Average weekly amount available aircraft fbm	Rating (spruce 100)
Spruce..........	7,000,000	5	350,000	100
Douglas fir......	150,000,000	2	3,000,000	860

Taking 2 per cent of the present production of fir and 5 per cent of the present production of spruce as representative averages of the foregoing data, the amounts available for aircraft use are given in Table 1.

Because spruce combines the qualities of lightness, great strength, and stiffness per unit of weight, and also a considerable degree of toughness, it has always been of great value in the manufacture of wood airplanes where just such qualities are demanded. It is now chiefly used in the construction of spars for wings, bulkheads and stringers for fuselages, and so on, which parts are usually made of solid or laminated spruce in preference to other species.

Faced with a declining spruce production, it became apparent as early as 1940 that there would not be sufficient high-quality aircraft spruce to meet the requirements of an expanding aircraft industry. At that time three possible solutions to the problem were investigated by the Civil Aeronautics Authority, as follows:

1 A possible broadening of the interpretation of current spruce specifications by establishing a mechanical method of judging the quality of the basic material and a more intelligent judging of the weakening effects of surface defects.

2 The use of varying thickness and short-length material through laminating and splicing.

3 The use of substitute material.

That the first method is still a subject of much controversy is evidenced from the report of a meeting held at Longview, Wash., February 26, 1943, and at Chicago, Ill., October 13, 1943, to discuss the problems involved in the inspection of aircraft lumber. At this meeting were representatives of the War Production Board, West Coast Lumberman's Association, West Coast Bureau of Lumber Grades and Inspection, Pacific Lumber Inspection Bureau, Army Air Force, U. S. Dept. of Agriculture, manufacturers of aircraft lumber, and interested aircraft manufacturers. (A discussion of their findings is not considered pertinent at this time since this paper is chiefly concerned with the last method.)

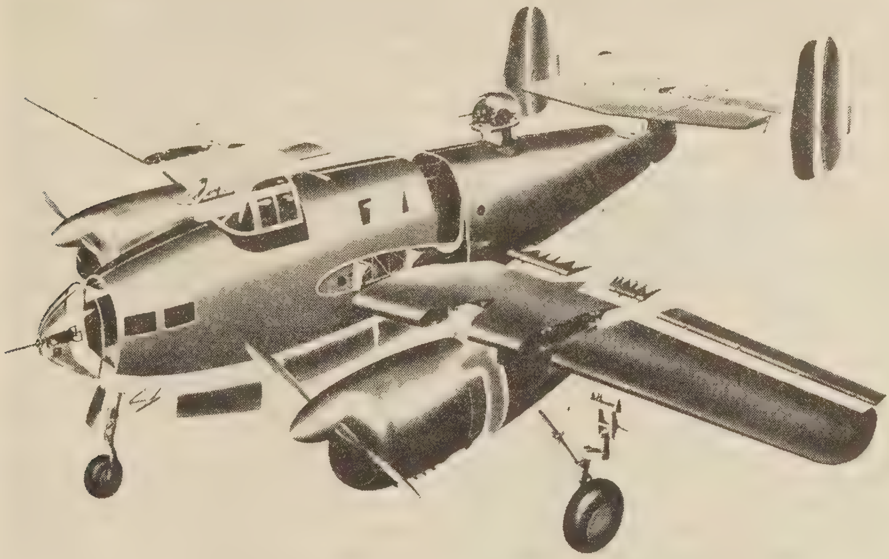
The second method, permitting the use of short lengths and narrow widths, has been in effect with most companies for some time; for example, laminations are being made of pieces edge-glued together to obtain a given width with the restriction that edge joints in adjacent laminations shall be not closer than the thickness of the lamination. Short lengths are joined by scarf joints provided that the slope is not steeper than 1 in 15 for softwoods or 1 in 15 for hardwoods.

Authority Given to Use Alternate Woods for Spruce

By so doing, the available supply of aircraft-grade spruce was greatly increased. However, a critical shortage still appeared imminent, so in June, 1942, wood-aircraft manufacturers were advised that certain alternate woods could be used in place of spruce without change in design. The alternate woods suggested were western hemlock, noble fir, and yellow poplar.

It was soon discovered that the lumber industry was not prepared to supply these alternate woods in the quality called for in AN specifications. Some of the reasons for this situation have already been outlined. The following month (July, 1942) Port Orford cedar, Douglas fir, Southern cypress, Alaska cedar, and Norway pine were also given approval as direct substitutes for spruce without change in design. However, spruce continued to be available, whereas the alternates were not.

In September, 1942, the use of Douglas fir was again recommended specifying normal Douglas fir with a specific gravity range of not less than 0.45. This was followed by an announce-



THE FUSELAGE SHELL, WINGS, TAIL SURFACES, AND MANY OTHER PARTS OF THE AT-21 ARE OF PLYWOOD CONSTRUCTION

ment, made in January, 1943, of a new classification of Douglas fir known as light Douglas fir, having a specific gravity of 0.38 to 0.47. As this latter class requires special selection and is therefore limited in quantity, its use has been restricted to certain types of wood aircraft. Latest information indicates that sufficient quantities of aircraft spruce, noble fir, western hemlock, and yellow poplar will be available to meet scheduled production requirements.

Thus it can be seen that, with the decline in the supply of aircraft spruce, the problem of what woods can or cannot be used as alternate materials has assumed ever greater importance. This is particularly true in the case of subcontractors, most of whom are having their first experience in aircraft construction and are frequently experiencing considerable difficulty in obtaining specified materials. In an attempt to clarify this somewhat confused situation a "Wood and Glue Alternate and Substitution Chart" was prepared by the author's company.

On this chart are listed all woods commonly used in aircraft construction. In one section are listed all alternates that may, without specific engineering approval, be used as replacements for the material specified on the drawing. In the second section is another group, classed as "substitutes," no one of which may be used as a replacement for the material specified on the drawing without specific engineering approval. Similar procedure has been adopted to aid in the proper selection and use of synthetic-resin adhesives. The current specifications are listed in order to provide an exact means of quality control.

In the preparation of this chart weight or specific gravity was the most important controlling factor in making each choice. Ease of working was also given consideration. As the availability of a given material cannot readily be determined and may change from day to day, this aspect was not considered pertinent in the analysis.

Weight is extremely important in all aircraft construction. For example, a complete change from spruce to western hemlock without any change in design would increase the weight of the PT-19 airplane by approximately 21 lb and the AT-21 airplane by approximately 92 lb. This is not overly serious. However, a complete change to Douglas fir (normal type) would increase the weights of these two airplanes by 70 and 294 lb, respectively. For this reason, Douglas fir cannot be considered as a desirable substitute for spruce without changes in design. Once an airplane is in production, any change in design is costly and time-consuming and should be avoided wherever possible.

PLYWOOD CONSTRUCTION PROBLEMS

Until quite recently, only a few woods, notably mahogany and poplar, were considered to be entirely satisfactory for flat and molded aircraft plywood constructions. Birch veneer has been used extensively for highly stressed plywood parts and for parts subject to excessive abrasion or wear. Other woods commonly but not extensively used for aircraft veneer are spruce, sweet gum, walnut, Douglas fir, and basswood.

The problems of veneer procurement and large molded-shell manufacture have long been complicated by the use of veneers under $^1/_{32}$ in. in thickness. Very flat veneers are required in the molding processes to permit accurate edge-fitting and fast laying of the veneer strips that are used to make up the large fuselage shells. Both gapping and overlapping which result from poor fitting subject these shells to extensive rework and occasionally to rejection. When veneers intended for layup into shells of simple curvature become buckled they cannot be joined and spliced successfully in the Diehl machines made for this purpose.

When veneer is cut and dried to $^1/_{48}$-in. thickness, only approximately 10 per cent of the total mahogany aircraft-veneer output, 7 per cent of the sliced-poplar aircraft-veneer output, and 3 per cent of the birch aircraft-veneer output are of suitable flatness for Duramold work. Use of any veneer other than flat stock results, as noted, in rework and rejection coupled with an increase in layup time ranging from 50 to 100 per cent. By increasing the cut thickness to $^1/_{32}$ in., it was found that approximately 25 per cent of the dried aircraft mahogany veneer, 20 per cent of the sliced-poplar veneer, and 10 per cent of the birch veneer could be efficiently utilized for Duramold work. This increase results from the following: (1) the thicker veneer, being stiffer, will remain flat when subjected to internal stresses developed during cutting and drying that would buckle a thinner veneer; (2) most modern veneer driers are so constructed that

they will flat-dry good veneer down to $^1/_{32}$ in. thick at a reasonable percentage of full capacity; full capacity being achieved when handling $^1/_{28}$-in. or thicker veneers. In order to get comparable results with $^1/_{48}$ in.-thick veneer, it is necessary to cut the drier capacity to 25 to 30 per cent of its total. In these times, veneer companies are unwilling to do this for even a 50 per cent premium.

Another factor, apart from that of thickness and flatness, complicating the present veneer situation, is the rapidity with which the various kinds of veneer fluctuate from noncritical to critical and back again. Several months ago mahogany appeared to be scarce, poplar and birch abundant. Today mahogany appears to be relatively plentiful, birch seems practically nonexistent, and poplar is very scarce. All veneer and plywood manufacturers are experiencing these difficulties, so these shortages can in no sense be considered as merely local.

Douglas fir and red gum provide two specific examples of procurement difficulties coincident with the manufacture of aircraft plywood. Douglas fir is only acceptable when quarter-sliced because vertical-grain fir veneer has more uniform strength properties than does rotary-cut veneer. Also, $^1/_{16}$ in. is about the usual minimum thickness that can be rotary-cut. As only three mills in the Pacific Northwest are equipped with slicing equipment, logs intended for aircraft veneer have to be shipped to the Midwest for processing. Red-gum veneer can be either sliced or rotary-cut but here again the rotary lathes located at the source of supply cannot cut aircraft veneers in quantity to the tolerances allowed, so it also becomes necessary to process these logs in the Midwest.

A third problem in the use of plywood has been, in the past, the lack of standardization of specifications. An example in point was the original Army and Navy plywood specification AN-NN-P-511 which did not specify the exact thicknesses of veneers that should be used in any given flat-plywood construction. This has since been corrected by revision "b" of this specification which calls out the exact thickness of faces, cross-bands, and cores for all standard aircraft plywood. These sizes are shown as decimal equivalents based on the kiln-dry-veneer thickness and will run slightly less than those given in the Wood Aircraft Fabrication Manual which are based on the green-veneer thickness.

Advantages of Standardization

To comply with a recent request of the Army Air Forces, who are attempting to increase production of plywood an estimated 80 per cent by standardization of veneer thicknesses and plywood constructions, and also to satisfy numerous and repeated requests from production departments, a detailed study was made of the plywood requirements of the AT-21 airplane with a view toward simplification of construction and species requirements wherever structurally possible. Such a study was made and the data prepared in chart form. Similar plywood charts have been prepared for other airplanes manufactured by the author's company.

In this chart, the plywood-construction preferences listed are based upon the following factors:

1 Minimum use of veneers under $^1/_{32}$ in. thick.
2 Adherence to standard veneer thicknesses.
3 Proper balance of finished plywood.
4 Availability of material.
5 Application to which material is put.
6 The use of over-all thicknesses and constructions that would give approximately the same strength as the original.

All-mahogany constructions have been given first preference in this chart. This reflects the relative ease of working and, apparently, of procuring this kind of veneer at this time as contrasted with the present scarcity of poplar. To offset to some extent the fluctuating supply of poplar, and to supplement the approved list of substitutes for poplar, it was recommended, wherever feasible, that the all-mahogany constructions be given preference over the mahogany-poplar constructions.

A detailed study, substantiated by test data on the fuselage assemblies, indicated that these simplified plywood constructions would offer the following advantages:

1 A net over-all decrease in weight on the airplane ranging from 9 to 19 lb, without any sacrifice in strength at highly stressed areas.

2 The procurement of satisfactory veneer would be simplified and use of critical material substantially reduced.

3. The time of layup and amount of glue (Tego film) used per unit would be materially reduced.

4 Production time per unit would be correspondingly decreased, thereby permitting the manufacture of more units than is now possible.

On this chart, in addition to the simplified plywood constructions, there is also an alternate and substitution chart for veneers. This is made up in the same manner as the wood chart previously described.

To check the comparative strengths of all the recommended alternate constructions, a series of tests was conducted of the various plywoods considered.

The results which follow are based upon comparative tests made from sample test panels constructed on a flat die by the bag-molding method. Process conditions were identical with those used in the manufacture of full-size shells. All veneers of a given thickness were taken from the same flitch in order that the variables inherent in tests of a nonhomogeneous material such as wood might be reduced as much as possible. It can be assumed therefore that the test results shown are reasonably indicative of what could be expected from any given Duramold shell.

Scope and Method of Testing Plywood

Tension Tests. Tests were made to determine the tensile strength of each plywood construction both parallel and perpendicular to the grain of the faces. Specimens 1 × 5 in. were used, with the center portion trimmed down to $^1/_2$ in. diam (radius $4^3/_4$ in.). The specimens were held by ordinary flat grips and tested in direct tension to rupture in a Riehle plywood-testing machine.

Shear Tests. Shear tests were made in the standard picture-frame-type shear-test fixture used by the Forest Products Laboratory. In all tests, face grain direction was either parallel (0 deg) or perpendicular (90 deg) to the direction of load.

Bending Tests. Bending tests, using strips of plywood 1 × 12 in. long, were used to obtain the modulus of rupture and the modulus of elasticity. Each plywood construction was tested in cross-bending as a simple beam. Bolts of $^1/_4$ in. diam mounted in roller bearings were used as end supports. This eliminated the effect of friction as the specimen bowed under load. Loads were applied by adding shot, in $2^1/_2$-lb increments, to a small container suspended from the center of the test specimen. Deflections at each increment of load were obtained from a Starrett gage, mounted directly over the point of application of load. Face-grain direction was either parallel (0 deg) or perpendicular (90 deg) to the long axis of the test specimen.

Summary of Test Data

Tables 2 and 3 summarize details of the test results. In these tables strengths have been averaged on both a pounds-per-square-inch and a total-load basis. The percentage comparison

TABLE 2 FRONT FUSELAGE; STRENGTH AND ELASTIC PROPERTIES

Plywood construction	M.C. at test, per cent	Ultimate strength in tension 0°	Ultimate strength in tension 90°	Ultimate strength in shear 0°	Ultimate strength in shear 90°	Static bending M R 0°	Static bending M R 90°	Static bending M E (× 1000) 0°	Static bending M E (× 1000) 90°
(a) Comparison on Basis of Strength, psi									
Present 9 ply	3.4	6850 (11)	6960 (12)	3560 (1)	2915 (1)	8520 (3)	4750 (3)	980 (3)	580 (3)
Revision (1) 7 ply	5.0	6770 (10)	6380 (10)	3415 (1)	3070 (1)	8870 (3)	5800 (3)	1042 (3)	537 (3)
Revision (2) 7 ply	3.5	6300 (11)	6350 (11)	3310 (1)	2880 (1)	8490 (3)	6430 (3)	1050 (3)	611 (3)
(b) Comparison of Actual Test Values,[a] lb									
Present 9 ply	3.4	740	740	8250	6750	29.61#	18.63#	...	.
Revision (1) 7 ply	5.0	760–103%	720–97%	8500–103%	7650–113%	35.89–121%	23.43–140%	...	
Revision (2) 7 ply	3.5	722–97%	720–97%	8575–104%	7470–111%	36.15–122%	27.38–162%	...	

TABLE 3 REAR FUSELAGE; STRENGTH AND ELASTIC PROPERTIES

Plywood construction	M.C. at test, per cent	Ultimate strength in tension 0°	Ultimate strength in tension 90°	Ultimate strength in shear 0°	Ultimate strength in shear 90°	Static bending M R 0°	Static bending M R 90°	Static bending M E (× 1000) 0°	Static bending M E (× 1000) 90°
(a) Comparison on Basis of Strength, psi									
Present 8 ply	3.4	6490 (13)	4650 (11)	2605 (1)	2515 (1)	8480 (3)	7580 (3)	976 (3)	602 (3)
Revision (1) 5 ply	5.6	6590 (22)	6340 (34)	3010 (1)	2460 (2)	7680 (3)	5910 (6)	1059 (3)	480 (6)
(b) Comparison of Actual Test Values,[a] lb									
Present 8 ply	3.4	590	425	5190	5000	22.02 #	18.93#	...	..
Revision (1) 5 ply	5.6	620–105%	610–143%	5610–108%	4970–99%	20.75–94%	15.64–83%	...	..

[a] Specimens identical except for differences in thickness.
NOTE: Numbers in parenthesis indicate numbers of tests made.

made on the basis of total load is more applicable to this type of analysis since the thicknesses of the alternates are not always the same as those of the original constructions.

The alternate plywood constructions shown in Tables 2 and 3 were found to be acceptable from the standpoints of balance and strength. A proposed revision (2) for the rear uselage, also a 5-ply construction, and several proposed revisions for the outer wing panel, substituting 3-ply for 5-ply, were tested but were not considered acceptable, and were therefore not given further consideration.

Analysis of Strength Data

(a) *Front Fuselage.* Tests at 45 deg grain direction were not considered necessary in tension, compression, and shear, because in a balanced plywood construction these results would be directly proportional to those obtained at 0 deg and 90 deg. Inspection of test results for revisions (1) and (2) show that both alternate constructions are balanced in tension parallel and perpendicular to the grain and are equivalent in actual tensile strength to the original. Shear and bending strengths are higher as are the moduli of elasticity (based upon pounds per square inch) except for revision (1), an all-mahogany construction which at 90 deg, is 7 per cent lower. It was concluded that both revisions could be considered as entirely satisfactory replacements for the present construction.

(b) *Rear Fuselage.* A comparison of actual test values between the present construction and proposed revision (1) shows that the 5-ply construction is better balanced in tensile strength parallel and perpendicular to the grain and considerably stronger, particularly perpendicular to the face grain. Shear strength at 0 deg and 90 deg, and modulus of rupture at 0 deg are equivalent to the present constructions. The modulus of elasticity at 0 deg is somewhat higher owing to the thicker face plies.

Both modulus-of-rupture and modulus-of-elasticity values at 90 deg are lower than those obtained from the construction now in use. However, this difference is not too pronounced and is to be expected when fewer plies are used, because, in the 5-ply construction, the cross-bands which provide most of the stiffness perpendicular to the face grain, are closer to the core or neutral axis. From a structural standpoint, this 5-ply construction can also be considered as a satisfactory replacement for the 8-ply construction.

(c) *Wings.* Tests were not made to substantiate the proposed revisions to the wing-shell constructions. While it is true that the proposed revisions in the outer panel shells will introduce plywood of lower strength than the originally used birch-face constructions, these shells are not critically stressed, and the consequent reduction in strength, in part offset by using a $^1/_{32}$-in. instead of $^1/_{48}$-in. core, can be safely taken. The same is true of the center-section leading edge. In the highly stressed center-section top panels, however, the birch faces have been maintained and thickened. The additional thickness of the birch faces will adequately offset the slight reduction in over-all thickness. Also, at the highly stressed points, the alternate reinforcement construction is thicker than was the original. This will additionally offset the decrease in thickness of the outer plywood.

(d) *Fin.* The proposed substitution of mahogany-core material for the present poplar will increase the strength slightly.

(e) *Secondary Structures.* The proposed alternate constructions for the nose-wheel door, bomb-bay doors, wing tip, gun-turret fairing and tail cone are all on the side of increased strength.

Conclusions

In conclusion, it is felt that most of the difficulties being experienced by plywood-aircraft manufacturers can be traced directly to a lack of specific information on the part of the lumber manufacturer, the veneer and plywood manufacturer, and the subcontractor on the exact requirements of the aircraft industry.

To prevent further unnecessary delays in production of wood aircraft, the following recommendations are made:

1 That suppliers of basic materials give more consideration to the production of larger quantities of aircraft grades in conformance with current AN Specifications.

2 That the use of acceptable alternate materials to replace critical species be more widely adopted by the industry.

3 That aircraft manufacturers, wherever it is structurally possible, simplify and standardize the design and construction of flat- and molded-plywood parts.

Analysis of Stretch-Forming Double-Curved Sheet-Metal Parts

By R. B. GLASSCO[1] and N. O. MYKLESTAD[2]

Double-curved surfaces, typical in airplane design, are classified according to direction of curvature. A qualitative stress analysis is made of stretch-forming (process of clamping metal sheet at two opposite edges and forcing a punch into the taut sheet) each class of surface, with special reference to the stress transverse to the direction of restraint. It is shown that there are two distinctly different sources of transverse stress during stretching. The resultant stress may be tension or compression, depending upon the direction and amount of curvature. When the resultant transverse stress is compression, undesirable wrinkling of the sheet may occur.

Introduction

STRETCH-FORMING is a process of forming sheet metal into a curved surface by clamping the sheet at two opposite edges and forcing a punch of the desired shape into the taut sheet. A typical application of the process is the forming of the skin or outer surface of the modern metal airplane.

There are two limiting conditions in the application of the stretch-forming process. These limitations depend upon the amount of curvature in the desired surface. The first and more obvious limitation is tearing or rupturing of the sheet before it has been stretched over the entire surface of the punch. This limitation of tearing is a function of the ductility properties of the particular metal being formed. Some work has already been done to determine and improve the limit of stretch-forming with regard to tearing (1, 2).[3]

The second limitation in the application of the stretch-forming process is buckling of the sheet. This buckling results from compressive stresses transverse to the direction of restraint and leaves the formed sheet with undesirable permanent wrinkles. The factors determining whether a particular curved surface is subject to wrinkling or not are more obscure than they are for the limitation of tearing. In order to understand and, in some cases, eliminate wrinkling, it is desirable to make a qualitative stress analysis of a sheet subjected to the stretch-forming process. This is done by determining the effect of the surface shape upon the deformations and loads in the stretched sheet.

Classification of Shapes

Double-curved surfaces may be classified by considering variations in the two principal curvatures which define the shape, that is, a surface may be convex, concave, or flat in one direction for a particular curvature in the other direction. In this report the curvature will refer to the top side of the sheet being stretch-formed, so that in the schematic illustration at the top of Fig. 1, both the curvatures $1/R_1$ and $1/R_2$ are concave as shown. The curvature $1/R_1$ will be called the transverse curvature, and $1/R_2$ the longitudinal curvature. In Fig. 1 (*a* to *i*) are shown the different combinations of curvature, defined in the foregoing manner. The longitudinal direction is the direction of restraint as determined by the location of the clamps. This location is the same for all sketches in Fig. 1 and is indicated by the schematic illustration at the top of the figure.

It is not possible to form all of the shapes shown in Fig. 1 with a punch approaching from above. Shapes with longitudinal convex curvature, illustrated in (*c*), (*f*), and (*i*) of Fig. 1 cannot be formed this way, as the area of the formed part of the sheet will be above the ends of the punch. However, the elimination of these three types does not limit the number of shapes which can be stretch-formed, since they are merely the inverted images of the three types (*g*), (*d*), and (*a*), diametrically opposite in Fig. 1. In addition, type (*e*) in the center of Fig. 1 may be disregarded since it is a flat sheet that requires no forming.

There remain only five cases, namely, (*a*), (*b*), (*d*), (*g*), and (*h*), to be considered in this analytical study of stretch-forming of curved parts. Of these five cases, (*b*), (*d*), and (*h*) have only single curvature and are considered special cases having zero curvature in one of the two principal directions. The other two types, (*a*) and (*g*), the real double-curved shapes, may be considered as combinations of these three, that is, (*a*) is a combination of (*b*) and (*d*), while (*g*) is a combination of (*d*) and (*h*). Thus, if a study is made of the three simple types,[4] (*b*), (*d*), and (*h*), first, and then of the effect of combining them, all simple types of double-curved surfaces will be covered.

Qualitative Analysis

The following qualitative analyses explain how wrinkles may occur during stretch-forming of the shapes shown in Fig. 1:

Case 1—Concave Cylindrical Part. Take as the first case the stretching of the concave cylindrical part shown in Fig. 1 (*b*). This would be a difficult part to form by stretching in the direction indicated because of the variation of longitudinal strain across the width. When the punch descends, it first touches the sheet along the longitudinal line *AB*, as shown in Fig. 2. If the sheet is flat, there is no punch pressure exerted in this position because a flat horizontal sheet can have no vertical reaction at its clamps.

As the punch descends farther, the part of the sheet that is in contact with it will be forced down below the clamps, as shown in Fig. 3. Since the longitudinal center line of the sheet now slopes down at the clamps, the restraining force will have a vertical component the total amount of which is the punch load P.

The longitudinal center line of the sheet has now been stretched a small amount which gives rise to a longitudinal stress determined by the stress-strain diagram of the material. As stretching of the sheet progresses, this stress is soon brought above the yield point at the middle of the sheet while it is still zero at the sides

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[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Aviation Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

[4] The cylindrical surfaces (*b*), (*d*), and (*h*) would not actually be formed by stretching in practice but probably would be bent by rolling; however, it is the analysis of the cylindrical parts which forms the basis for the analysis of double-curved parts.

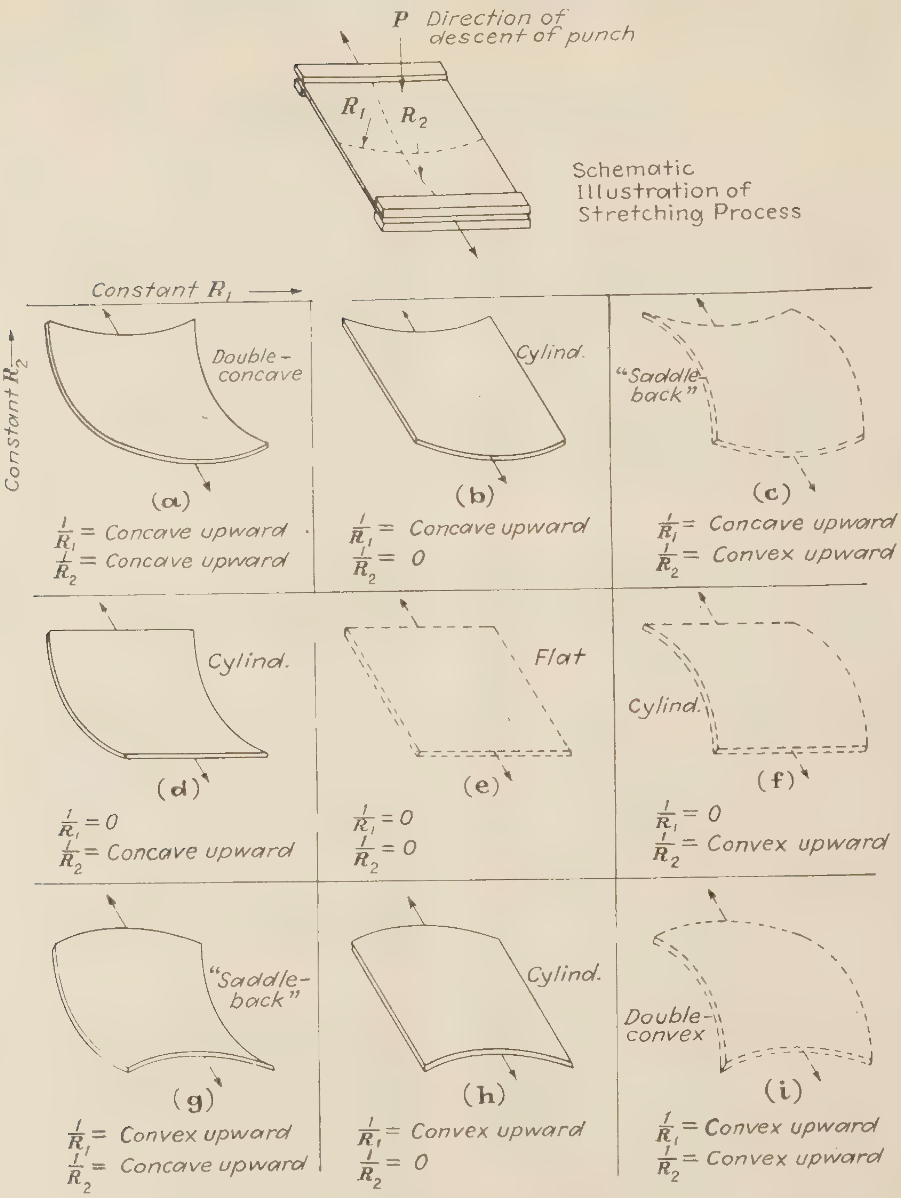

Fig. 1 Combinations of Curvature

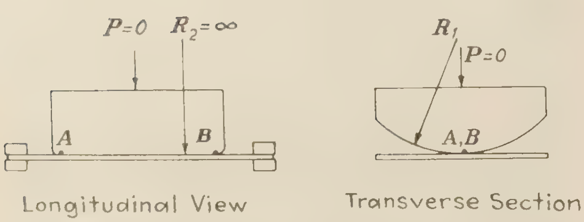

Fig. 2 Forming of Concave Cylindrical Part
(Condition just before stretching begins.)

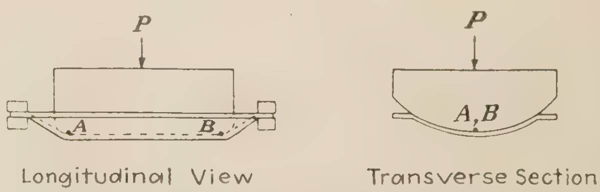

Fig. 3 Forming of Concave Cylindrical Part
(Condition just after stretching begins.)

where the punch has not yet contacted the sheet. This variation of strain across the width of the sheet is the cause of the tendency to wrinkle as explained later. Between points A and B there will be no pressure between the punch and the sheet, as such pressure could be only brought about by a tendency on the part of the sheet to bend up and press against the punch. Such a tendency is of course not present. This lack of pressure between the sheet and the punch means that the sheet has no support against wrinkling in this central portion between points A and B.

Transverse Stress Due to Uneven Stretching. How a transverse

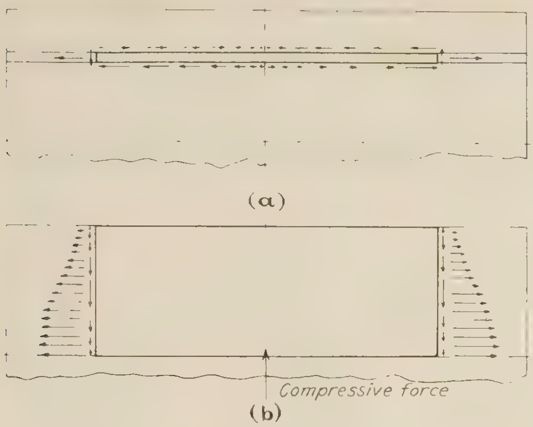

Fig. 4 Free Body Sketches, Illustrating Transverse Compression

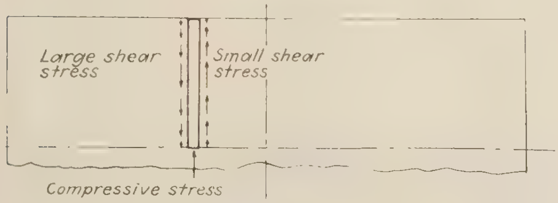

Fig. 5 Transverse Half-Strip, Illustrating How Transverse Stress Depends on Change in Shear Stress

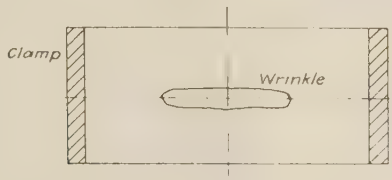

Fig. 6 Lengthwise Wrinkle Caused by Uneven Longitudinal Stretching

compressive stress can result from the uneven longitudinal stretching may be seen as follows: Consider the middle part of a narrow longitudinal strip of the sheet as a free body; it will have shear stresses along its sides as shown in Fig. 4 (*a*) in order to make its sides conform to the adjacent strips which were stretched different amounts. At the end of the strip there will be a transverse shear stress equal to the longitudinal one. When all adjacent strips are considered, from the edge of the sheet to the middle, a free body such as shown in Fig. 4 (*b*) will be obtained. It is seen from this that there must be a transverse compressive force on the sheet balancing the transverse shear forces.

The distribution of this transverse compression depends upon the longitudinal rate of change of shear stress. This is illustrated by the transverse half-strip shown in Fig. 5. The increase in the shear stress from the right side to the left side of the strip determines the magnitude of the transverse compression on this strip.

From symmetry the shear stress must be zero halfway between the clamps. It increases toward the ends of the punch, as shown in Fig. 4. This produces transverse compression over this region, resulting in a tendency to develop lengthwise wrinkling, as shown in Fig. 6.

Transverse Stress Due to Slope of Punch. At the ends of the punch, where the sheet bends up toward the clamps, the stretching force on a longitudinal strip will have a vertical component as shown by force S in Fig. 7. Since the side of the punch slopes, the normal pressure N between the punch and the strip will not be vertical but at an angle. To balance these forces S and N on the strip, there will have to be a third force which in this case must be a transverse tension T in the sheet.

The force T will be tangent to the punch and will be diminished by the friction force μN, where μ is the coefficient of friction. It is seen that as long as the slope of the strip is less than μ, the transverse stress remains zero. It must be remembered also that for a cylindrical punch this transverse tension is confined entirely to the part of the sheet which is pressed against the end edges of the punch, as there is no pressure between the punch and the sheet along the straight middle part. For a long cylindrical punch, then, this transverse tension will have little effect at the middle of the sheet and its influence on the wrinkling due to unequal tension, Fig. 6, will be but slight.

Case 2—Double Concave Part. Consider now the type of part which has a concave curvature in the longitudinal direction as well as in the transverse direction as shown in Fig. 1 (*a*). The longitudinal view, corresponding to Fig. 3, will now be as shown in Fig. 8. Due to the longitudinal curvature there will be normal pressure between the punch and the sheet all along its length, and the condition shown in Fig. 7 will hold true all along the length of the punch. The resulting transverse tension will, therefore, be effective in opposing the wrinkling shown in Fig. 6. The friction resulting from the normal pressure will also help to prevent wrinkling as it resists any relative motion between the punch and the sheet. Finally, the normal pressure also affords support to the middle of the sheet, which greatly increases its resistance to wrinkling.

The magnitude of the normal pressure between the punch and the sheet at any point is directly proportional to the longitudinal curvature at that point. How sharp this longitudinal curvature must be in order to avoid the wrinkling due to nonuniform stretching can best be determined by experiment. The minimum longitudinal curvature will be a function of the transverse curvature and the sheet thickness, as well as the dimensions of the punch, the method of support of the sheet, and the coefficient of friction.

For a longitudinal curvature too small to prevent wrinkling, it has been found that rubber pads stacked under the sheet give enough support to prevent wrinkling in many cases. This is illustrated by the two aluminum-alloy parts shown in Fig. 15.

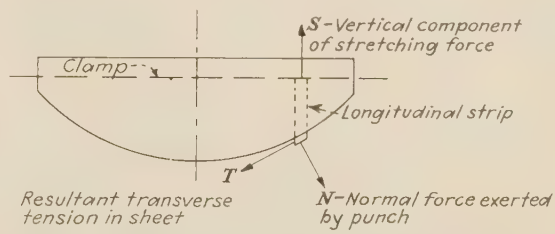

Fig. 7 Transverse Section at End of Punch Showing Forces on Longitudinal Strip of Sheet

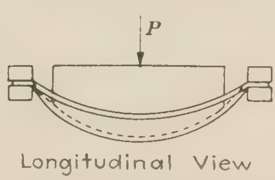

Fig. 8 Forming of Double Concave Part
(Condition just after stretching begins.)

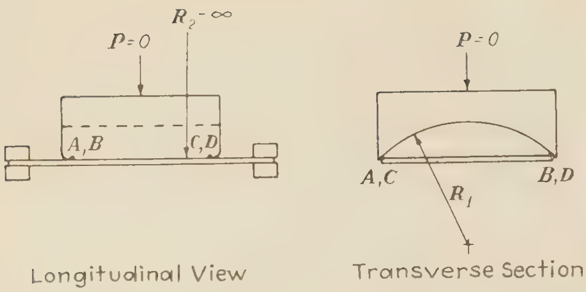

Fig. 9 Forming of Convex Cylindrical Part
(Condition just before stretching begins.)

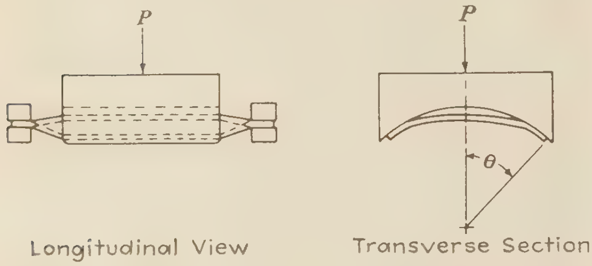

Fig. 10 Forming of Convex Cylindrical Part
(Condition just after stretching begins.)

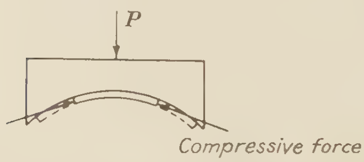

Fig. 11 Transverse Section Illustrating How Transverse Compression Pushes Sheet Against Punch

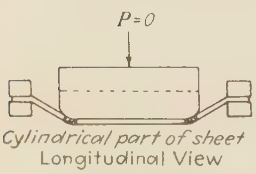

Fig. 12 Forming of Convex Cylindrical Part (Shaded Portion) From Sagging Sheet
(Condition just before stretching begins.)

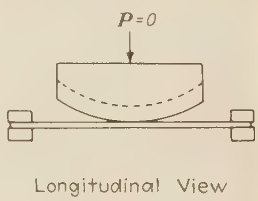

Fig. 13 Forming of Saddleback Part
(Condition just before stretching begins.)

This expedient can be used in certain types of stretching machines only.

Case 3—Convex Cylindrical Part. Consider now the stretch-forming of a convex cylindrical part with the direction of restraint parallel to the axis of the cylinder as shown in Fig. 1 (*h*). Such a part would not be formed in this manner in actual production but is important as the limiting case of saddleback parts.

This case of the convex cylindrical part is entirely different from Case 1, the concave part. The uneven tension now varies in the opposite manner across the width of the sheet and consequently produces transverse tension where Case 1 produced compression. Similarly, the transverse stress in the sheet due to the slope of the punch now becomes compression, rather than tension as in Case 1.

Assume first that the sheet is clamped perfectly flat before stretching begins. Then the picture will look like Fig. 9, just as the punch touches the sheet.

As the punch descends farther, it puts pressure on the four points, A, B, C, and D, of the sheet. No pressure is exerted between points A and C nor between points B and D for the same reason as explained in Case 1. The descent of the punch now stretches the sheet along the sides, rather than along the center line, and in the transverse section there will be two arcs of contact as shown in Fig. 10. The bending moments due to the curvature of these arcs will have a tendency to bend the center of the sheet also and actually raise the center of the sheet an infinitesimal amount above the clamps, as shown exaggerated in Fig. 10.

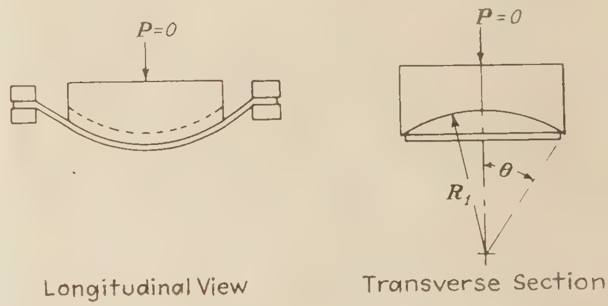

Fig. 14 Forming of Saddleback Part From Sagging Sheet
(Condition just before stretching begins.)

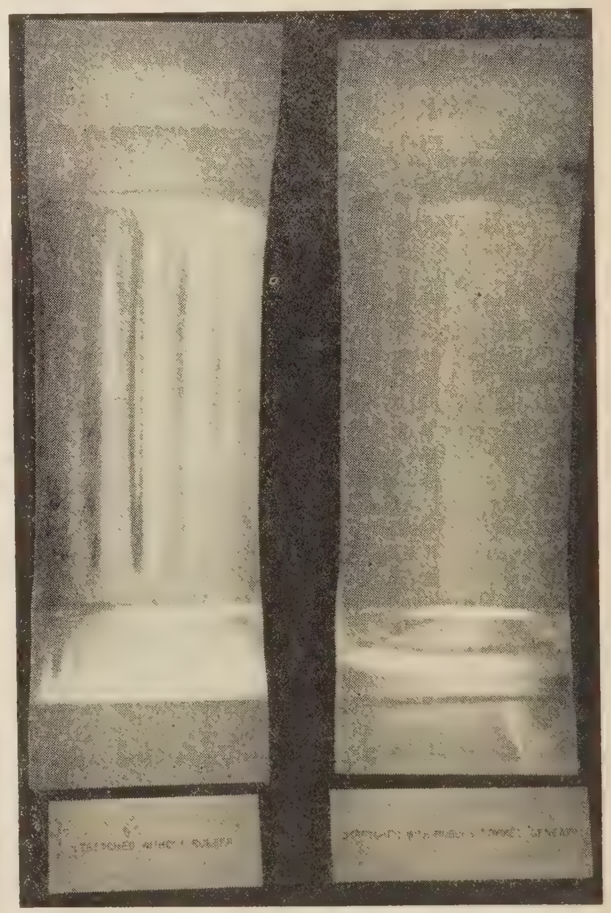

Fig. 15 Concave Cylindrical Parts
(Material, 0.010 24S-T Alclad. Stretched with and without rubber stacked beneath.)

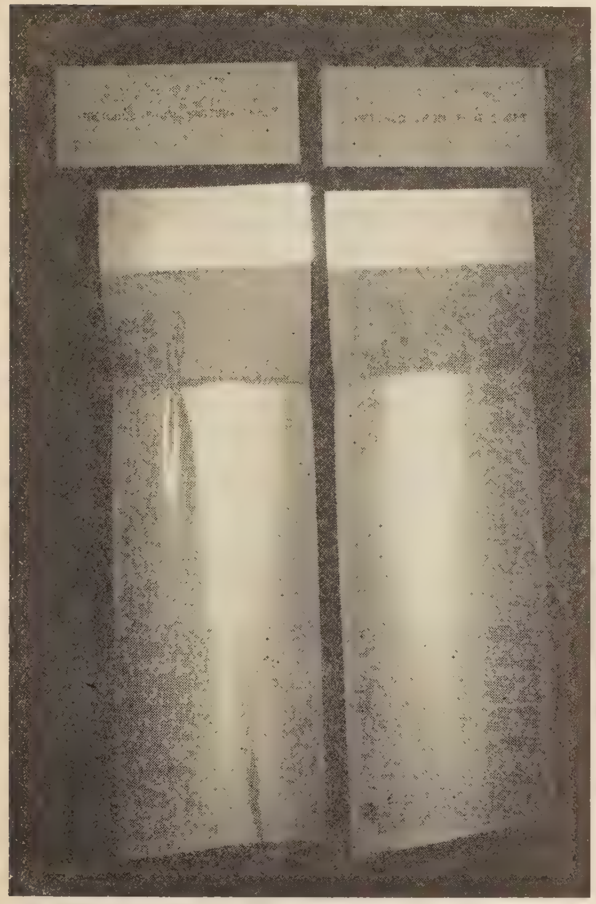

FIG. 16 CONVEX CYLINDRICAL PARTS
(Material, 0.010 24S-T Alclad. Stretched from sagging and flat sheets.)

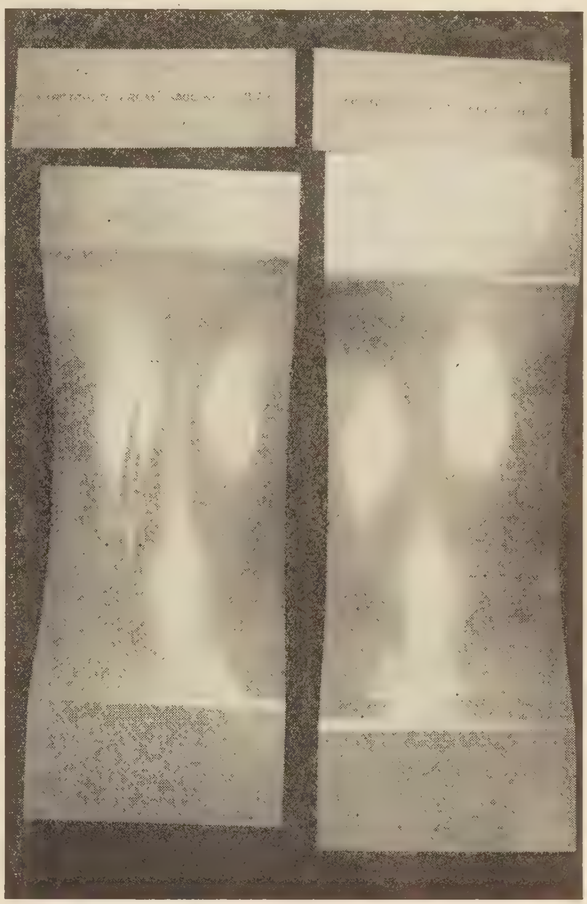

FIG. 17 SADDLEBACK PARTS
(Material, 0.010 24S-T Alclad. Stretched from sagging and flat sheets.)

Because of the fact that the sheet first comes in contact with the part of the punch that has the steepest slope, as indicated by the angle θ in Fig. 10, the transverse compressive stress due to this slope will be predominant over the transverse tension resulting from unequal stretching. This transverse compressive force will tend to push the center of the sheet farther in against the punch, as shown in Fig. 11. This represents buckling of the sheet, but in such a way that the buckle conforms to the transverse curvature of the punch. Additional buckles of smaller wave length (wrinkles) cannot occur before the sheet rests firmly against the center of the punch. Then this firm support of the sheet by the punch will help to prevent such wrinkles as the transverse compressive stress increases to its maximum value.

On the other hand, a slight variation in the method of clamping the sheet will alter considerably the possibility of wrinkles forming. If the sheet is placed in the press in such a way that it is allowed to sag between the clamps so that just before stretching begins it has a shape such as shown in Fig. 12, the case will be entirely different. In this case the sheet will be bent into a cylindrical shape at each end of the punch, as shown in Fig. 12. This longitudinal curvature at each end of the punch will offer a considerable resistance to transverse buckling of the sheet and will also prevent the sheet from bending easily to conform to the curvature of the punch. However, if the transverse compressive forces become large enough to cause buckling, these buckles will be of small wave length and will be likely to remain as permanent wrinkles when the stretching operation is completed. This is illustrated by the aluminum-alloy parts shown in Figs. 16 and 17.

Case 4—Convex-Concave Part (Saddleback). Consider that the part in Case 3 has in addition to the transverse convex curvature a longitudinal concave curvature. This results in a saddleback shape, such as shown in Fig. 1 (*g*). The same reasoning as in Case 3 also applies here regarding the transverse stresses due to uneven stretching and transverse slope of punch. The difference is, however, that in this case the transverse compressive stress is distributed over the entire part of the sheet in contact with the punch instead of being confined to the ends of the punch.

If a flat sheet is used (see Fig. 13), it will have a tendency to conform easily to the transverse curvature of the punch just as in Case 3, and the resulting support by the punch will help to prevent wrinkles as the transverse compressive stress increases. If the sheet is allowed to sag, however, it will conform to the longitudinal curvature of the punch just before stretching begins (see Fig. 14). This cylindrical part of the sheet will offer a considerable resistance to buckling just as for the sagging sheet in Case 3 (see Fig. 12). When buckling finally does occur, however, the resulting short waves are likely to remain in the sheet as wrinkles. It has been found that even a small amount of sag is often enough to cause wrinkling.

Quantitative Analysis of Saddleback Stretch-Forming. If a

saddleback part is stretch-formed from a sagging sheet, an analysis of the transverse compressive stress due to the transverse slope of the die can be made under certain assumptions. As already pointed out, starting with a sagging sheet is generally not the most favorable method of stretch-forming saddleback parts, but for cases where it is necessary, this analysis is applicable. The detailed analysis for this case may be found in the Appendix.

The assumptions made are as follows: The sheet has an initial sag just large enough to make it tangent to the punch near its ends (which corresponds fairly well to present methods); the influence of bending of the sheet is negligible; the width of the sheet does not change during stretching; and, in regions being stretched, the strain is in the plastic range, and hence the stress is approximately constant across the stretched part of the sheet. Also, the transverse stress due to unequal stretching is neglected.

With these assumptions it is found that the maximum transverse compressive stress in the cylindrical part of the sheet would be

$$(f_c)_{\max} = \frac{R_1}{R_2} f_p[1 - e^{-\mu\theta}(2\mu \sin\theta + \cos\theta)] \ldots\ldots [1]$$

in which equation the following notations are used:

R_1 = radius of transverse convex curvature of the part, in.
R_2 = radius of longitudinal concave curvature of the part, in.
f_p = longitudinal plastic tensile stress due to stretching (assumed constant), psi
θ = angle between vertical axis of punch and normal to stretched edge of sheet (see Fig. 14) (assumed constant, but does actually diminish slightly as the stretching progresses), radians
μ = coefficient of friction
e = 2.718, base of natural logarithms

The resistance to transverse buckling of a cylindrical sheet of radius R_2 and thickness t may be given by the following formula (3)

$$f_{cr} = E\,\frac{0.6\,\dfrac{t}{R_2} - 10^{-7}\,\dfrac{R_2}{t}}{1 + 0.004\,\dfrac{E}{f_{cy}}} \ldots\ldots\ldots\ldots [2]$$

In this equation

E = modulus of elasticity in elastic range, psi
f_{cy} = compressive yield stress, psi

If $(f_c)_{\max}$ according to Equation [1] is larger than f_{cr} according to Equation [2], buckling of the unstretched cylindrical center part of the sheet is likely to occur.

Conclusions

The conclusions to be drawn from the qualitative analyses and verifying tests made are:

1 There are two distinctly different sources of transverse stress during stretching. One is due to the variation of longitudinal tensile strain across the width of the sheet, and the other is due to the slope of a transverse cross section of the punch. These two effects are often opposed to each other so that if one causes compression, the other causes tension in the transverse direction. If the compressive effect predominates, a tendency to form longitudinal wrinkles will result.

2 The methods of stretch-forming may be so improved that, in some cases, the compressive forces causing wrinkling may be avoided, and in other cases enough support may be given to the sheet to prevent wrinkles even though the compressive forces are present. The only limit to stretch-forming in these cases is then the tearing of the sheet due to excessive elongation.

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Appendix

Nomenclature

The following nomenclature is used in the Appendix:

R_1 = radius of transverse convex curvature of part, in.
R_2 = radius of longitudinal concave curvature of part, in.
f_p = longitudinal plastic tensile stress due to stretching (assumed constant), psi
f_c = transverse compressive stress on cylindrical part of sheet, psi
ϕ = angle between vertical axis of punch and normal to any element in sheet in contact with punch (see Fig. 18), radians
β = value of ϕ for innermost element of sheet in contact with punch (see Fig. 18), radians
t = sheet thickness, in.
dl = longitudinal dimension of an infinitesimal element of sheet, in.
N = normal force of punch on an infinitesimal element of sheet (see Fig. 20), lb
V = vertical resultant of longitudinal tensile forces on an infinitesimal element of sheet (see Fig. 20), lb
F = transverse compressive force on an infinitesimal element of sheet (see Fig. 20), lb
dF = change in F across infinitesimal element of sheet (see Fig. 20), lb
μ = coefficient of friction
e = 2.718, base of natural logarithms
E = modulus of elasticity, psi
f_{cy} = compressive yield stress, psi

Quantitative Analysis of Saddleback Stretch-Forming

Saddleback parts often wrinkle because the transverse curvature causes a transverse compression during the stretch-forming operation. The wrinkles are more apt to occur if the sheet is clamped with an initial sag. A quantitative analysis for the buckling stress in a saddleback part formed from a sheet with an initial sag will be given.

The following assumptions are made:

1 The initial sag in the sheet is enough to make the sheet tangent to the punch near its ends (Fig. 14).

2 The influence of bending is negligible.

3 The width of the sheet does not change during stretching.

4 The strain is in the plastic range, and hence the tensile

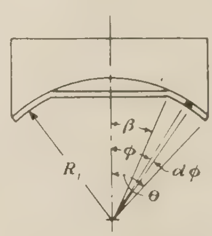

Fig. 18 Forming of Saddleback Part From Sagging Sheet
(Condition after stretching begins.)

stress is approximately constant across the stretched part of the sheet.

5 The transverse stress due to unequal stretching is negligible.

Just before stretching begins the sheet has a cylindrical shape with the radius R_2 and touches the punch all along its circular edges, as shown in Fig. 14. After the forming is partially completed, the sheet will be stretched into contact with the punch for some distance in from the edges, and the center portion will still be cylindrical, as shown in Fig. 18.

Consider now an element of the stretched part of the sheet located at an angle ϕ from the vertical axis of the punch, as shown in Fig. 18. The width of this strip is $R_1 d\phi$, and the length is taken as dl as shown in Figs. 19 and 20. Due to the stretching

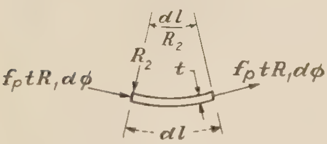

Fig. 19 Longitudinal View of Element in Stretched Part of Sheet

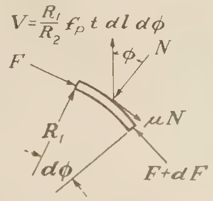

Fig. 20 Transverse View of Element in Stretched Part of Sheet

stress, the longitudinal forces acting on this strip are $f_p t R_1 d\phi$ (Fig. 19). The vertical resultant V of these forces is then

$$V = 2(f_p t R_1 d\phi) \sin \frac{dl}{2R_2} = \frac{R_1}{R_2} f_p t\, dl\, d\phi$$

which is shown in Fig. 20. Then in the transverse plane, Fig. 20, the condition for equilibrium in the tangential direction is satisfied if

$$\begin{aligned} \mu N &= (F + dF) - F + V \sin \phi \\ &= dF + \frac{R_1}{R_2} f_p t\, dl\, d\phi \sin \phi \qquad [3] \end{aligned}$$

The condition for equilibrium in the normal direction is satisfied if

$$N = 2F \sin \frac{d\phi}{2} + V \cos \phi$$

Since

$$\lim_{d\phi \to 0} \sin \frac{d\phi}{2} = \frac{d\phi}{2},$$

the foregoing equation becomes

$$N = F d\phi + \frac{R_1}{R_2} f_p t\, dl\, d\phi \cos \phi \qquad [4]$$

Eliminating N from Equations [3] and [4] gives

$$\frac{dF}{d\phi} - \mu F = \frac{R_1}{R_2} f_p t\, dl\, (\mu \cos \phi - \sin \phi) \qquad [5]$$

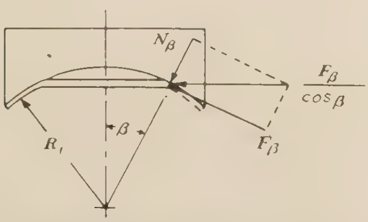

Fig. 21 Transverse Compressive Force on Center Cylindrical Section After Stretching Begins

Solving Equation [5] for the boundary condition that $F = 0$ when $\phi = \theta$ gives

$$F = \frac{R_1}{R_2} \cdot \frac{f_p t\, dl}{1 + \mu^2} \left\{ 2\mu \sin \phi + (1 - \mu^2) \cos \phi - e^{-\mu(\theta - \phi)} [2\mu \sin \theta + (1 - \mu^2) \cos \theta] \right\} \qquad [6]$$

Equation [6] gives the value of F for any angle ϕ between β and θ, but the particular value of interest is F_β, the one for $\phi = \beta$ where the cylindrical part of the sheet meets the punch.

The compressive force on the cylindrical part of the sheet will be $\frac{F_\beta}{\cos \beta}$, as shown in Fig. 21, where N_β is the punch reaction at that point. (The friction force μN_β is neglected here for simplicity.) The compressive stress f_c on the cylindrical part of the sheet is then

$$\begin{aligned} f_c &= \frac{F_\beta}{t\, dl \cos \beta} \\ &= \frac{R_1}{R_2} \cdot \frac{f_p}{1 + \mu^2} \left\{ 2\mu \tan \beta + (1 - \mu^2) - \frac{e^{-\mu(\theta - \beta)}}{\cos \beta} [2\mu \sin \theta + (1 - \mu^2) \cos \theta] \right\} \qquad [7] \end{aligned}$$

Since the buckling load of a thin-walled cylinder is independent of its length, it is only the maximum compressive stress $(f_c)_{\max}$ that is important in determining whether or not buckling will take place. The value of β at which $(f_c)_{\max}$ occurs is found by setting the first derivative of f_c with respect to β equal to zero

$$\frac{df_c}{d\beta} = 0 \qquad [8]$$

By differentiating Equation [7], it is found that

$$\frac{df_c}{d\beta} = \frac{R_1}{R_2} \frac{f_p}{(1 + \mu^2) \cos^2 \beta} \left\{ 2\mu - (\sin \beta + \mu \cos \beta) e^{-\mu(\theta - \beta)} [2\mu \sin \theta + (1 - \mu^2) \cos \theta] \right\}$$

To satisfy the condition of Equation [8], the expression within the braces must be zero

$$2\mu - (\sin \beta' + \mu \cos \beta') e^{-\mu(\theta - \beta')} [2\mu \sin \theta + (1 - \mu^2) \cos \theta] = 0$$

where β' is the particular value of β giving the maximum value of f_c. Placing all terms containing the variable β' on the left gives

$$(\sin \beta' + \mu \cos \beta') e^{-\mu(\theta - \beta')} = \frac{2\mu}{2\mu \sin \theta + (1 - \mu^2) \cos \theta} \qquad [9]$$

This value of β', found from Equation [9] and substituted into Equation [7] for the compressive stress, gives $(f_c)_{\max}$. Thus

$$(f_c)_{\max} = \frac{R_1}{R_2} \cdot \frac{f_p}{1 + \mu^2} \left\{ 2\mu \tan \beta' + (1 - \mu^2) - \frac{e^{-\mu(\theta - \beta')}}{\cos \beta'} [2\mu \sin \theta + (1 - \mu^2) \cos \theta] \right\} \qquad [7a]$$

However, no simple explicit expression for β' can be obtained from Equation [9] to substitute into Equation [7a], but the following study of Equation [9] indicates that an accurate approximate formula for $(f_c)_{\max}$ can be obtained which does not contain β'.

Take a representative case such as that wherein $\theta = 60$ deg. Solving Equation [9] for β' for this case, it is found that

$$\begin{aligned} \beta' &= 21.6 \text{ deg when } \mu = 0.20 \\ \beta' &= 13.1 \text{ deg when } \mu = 0.10 \\ \beta' &= 1.7 \text{ deg when } \mu = 0.01 \\ \beta' &= 0 \text{ deg when } \mu = 0.00 \end{aligned}$$

Thus, when the coefficient of friction μ is very small, as would be the case for a lubricated punch, the value of β' is also very small, that is, the maximum compressive stress occurs when almost the entire sheet is in contact with the punch. An approximate form of Equation [7a] may be obtained for small values of μ and β' by considering that

$$\begin{aligned} \mu^2 &\cong 0 \\ \mu \tan \beta' &\cong 0 \\ e^{\mu\beta'} &\cong 1 \\ \cos \beta' &\cong 1 \end{aligned}$$

The approximate equation for $(f_c)_{\max}$ then becomes

$$(f_c)_{\max} \cong \frac{R_1}{R_2} f_p \left[1 - e^{-\mu\theta}(2\mu \sin\theta + \cos\theta)\right] \ldots\ldots [7b]$$

This is the equation given as Equation [1] in the main report.

The resistance to transverse buckling of the cylindrical portion of the sheet is given by the following formula (3)

$$f_{cr} = E \frac{0.6 \dfrac{t}{R_2} - 10^{-7} \dfrac{R_2}{t}}{1 + 0.004 \dfrac{E}{f_{cy}}} \ldots\ldots\ldots\ldots [10]$$

This is the equation given as Equation [2] of the main report. If $(f_c)_{\max}$ according to Equation [7b] is larger than f_{cr} according to Equation [10], buckling of the unstretched cylindrical center part of the sheet is likely to occur.

Plastic Plywoods in Aircraft Construction

By R. D. HISCOCKS,[1] OTTAWA, CANADA

Using a typical fuselage fabricated from plastic plywood and veneer as an example, the author indicates the general procedure followed in building wood aircraft. Basic design problems were solved by extensive investigations and tests conducted by the National Research Council. A method is described for fabricating disposable fuel tanks for fighter planes from veneer and plywood, as representing the wide range of possiblities of wood in war and post-war industry.

At the beginning of the war it was felt in the National Research Council that the possibilities of the wooden airplane were not fully appreciated. It was considered that the factors which govern the speed with which a prototype machine might be developed and placed in full-scale production weighed heavily in favor of wood, especially when the question is examined in the light of Canadian facilities.

As a result, we undertook in the Structures Section to set up a small pilot plant in order to test and develop these ideas and, in particular, with the object of demonstrating that recent advances in materials and techniques can be applied rapidly to produce an airplane which is not merely a metal substitute but a machine of technical merit.

Strength Properties of Wood

Although wood is one of the oldest structural materials in the world there has accumulated surprisingly little information as to its strength properties. Admittedly far from the isotropic ideal dear to the elastician, the elastic properties of the material are specified by at least 12 constants or rather parameters which vary with moisture content, specific gravity, and even from the heartwood to the sapwood of the same tree. Time effects are important and, particularly when shear loads are involved, both deflections and loads at failure will vary with the time the load has been applied.

This may sound like a rather vague proposition for an engineering material but if the experimental data on the external loading condition of an airframe is critically examined, it is apparent that the only logical assessment should be on a statistical basis; and therefore, providing the probable variations fall within reasonable limits, there is no inconsistency in employing a material the strength properties of which are not uniquely determined. Moreover, since the leading dimensions of an airplane are defined chiefly by aerodynamical and operational requirements, few compact structural sections can be employed without an excessive penalty in weight. Stiffness, therefore, rather than ultimate strength considerations usually govern the design. In elastic-stability problems, it can be shown that the ratio of Young's modulus to density, to density squared, or even to density cubed is the figure of merit for any material and that, under such conditions, an expanded material such as wood is supreme. Finally, with the present insistence on perfectly smooth surfaces, it is not difficult to see that contemporary metal construction is considerably handicapped.

Tests on Basic Design Problems

In view of the broad scope of the problem, we felt at the Research Council that useful results could most quickly be obtained by concentrating on one or two design problems, performing such basic tests as from time to time appeared desirable and acquiring in the same process a good deal of useful experience in shop technique.

One cannot proceed very far with the design of shell structures without some basic data, and a few of our early tests were conducted on cylinders in compression, Fig. 1. Some of these

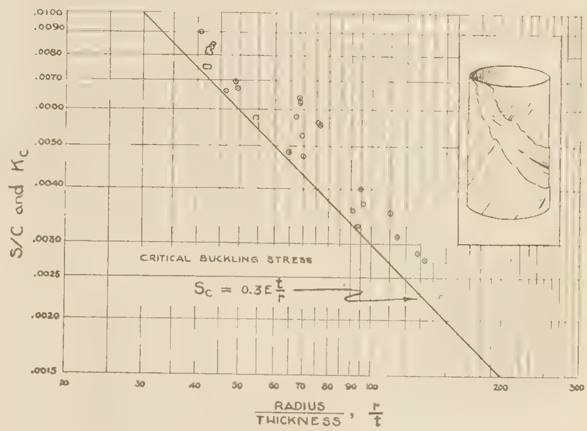

Fig. 1 Buckling Tests on 45-Deg Birch and Spruce Plywood Cylinders

cylinders were hot-press-molded with a rubber bag in the autoclave, others were formed with two-ply veneer and cold-press resin on a wrapping machine. The grain of the wood was in all cases at 45 deg to a generator, with inner and outer plies parallel and the grain of the core plies at right angles. The diameter of the cylinders ranged from 6 to 24 in. We have been able to detect a slight length effect with matched specimens but consider that it is negligible for all practical purposes.

It is interesting to observe that the critical compressive buckling stress can be expressed in the familiar form $F_{crit} = KE\frac{t}{R}$, where K is an experimental constant, E is the modulus of elasticity of the material, in our case parallel to the direction of the load, t is the wall thickness, and R the radius of curvature. A conservative value for design purposes is obtained by taking $K = 0.3$, a value which has enjoyed considerable popularity in the design of metal structures for years.

We have conducted few bending tests on cylinders to date and, in most applications, the problem is complicated by the addition of shear loads. The results of a few pure bending tests indicate that the critical stress again, as in the case of metal cylinders, is some 25 to 40 per cent higher than the critical value in compression.

The curves shown in Fig. 2 are based on the results of a fair number of compression tests on plywood blocks. With compact section, the elastic modulus and ultimate strength vary rapidly with changes in the grain angle.

The low bearing strength of wood perpendicular to the grain

[1] Aeronautical Division, National Research Council of Canada.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

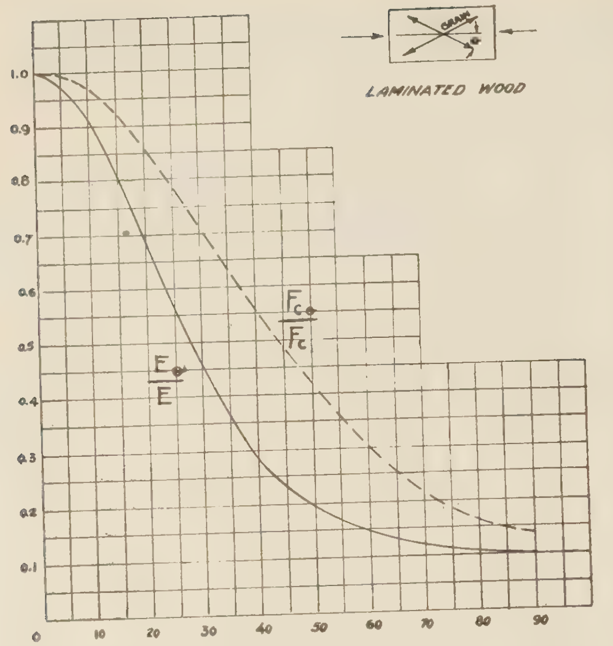

Fig. 2 Relative Ultimate Compressive Strength and Young's Modulus Versus Grain Angle

is a shortcoming which can be avoided by the use of plywood. The values given in Fig. 3 have been found useful in the design of joints employing rivets or short bolts. With concentrated loads of any magnitude, the size and number of pins required to transmit the forces directly to the plywood structure usually become formidable, and the need arises for an intermediate material having a high bearing strength, reasonable tensile and shear values, and a moderate density. We have found phenolic-fabric-base laminates satisfactory for this purpose. These are available with a bearing strength of the order of 30,000 psi. Form factors are apt to be unpredictable, however, and it is good practice to conduct a few simple tests in order to determine safe loads for any specific condition.

Adhesives an Important Factor in Plywood Structures

The phenolics usually will bond to wood effectively only with hot-press resins. Since the temperature coefficients of linear expansion for the plastics usually differ substantially from that of wood, it is important to avoid excessive temperature stresses. We consider that the most satisfactory expedient is first to glue a thin layer of veneer to the laminate with hot-press adhesive and then to bond the wood face to the remainder of the structure with a cold-setting adhesive. The same considerations apply in bonding metal to wood.

The problem of diffusing a concentrated load into a shell structure is always troublesome and particular care should be taken with plywood structures because wood has no elastic limit in tension. Highly localized tensile stresses may cause failure even if the average stress in a member is quite moderate, be-

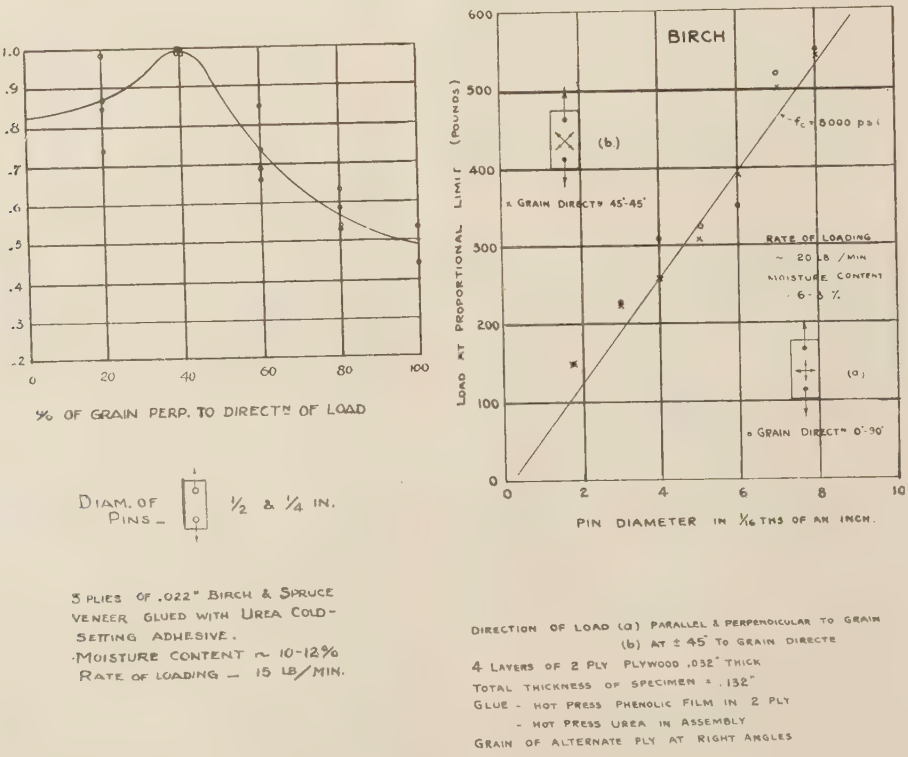

Fig. 3 Bearing Strength of Plywood

(*a*, Variation with percentage of end grain. *b*, Variation with pin diameter.)

cause the material cannot yield plastically and distribute the local stresses.

Many of the weathering difficulties which have arisen with wooden airplanes in the past can be attributed to the use of primitive glues, and it is safe to say that recent advances in the art largely are the outcome of improvements which have occurred in glues. Many synthetic-resin adhesives are now available which have a high resistance to water, ample shear strength, and are proof against mold and extremes of temperature. Quite a number will even survive the boiling-water test still cherished by some specifications, although it has never been explained why anyone should want to boil an airplane.

Contrary to popular opinion, high pressures are not required in gluing, and it is quite possible to construct satisfactory joints with a pressure of less than 1 psi if the viscosity of the glue is not excessive. The pressure requirements are determined solely by the forces necessary to obtain a thin glue line and, if the veneers are thin and the pressure uniformly distributed, then the forces required are small. With more rigid sections, high pressures are essential only if the faces of the wood do not match reasonably well. On thin members, nails or screws will provide ample pressure but it should not be assumed that the nails or screws will be effective after the glue has set because large shear deflections, sufficient to cause the glue to fail, are necessary for these elements to assume any load.

It is not unduly difficult to produce a bond with a shear strength equal to that of the wood, but it does not follow that every joint can transmit a load corresponding to the block shear strength of the wood. For example, in the standard three-ply test specimen, the stresses in the joint are far from pure shear, and it is not surprising that the failing load, for a thin specimen and with the grain in the core parallel to the face grain, is of the order of 25 per cent of the strength of the wood in pure shear. A further reduction results when the grain of the intermediate ply is not parallel to the load; at 90 deg, for example, there is a further decrement of 30 to 50 per cent due to the rolling effect of the wood fibers.

The rolling-shear type of failure has always to be guarded against. Fig. 4 is a very convenient spar section which behaved rather badly until we provided sufficient glue area in the flange-web joint. The bond between the parallel fibers in the flange and the web was entirely satisfactory but trouble arose where the veneers crossed at right angles further in the web.

Sometimes failures in box beams are unfairly charged to the glue. A thin web will buckle at very moderate shear stresses and continue to carry the load in diagonal tension. As a result the maximum shear stresses in the flange-web joint are more than double the values calculated from simple shear considerations.

It relieves the load on glued joints considerably if the ends of stiffeners are tapered. In flooring, decking, and the like, where local distortion may be large, we find rivets extremely effective in discouraging peeling in the glue joint.

In gluing it is of the utmost importance to match the moisture content of the pieces which are to be glued together. If a thin strip of veneer is coated with liberal quantities of a water-soluble glue and bonded to a heavier piece which is perfectly dry, it is obvious that the wet member will contract in drying and the result will be a badly warped component. Moreover, wood is a hygroscopic material and no amount of finishing will prevent it from shrinking and swelling with changes in the moisture content of the air. Most of the dimensional change occurs in a direction perpendicular to the grain and, if two pieces of veneer are glued together with the grain at right angles, then at only one value of the relative humidity will the panel remain flat.

For a stable material, it follows that three plies of veneer are the minimum. Unfortunately, conventional three-ply is composed of veneers of equal thickness and the bending stiffness parallel to the face grain is some 12 times the stiffness at right angles. Therefore, unless special design considerations demand a much superior strength in one direction, from the standpoint of general stiffness and dimensional stability, it is preferable to employ a construction which gives a more comparable stiffness in both directions (Fig. 5).

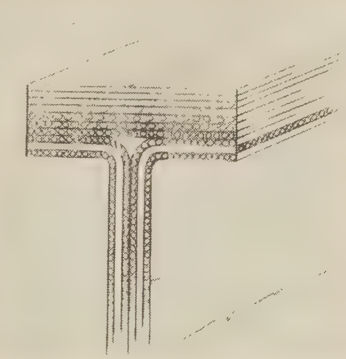

Fig. 4 Spar Section Showing Construction to Avoid Rolling-Shear Failure

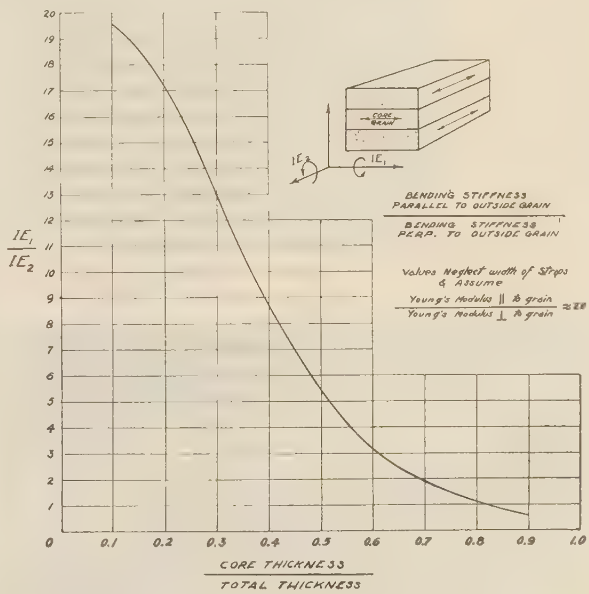

Fig. 5 Three-Ply Plywood in Flexure

Tests on Molded Fuselage

Turning our attention to more ambitious experiments, Fig. 6 indicates the general structure of a rear fuselage which we designed and constructed in molded plywood. It did not appear desirable to redesign the forward portion of the structure because the existing steel-tube framework with detachable side panels permitted access to a large amount of equipment which could not readily be moved. Hence the detail design of this portion was predicated somewhat by the original metal layout, but every attempt has been made to approach the true monocoque form. The skin is sufficiently thick to carry all loads ex-

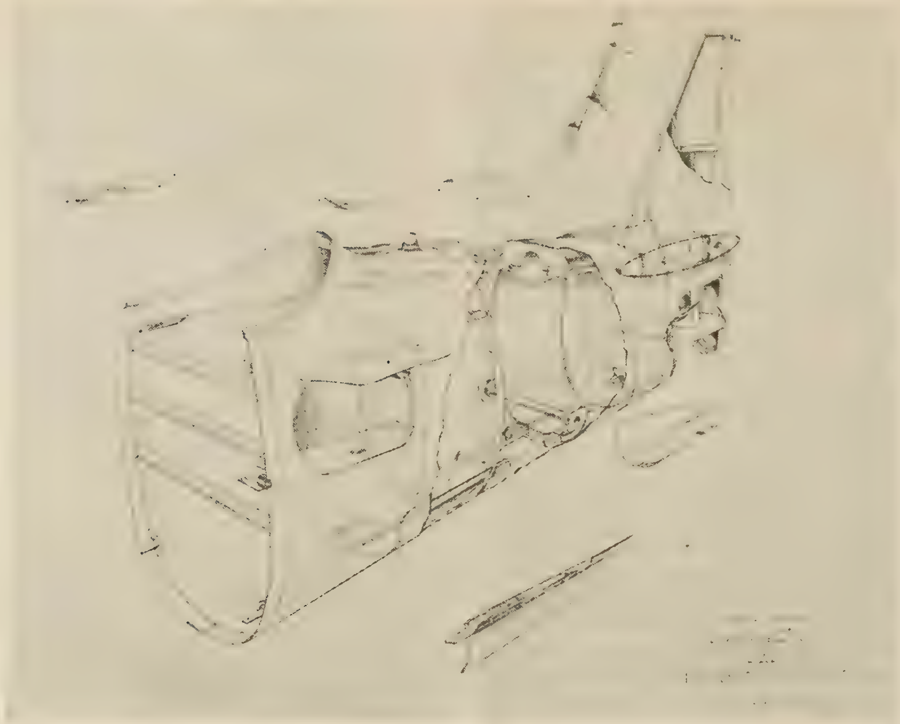

Fig. 6 Rear Fuselage Structure of Molded Plywood

Fig. 7 Four Layers of Spruce Veneer Bonded With Thermosetting Urea Glue Form the Fuselage Skin Which Is Molded in Halves

cept in the vicinity of cutouts and in the forward portion where the loads from the four attachment points are distributed by stub longerons. Light wooden frames maintain the correct shape. Plastic pads transmit the axial loads from the attachment fittings to the longerons. It is interesting to note that the problem of stress diffusion is frequently less difficult with a plywood than with a metal structure. With the longerons, for example, we have endeavored to taper the cross-sectional area and to graduate the elastic properties by varying the construction so that the glue joint between the longeron and the skin is uniformly stressed.

The skin, Fig. 7, which is molded in two halves, is composed of four layers of spruce veneer bonded with thermosetting urea glue. The veneers are trimmed to shape and stapled in place on a wooden mold with the inner and outer layers at right angles to the middle layers and at an angle of 45 deg to the fuselage center line. Successive layers are coated with adhesive, the

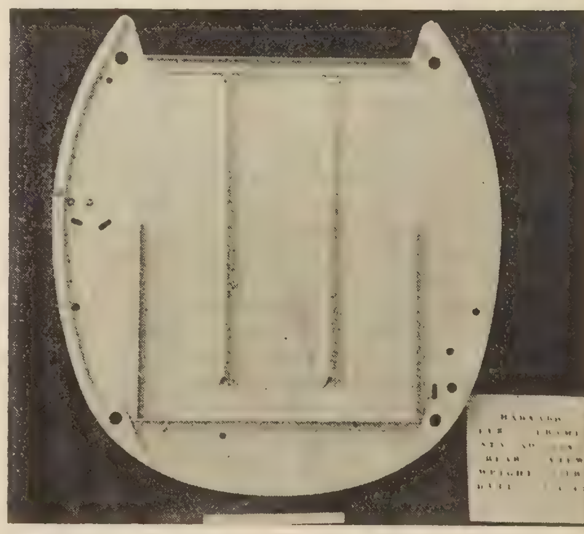

Fig. 8 View of Front Bulkhead

assembly is sealed in a rubber bag and cooked with steam in the autoclave for about 35 min at a temperature of 240 F and a pressure of 30 psi.

Fig. 8 is a view of the front bulkhead. Plastic pads again transmit shear loads from the fittings to the frame. It will be observed that we have borrowed a leaf from metal practice in using molded top-hat sections to stiffen the flat panels. We prefer to fabricate the frames separately, mounting these in a jig to glue the half-skin to the frames with cold-setting glue. Woodscrews at a liberal pitch are supplemented with webbing

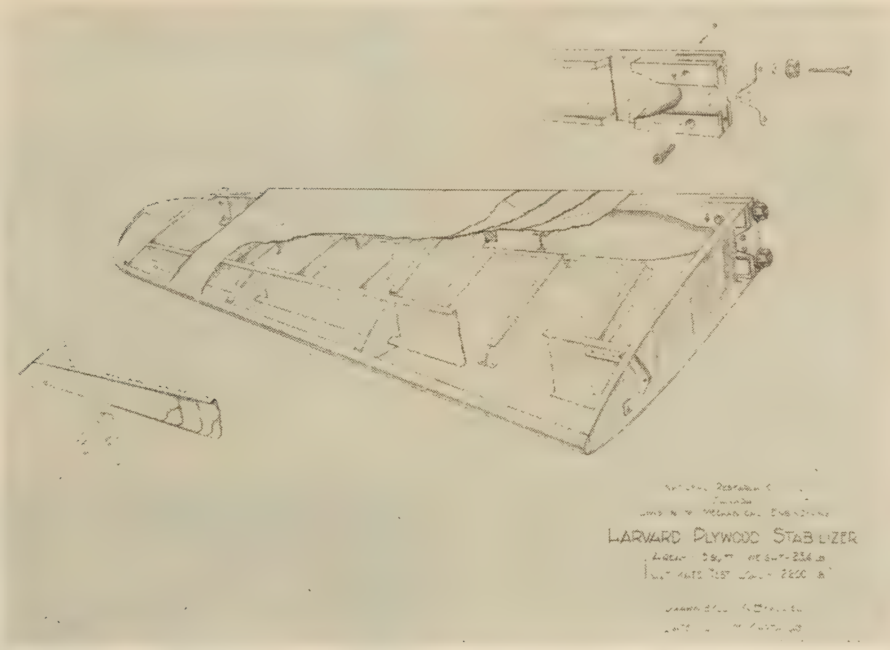

Fig. 10 Stabilizer Structure

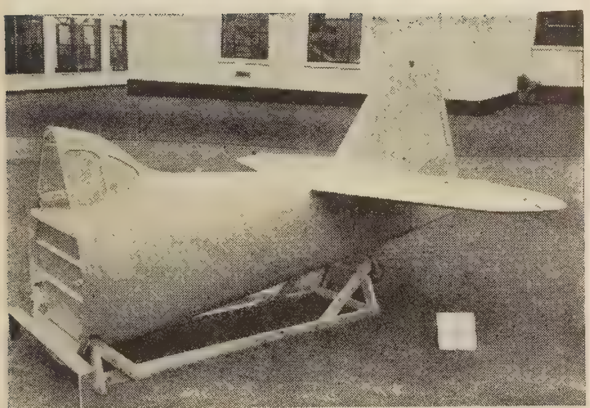

Fig. 9 Completed Rear Fuselage

at appropriate sections to apply the pressure and, with the drying accelerated by infrared lamps, the entire operation is completed in less than 1 hr.

The fuselage, Fig. 9, was static-tested in six different flight and landing conditions and proved well up to design requirements. Weighing 154 lb, as tested, it has carried a down load on the tail equal to the design breaking load for a Spitfire fighter aircraft. In the landing attitude it has sustained a tailwheel reaction slightly in excess of 2.5 tons.

Fig. 10 is a view of the stabilizer structure. The skin is hot-press-molded and stiffened with channel-section molded stiffeners. The ribs are also flanged and bonded in a rubber bag. For the rear spar we chose an I-beam section with the grain in the plywood web running diagonally. The root fitting consists of a pair of ball-and-socket joints attached to trunnions. These trunnions transmit the loads to a phenolic laminate inserted in the spar flanges. The total stabilizer weight is 23 lb. On test it carried 2200 lb without evidence of distress.

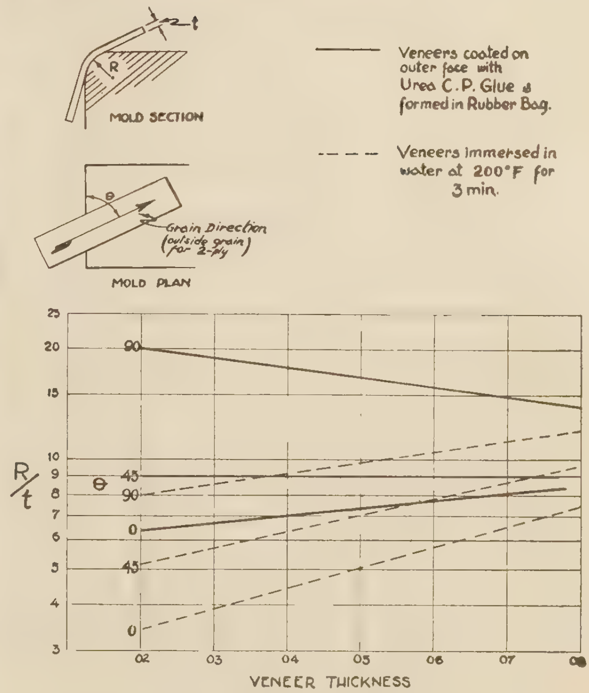

Fig. 11 Minimum-Bend Radius for Birch and Spruce Veneer and Two-Ply Plywood

The elevator is similar in construction to the stabilizer shown in Fig. 10. The skins and stiffeners are assembled on the bench and then glued to the channel-section spar and end ribs in one operation. Finally the molded leading edge is glued in place. The complete weight less mass balance is 13 lb. The structure failed at a test load of 1040 lb or 140 per cent of the design load.

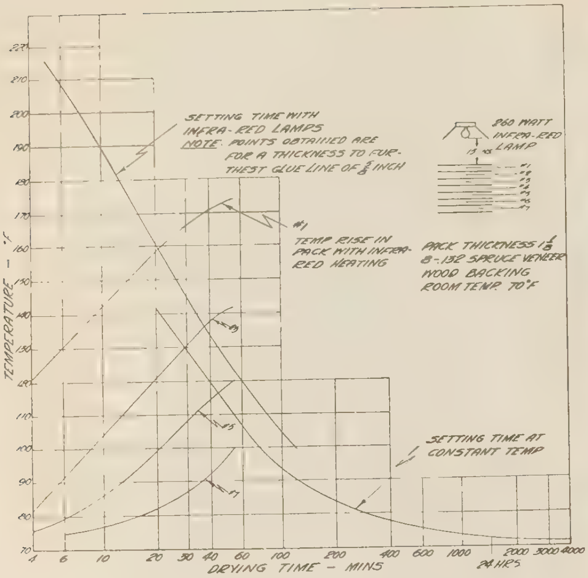

Fig. 12 Urea Cold-Press Drying Time

With the rudder, as in the foregoing components, we endeavored to make the skin the major structural element. The bottom fairing which is detachable is composed of fabric impregnated with urea-formaldehyde resin, and molded in the autoclave. Weighing 18 lb complete with mass balance, it was tested to 150 per cent of the design load without any evidence of failure.

One of the first things one needs to know in designing these structures is a safe bend radius to use with the various veneer thicknesses. Fig. 11 gives values which we consider to be a practical minimum.

Infrared lamps will greatly accelerate the drying of cold-setting resin adhesives. Fig. 12 indicates the time required to achieve a reasonable shear strength at a constant temperature. The time is very considerable at the temperatures which frequently prevail in workshops. Referring to Fig. 12, the time-temperature curves for a thick pack of veneers exposed to an infrared lamp are given. As a simple example, consider a glue line $^1/_8$ in. below the surface. During the first 8-min interval, the average temperature is roughly 120 F. At that temperature, we see from this curve the total time required would be 40 min for the glue to set. Therefore, in 8 min we obtain 20 per cent of the required cooking. During the next interval, the average temperature is about 145 F, and the total time required would be 20 min. Therefore, in 8 min, we obtain an additional 40 per cent of the required cooking. Adding the percentages to a total of 100 per cent, we obtain the required time with this lamp and distance to set a glue line $^1/_8$ in. below the surface. This curve is based on experimental results with a $^5/_8$-in. panel and shows the considerable time saving which can be realized.

The Technique of Bag Molding

No description of the bag-molding art is necessary, as it has received a good deal of attention in the literature lately. Lockheed employed the fluid-pressure idea 15 years ago to produce a number of extremely refined plywood airplanes, but the production possibilities of the method have only recently received attention. The wooden mold, rubber bag, and autoclave, as currently used, form a convenient arrangement for experimental work but suffer numerous drawbacks for large-scale production.

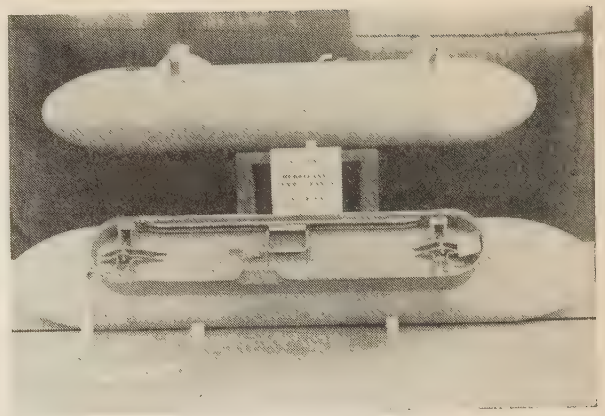

Fig. 13 Disposable Fuel Tanks for Long-Range Fighter Planes

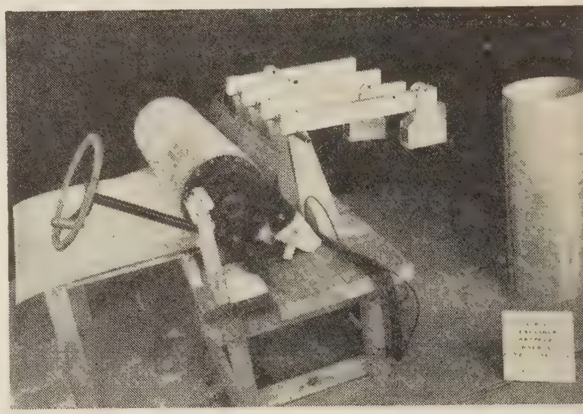

Fig. 14 Collapsible Drum for Winding Shell of Fuel Tank

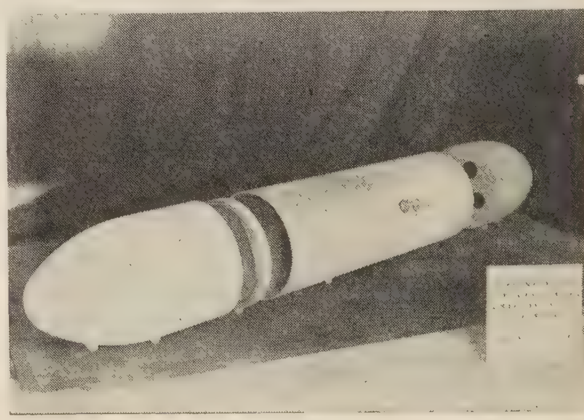

Fig. 15 Assembly of Center Section, Doubler Rings, and Laminated End Sections of Fuel Tank

Wooden molds are woefully impermanent when subjected to frequent heating and cooling cycles, a great deal of time and heat are wasted in cooking the great bulk of the mold, and its size and general awkwardness present a considerable handling problem not to mention the wear and tear on rubber bags. Alternative methods are in use with varying degrees of success, but we believe that a metal mold in the majority of production applications will quickly pay for itself. The form is metal, and internally

heated, preferably with steam. The veneer-and-glue assembly is placed on the form and stapled to wooden inserts at the edges. Over this assembly is placed a reasonably airtight blanket; it does not need to be rubber and even a few leaks do not matter. Finally a light outer shell is dropped in place and sealed. Compressed air can therefore provide the pressure independently of the temperature. We have performed a number of cookings on a unit of this type; the temperature rise is extremely rapid and the results uniformly satisfactory.

Forming Fuel Tanks to Be Jettisoned

Another plywood enterprise in which we have been engaged is a jettisonable or disposable fuel tank for a long-range fighter aircraft, Fig. 13. The machines carry two of these tanks, one under each wing, and when the fuel is consumed the tanks are dropped. Therefore cheapness and simplicity of construction are essential.

We start with a two-ply veneer which is wound on a collapsible drum, shown in Fig. 14. Metal bands are loaded with a lever weight arrangement to apply pressure. Infrared lamps supply the heat necessary to set the cold-press resin as the two-ply ribbon is wound on the drum. The nose and tail are of the same shape and are composed of 5 layers of two-ply hot-press-molded wood. Laminated rings, which also carry the loads from the attachment fittings, serve as doublers between the ends and the cylindrical portion, Fig. 15. Pipe and attachment fittings are riveted in place first, then the ends are glued on and a fairing added. The tanks are flushed with a gasoline-resistant sealing compound, pressure-tested, and with a coat of paint are ready for service.

We have performed a number of tests on these tanks but none so drastic as that which recently occurred when one laden with 100-octane fuel was accidently released from an aircraft about to land. The tank dropped a distance of 4 or 5 ft and commenced a journey across the field at some 100 mph. Although enveloped in spray, it did not ignite and despite considerable damage was still a complete unit when retrieved by the somewhat shaken ground crew.

Pressure Loss in Elbows and Duct Branches

By ANDREW VAZSONYI,[1] CAMBRIDGE, MASS.

The pressure loss in a duct branch is a function of at least six independent variables, and there is no complete set of data available for correlation. Equations [II] and [III] of this paper are derived on a theoretical and experimental basis. These equations correlate all available data with fair accuracy with the aid of four experimental curves (Figs. 2, 3, and 5). Furthermore, they permit prediction of probable pressure losses for ducts heretofore untested. Similar formulas can be derived for elbows, diffusers, contracting passages, and other configurations, e.g., Equation (1) presents losses for elbows.

Nomenclature

The following nomenclature is used in the paper:

- A = area of cross section
- d = diameter of pipe
- f = friction factor
- h = mixing loss = total loss — friction loss (given in Equation [9])
- H = total loss = $(p_1 + {}^1/_2\, \rho V_1^2) - (p_2 + {}^1/_2\, \rho\, V_2^2)$
- l = length of pipe
- p = pressure
- P = circumference of pipe
- Q = mass rate of flow
- V = velocity
- $\bar{V}$ = mean velocity based on mass rate of flow
- α, β, γ = angle of deflection in elbow or branch pipe
- α', β', γ' = effective angle of deflection of stream
- ρ = density
- τ = shear force on wall of duct
- λ = new loss coefficient
- η = diffuser efficiency
- ζ = standard loss coefficient

Introduction

When the practicing engineer is confronted with the problem of predicting pressure losses in ducts, he can find little help from the available literature. There are so many different types of ducts that one could hardly expect to chance upon the one desired. However, it is somehow expected that a duct system can be separated into different types of "elementary" ducts, and one hopes that knowing pressure losses for these elementary ducts, the total pressure loss in a duct system can be predicted by summation. Therefore it is very desirable to know pressure losses for a great variety of ducts, and indeed a vast number of experiments have been run for this purpose. Unfortunately, nothing like a convenient and practical summary of data exists. Furthermore, often the available material is somehow confused, even contradictory.

The subject of this paper is the presentation of losses for duct branches, and so let us examine this configuration in detail. What does the pressure loss depend upon in this duct system? In order to obtain any answer, we have first to specify completely the configuration considered. Let us assume that we have three circular pipes "meeting" at a point. In order to avoid flow disturbances by the entrance and exit, let us assume that all the pipes are very long. Now we have a well-defined elementary duct, and we can ask for the pressure loss in this duct. The pressure loss will obviously depend (at least) upon the following quantities: The diameters of the pipes; the respective fluid velocities; the angles of inclination of the branching pipes; roughness of the wall; and the kinematic viscosity of the fluid—altogether ten variables.

Using the continuity equation and similarity considerations, four of these variables can be eliminated and the remaining six can be listed as follows: Ratios of exit areas to inlet area; ratio of velocities in two of the pipes; the branching angles; friction factor. No more reduction in the number of variables is possible and experiments have shown that all these variables are of major importance. Hence we see that to obtain a complete survey of the losses would necessitate an enormous number of experiments. But even if all these data were known, how could they be put into a useful form for the designing engineer? The situation is somehow similar to that of the friction-factor data at the time when the Reynolds number was not yet used.

Investigators of the Hydraulic Institute of the Technical University of München have run extensive tests on duct branches (13, 28, 30, 35).[2] Nine different configurations were selected. In all of them there is a main pipe continuing ahead; the duct branch forms an angle of 30, 45, or 90 deg with the main pipe; the ratio of the diameter of the branch pipe to that of the main pipe is $\frac{15}{43}, \frac{25}{43}, \frac{43}{43}$.

The pressure loss is expressed as a fraction of the kinetic energy (per unit of mass) of the fluid in the main pipe, as is customary for friction-factor data. For the independent variable the ratio $\frac{Q_1}{Q_0}$ is used. We have replotted these curves, using the velocity ratio $\frac{V_1}{V_0}$ for the independent variable, as this appears to suit better the nature of the problem, e.g., Fig. 10 shows losses for three different duct branches, each of them having an angle of deflection of 45 deg. The data lie fairly well on one and the same curve, and so one curve replaces the three different curves given in the original papers. Figs. 11 and 12 show two further curves of the many presented by the German investigators. It can be seen that the pressure-loss coefficient varies between —10 and +45. (Negative pressure-loss coefficients indicate pressure rises.) Similar curves can be found in the original papers for the nine configurations, and also for some slightly different ducts. But let us suppose we do not have a duct with an angle of 30, 45, or 90 deg, or the area ratios do not check with those tested. The coefficients vary so much, that interpolation can be of little help.

The pressure-loss-data presentation as a fraction of the kinetic energy of the main pipe is a fully arbitrary one, and its only justification is that it works well for straight pipes. As it is certainly not satisfactory in the duct-branch case, why not use some combination of the kinetic energies of the different pipes of the duct system? The difficulty now is how to select suitable formulas.

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[1] Teaching Fellow in Mechanical Engineering, Graduate School of Engineering, Harvard University.

Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

In this paper a trial is made in this direction. First of all the pressure loss is divided into two parts, i.e., friction loss and mixing loss. The former can be determined by the knowledge of the friction factor; the latter is supposed to be (approximately) independent of the wall roughness. After some theoretical investigation, formulas are derived for the mixing loss. There are experimental coefficients in these formulas which are determined to fit the available test data. The main results are as follows:

Elbow Losses:

$$2h_{12} = V_1^2 + (2\lambda - 1)V_2^2 - 2\lambda V_1 V_2 \cos\alpha \quad \text{[I]}$$

where for circular elbow $\lambda = 0.6$, for rectangular elbow $\lambda = 0.75$.

Straight Pipe Branching Into Two Pipes:

$$2h_{01} = \lambda_1 V_0^2 + (2\lambda_1 - \lambda_2)V_1^2 - 2\lambda_2 V_0 V_1 \cos\alpha' \quad \text{[II]}$$

where λ_1, λ_2, α' are functions of α given in Figs. 2 and 3.

Two Straight Pipes Discharging Into One Pipe:

$$2h_{10} = \lambda_3 V_1^2 + V_0^2 - 2V_0\left(\frac{V_1Q_1}{Q_0}\cos\beta' + \frac{V_2Q_2}{Q_0}\cos\gamma'\right). \quad \text{[III]}$$

where λ_3 is a function of β given in Fig. 5, β' and γ' are functions of β and γ in Fig. 3.

The Energy Loss

Consider some kind of duct system. The law of conservation of energy states that

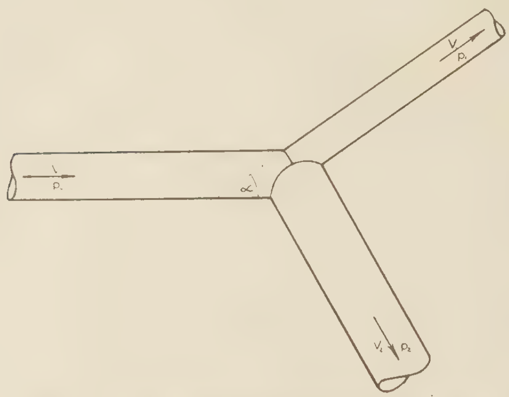

Fig. 1 Duct Branch With Separating Flow

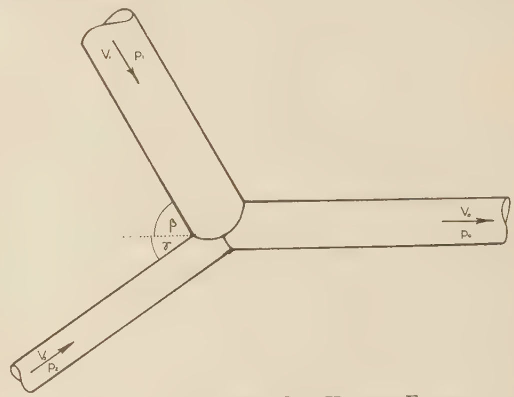

Fig. 4 Duct Branch With Uniting Flow

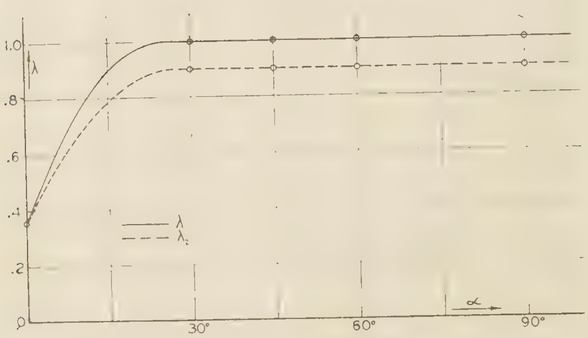

Fig. 2 Pressure Loss in Duct Branch With Separating Flow
(λ_1 and λ_2 of Equation [II], as functions of angle of deflection α.)

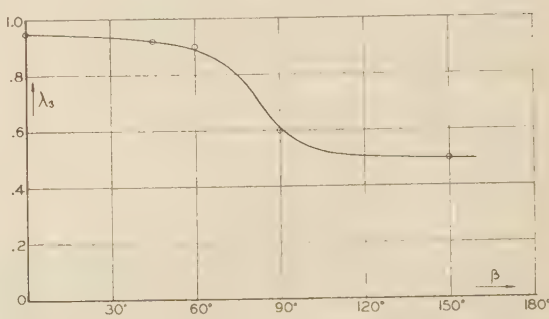

Fig. 5 Pressure Loss in Duct Branch With Uniting Flow
(λ_3 of Equation [III] as function of β the angle of deflection.)

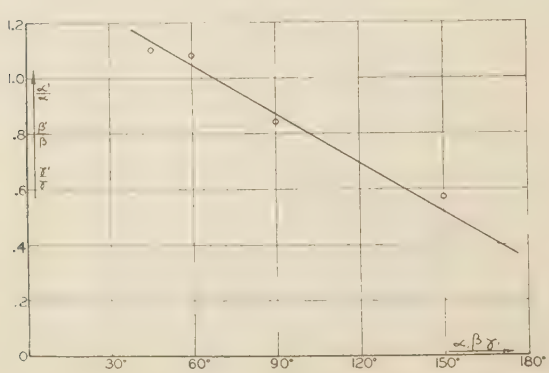

Fig. 3 Effective Angle of Deflection as Function of Angle of Deflection; Equations [II] and [III]

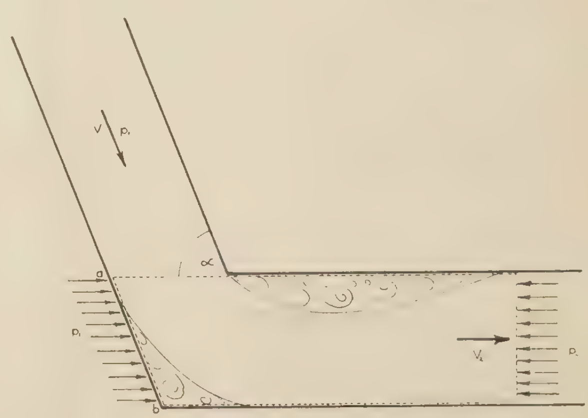

Fig. 6 Pressure Loss in a Miter Elbow
(Dotted lines enclose section to which momentum equation is applied.)

$$\int_{A_1}\left(p + \frac{1}{2}\rho V^2\right)VdA = \int_{A_2}\left(p + \frac{1}{2}\rho V^2\right)VdA + \rho HQ \ldots [1]$$

where Q is the mass rate of flow and H is the work done by the fluid per unit of mass flow. By definition, H is the loss through the duct system. In what follows, we will consider only incompressible fluids, and so we can assume that ρ is constant.

Straight Pipe

Consider the case of a straight pipe and assume that p is constant across the pipe. Then Equation [1] can be written in the form

$$\frac{p_1}{\rho} + \frac{1}{Q}\int_{A_1}\frac{1}{2}V^3dA = \frac{p_2}{\rho} + \frac{1}{Q}\int\frac{1}{2}V^3dA + H \ldots [2]$$

Now, if we further assume that we are dealing with a steady-state case, i.e., the velocity profile is everywhere the same, we get

$$\frac{p_1 - p_2}{\rho} = H \ldots [3]$$

Writing the momentum equation for the case considered, we get

$$A(p_1 - p_2) = Pl\tau \ldots [4]$$

where P is the circumference of the pipe and τ is the shear force on the wall. Introducing the friction factor

$$f = \frac{\tau}{\frac{1}{2}\rho\bar{V}^2} \ldots [5]$$

we get

$$\frac{p_1 - p_2}{\frac{1}{2}\rho\bar{V}^2} = f\frac{lP}{A} \ldots [6]$$

and for the loss

$$\frac{H}{\frac{1}{2}\bar{V}^2} = f\frac{lP}{A} \ldots [7]$$

Equations [6] and [7] show that the pressure drop along the pipe measures the energy dissipation. It is important to emphasize that this is true only in the case of a straight pipe, and under the assumption made that the velocity profiles are the same along the pipe.

More General Ducts

Let us now consider Equation [1] in a more general case, e.g., a diffuser or elbow. In order to determine the loss, it is necessary to measure the pressure and the velocity across the entrance and exit areas and to integrate Equation [1]. This is a much too elaborate procedure, and so it is usually assumed that both p and V are constant across the cross sections. This condition is sufficiently well approximated for turbulent flows by inserting a long straight pipe before and after the duct. Then Equation [1] becomes

$$p_1 + \frac{1}{2}\rho V_1^2 = p_2 + \frac{1}{2}\rho V_2^2 + H \ldots [8]$$

or the loss is given by the difference of the total pressures. It is to be emphasized that Equation [8] holds true only when both p and V are constant across the entrance and exit areas. Some investigators have used the total pressure difference as the loss for cases where these conditions did not hold, and obtained contradic-

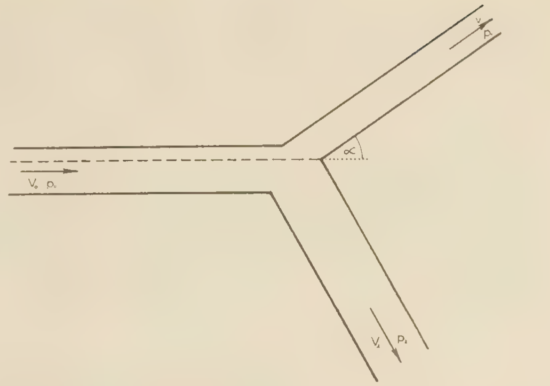

Fig. 7 Duct Branch With Separating Flow Decomposed Into Two Miter Elbows

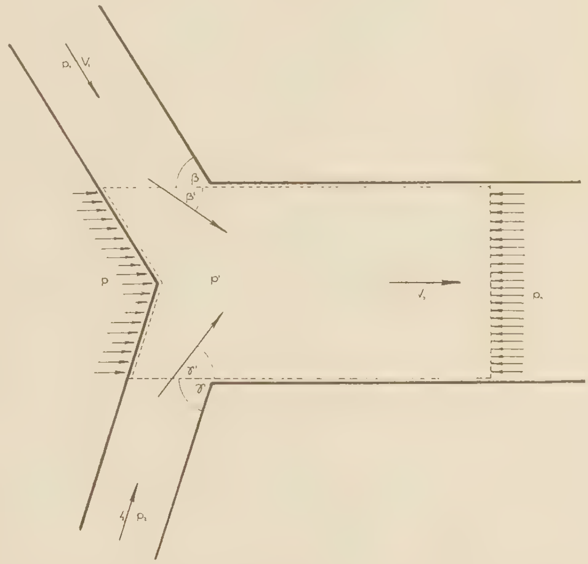

Fig. 8 Pressure Loss in Duct Branch With Uniting Flow (Dotted lines enclose section to which momentum equation is applied.)

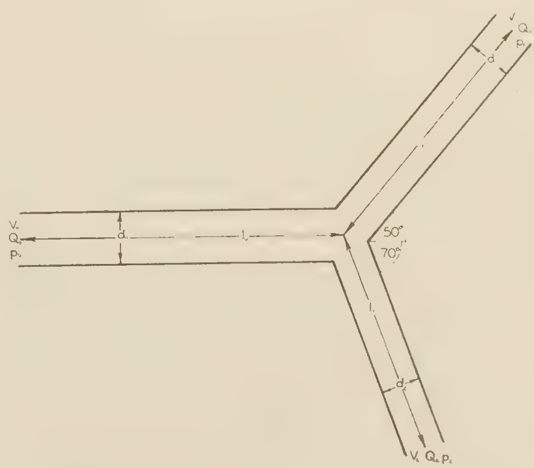

Fig. 9 Example for Computation of Pressure Loss in Circular Duct Branch With Separating Flow

tory loss coefficients. It makes considerable difference whether an elbow discharges into open air or into another pipe. In the latter case there is a recovery in the static pressure, while in the former case the losses are much higher. From now on it will be assumed that we are considering ducts for which Equation [8] can be used.

Obviously, the lengths of the inserted pipes before and after the duct are of extreme importance for the tests. Probably a pipe 50 diam in length is long enough to dissipate disturbances. However, there are examples where disturbances prevail for a distance of over 100 diam. It appears that a rotation of the flow (as for instance after a double elbow) persists for a particularly long distance.

Friction and Mixing Loss

The loss in a duct will obviously depend upon the wall roughness, besides the geometry of the duct, the velocities, etc. However, it is possible to divide the losses into two groups "the friction loss" and the "mixing loss." Consider the duct "straightened out" in such a way that the center line becomes straight but its length is unchanged. This is a rather indeterminate procedure in general but rather well defined for elbows or a system of long pipes. The loss due to wall friction in the straightened-out duct of the same center-line length is the so-called friction loss of the original duct. The remaining part will be called the mixing loss. Consider, for example, an elbow; then Equation [8] can be written in the form

$$p_1 + \frac{1}{2}\rho V_1^2 = p_2 + \frac{1}{2}\rho V_2^2 + \frac{f_1 l_1 P_1}{A_1} \cdot \frac{1}{2}\rho V_1^2 + \frac{f_2 l_2 P_2}{A_2} \cdot \frac{1}{2}\rho V_2^2 + h\rho \qquad [9]$$

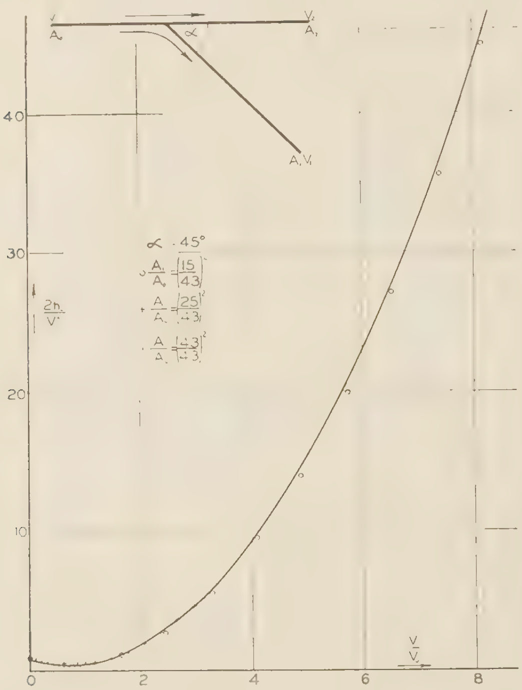

Fig. 10 Pressure Loss in Circular Duct Branch With Separating Flow, Expressed as Fraction of Kinetic Energy of Entering Fluid

(Angle of deflection 45 deg; diameter ratio d_1/d_0 = 15/43, 25/43, 43/43. Data from Bibliography (28), pp. 106, 107, 108.)

where h is now the mixing loss by definition. Experiments have shown that h does not vary too much when the wall roughness varies while, obviously, the total loss varies considerably with the roughness, both entrance and exit pipes being necessarily

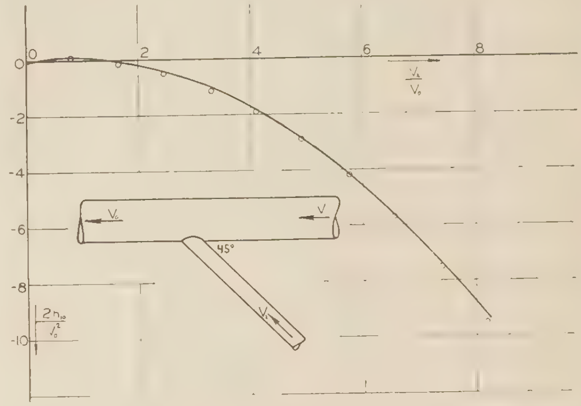

Fig. 11 Pressure Loss in Circular Duct Branch With Uniting Flow Expressed as Fraction of Kinetic Energy of Leaving Fluid

(Angle of deflection 45 deg; diameter ratio d_2/d_0 = 15/43. Data from Bibliography (28), p. 106.)

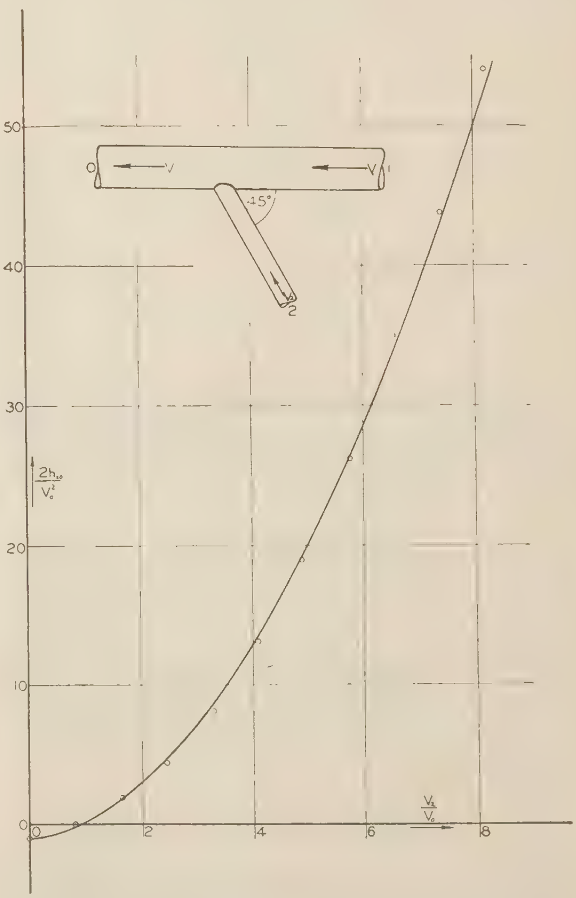

Fig. 12 Pressure Loss in Circular Duct Branch With Uniting Flow, Expressed as Fraction of Kinetic Energy of Leaving Fluid

(Angle of deflection 45 deg; diameter ratio d_2/d_0 = 15/43. Data from Bibliography (28), p. 107.)

long. Furthermore, the total loss depends also upon the lengths of the exit and entrance pipes.

Miter Elbow

In order to develop an appropriate formula for this duct, let us consider first qualitatively what happens to the fluid in this passage.

Let us assume that the angle of deflection is small. The elbow acts as a diffuser and little turbulence occurs in the passage. Diffuser losses have been successfully correlated with the expression

$$2h = (1 - \eta)(V_2^2 - V_1^2) \quad [10]$$

where η is the efficiency of the diffuser, and it might be expected that elbows with a small angle of turn can be correlated with the same formula.

On the other hand, let us consider the case when the angle of turn is large. In this case, considerable turbulence occurs in the fluid and we have a situation similar to that of the suddenly enlarging pipe. But it is known that for this case the so-called Carnot formula, which is based on the momentum equation, works very well (33). So we are going to try to use the momentum equation for an elbow. The pressure in the dead-water region is unknown, and so we write the momentum equation in the direction of the leaving fluid, eliminating thereby this unknown pressure. We assume, furthermore, that the pressure on the wall $a - b$ (Fig. 6) equals p_1, the initial pressure. The momentum equation yields

$$p_1 A_2 + \rho V_1^2 A_1 \cos \alpha = p_2 A_2 + \rho V_2^2 A_2 \quad [11]$$

where it is assumed that the fluid enters under an average angle α and that the effect of wall friction is negligible. Combining

$$p_1 + \frac{1}{2} \rho V_1^2 = p_2 + \frac{1}{2} \rho V_2^2 + h\rho \quad [12]$$

with Equation [11], we get for the mixing loss

$$2h = V_1^2 + V_2^2 - 2V_1V_2 \cos \alpha \quad [13]$$

This equation is not to be expected to be exact because of our assumptions. So we complete it with an experimental coefficient

$$2h = \xi(V_1^2 + V_2^2 - 2V_1V_2 \cos \alpha) \quad [14]$$

Now we have two different expressions for the elbow loss, Equations [10] and [14], and we combine the two expressions to get

$$2h = \xi(V_1^2 + V_2^2 - 2V_1V_2 \cos \alpha) + (1 - \eta)(V_2^2 - V_1^2) \quad [15]$$

or

$$2h = \lambda_4 V_1^2 - 2\lambda V_1 V_2 \cos \alpha + (2\lambda - \lambda_4) V_2^2 \quad [16]$$

Now let us consider an elbow with an extremely narrow inlet pipe, where $V_2 = 0$. Obviously, all the entering kinetic energy will be dissipated and so we get $\lambda_4 = 1$. Hence the final formula is

$$2h_{12} = V_1^2 - 2\lambda V_1 V_2 \cos \alpha + (2\lambda - 1) V_2^2 \quad [17]$$

Fig. 13 shows losses for a circular miter elbow with equal entrance and exit cross sections. Putting $\lambda = 0.6$ in the formula, a curve results which fits the data well. Fig. 14 shows losses for rectangular elbows. The experimental points are too badly scattered to get a reliable estimate of λ. But by considering the limiting case when fluid is discharged from a large container into a pipe, it appears that $\lambda = 0.75$ is a good value.

Pipe Branches With Separating Flow

Consider the branch pipe composed of two elbows, as shown in Fig. 7. Then we can apply our elbow formula, Equation [16], to each of these elbows. We get for the mixing loss in the branch pipe

$$2h_{01} = \lambda_1 V_0^2 + (2\lambda_1 - \lambda_2) V_1^2 - 2\lambda_0 V_0 V_1 \cos \alpha' \quad [18]$$

where we assume that the fluid leaves the branch pipe through opening 1 at an average angle of α'. This angle will be called the effective angle of turn. In order to determine λ_1 we consider experiments when all the fluid is discharged through opening 2. Then $V_1 = 0$ and an experimental curve for λ_1 can be drawn. This curve is plotted in Fig. 2. Then by drawing best fitting

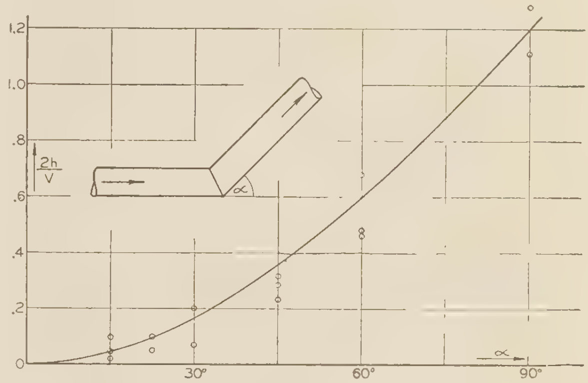

Fig. 13 Pressure Loss in Circular Elbow, Expressed as Fraction of Kinetic Energy of Fluid
(Inlet diameter equals exit diameter. Data from Bibliography (34).)

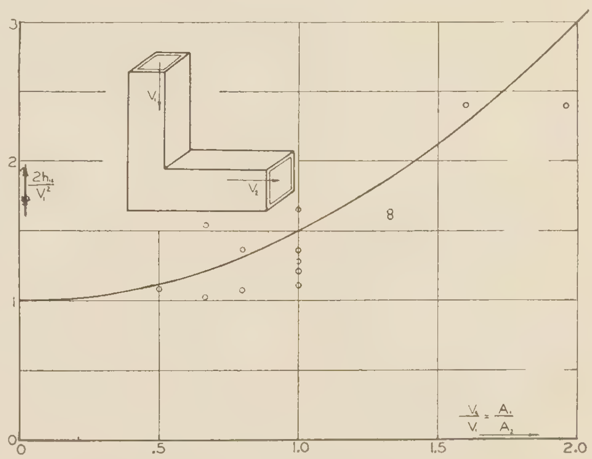

Fig. 14 Pressure Loss in Various Rectangular Elbows With 90-deg Angle of Deflection, Expressed as Fraction of Kinetic Energy of Entering Fluid
(Data from Bibliography (10), (25), (34).)

curves, λ_2 and the effective angle of turn are determined as functions of α (Figs. 2 and 3).

Pipe Branch With Uniting Flow

The derivation of the loss formula for this case will be briefly indicated. Let p' be the "average" pressure in the section of the duct where the two entering fluid streams meet (see Fig. 8). The inlet pipes act as diffusers, and so it is to be expected that the values of p' can be correlated with a formula similar to Equation [10]. This is indeed the case, and the unknown coefficients can be determined from the special experimental cases when $V_1 = 0$ or $V_2 = 0$. After p' has been determined in this fashion,

the momentum equation can be used in the direction of the leaving flow as indicated in Fig. 8. The final result is Equation [III] of the introduction.

Example

Problem. Determine the pressure loss with uniting flow through the duct in Fig. 9, assuming that the rate of flow through pipes 1 and 2 is given as Q_1 and Q_2.

Solution. First we determine the velocities in the pipes

$$V_0 = \frac{4(Q_1 + Q_2)}{\pi d_0^2}, \quad V_1 = \frac{4Q_1}{\pi d_1^2}, \quad V_2 = \frac{4Q_2}{\pi d_2^2} \ldots\ldots [19]$$

Now, knowing the velocities, we can compute the Reynolds number and look up the friction factors corresponding to the given wall roughness, and determine the friction loss as follows

$$\text{Friction loss from 0 to 1} = \frac{4f_0l_0}{d_0} + \frac{4f_1l_1}{d_1} \ldots\ldots [20a]$$

$$\text{Friction loss from 0 to 2} = \frac{4f_0l_0}{d_0} + \frac{4f_2l_2}{d_2} \ldots\ldots [20b]$$

In order to get the mixing loss, we have to find the effective angle of turn and the λ_1, λ_2 values corresponding to the angles 50 and 70 deg. We find from Figs. 2 and 3

when $\alpha = 50$ deg, then $\alpha' = 55$ deg, $\lambda_1 = 1, \lambda_2 = 0.9, 2\lambda_1 - \lambda_2 = .2$

when $\alpha = 70$ deg, then $\alpha' = 68.5$ deg, $\lambda_1 = 1, \lambda_2 = 0.9, 2\lambda_1 - \lambda_2 = .2$

Putting these values into Equation [II], we get for the mixing loss

$$2h_{01} = V_0^2 + 0.2V_1^2 - 1.8V_0V_1 \cos 55 \text{ deg} \ldots\ldots\ldots\ldots [21a]$$

$$2h_{02} = V_0^2 + 0.2V_2^2 - 1.8V_0V_2 \cos 68.5 \text{ deg} \ldots\ldots\ldots [21b]$$

From Equations [20] and [21], we get the total loss by addition.

Conclusions

The results and methods of this paper are believed useful in the following respects:

1 The formulas presented permit the calculation of losses in many duct branches by reasonably accurate correlation of sparse test data.

2 The number of curves required to present experimental data can be greatly reduced.

3 The method is applicable to many other configurations, including diffusers.

4 The formulas obtained by the method presented point the way to necessary tests in future test programs.

Acknowledgment

The author wishes to expresses his appreciation to Prof. H. W. Emmons of Harvard University for the suggestion of the subject and for his invaluable aid in the work.

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Discussion

Maurice Martin.[6] The author should be commended for attempting to correlate the results of previous experiments. One of the outstanding features of previous work in this field has been the divergence of test results. This is due, of course, to the many unknown variables mentioned by the author. The writer has found that the best method for determining losses is to make and test typical duct layouts utilizing workmanship and material available to the engineer in actual construction. If this proves too cumbersome a method, the writer has found remarkable agreements in present literature on the losses in elbows and branches, i.e., "Fan Engineering" of the Buffalo Forge Company.

The writer has found that in Dr. Vazsonyi's attempt to apply the equation developed for elbows to losses in branches, he has failed to take into account the fact that the amount of air that will turn from the main and enter the branch will depend almost entirely on the resistance or back pressure set up by the entire length of branch as compared to the resistance of the remaining portion of main duct. One cannot, therefore, consider an imaginary splitter n the main to take off a definite portion of air. This is especially true in consideration of the two branches at the end of the main.

Most modern experiments on the subject agree that the losses in both elbows and branches can be expressed best in terms of the velocity or velocity pressure in the duct and, workmanship and other things being equal, for geometrically similar elbows and branches will depend directly on this pressure. When approached from this viewpoint I think that the author will find little disagreement in the more recent experiments.

E. O. Bergman.[7] Several tests of branching pipes have been made by the U. S. Bureau of Reclamation to check and extend the work of Professor Thoma. These have been published in Boulder Canyon Projects Final Reports; Part VI, Hydraulic Investigations; Bulletin 2, Model Studies of Penstocks and Outlet Works.

Author's Closure

Mr. Martin is right in stating that in certain cases "the amount of air that will turn from the main and enter the branch will depend almost entirely on the resistance or back pressure set up by the entire length of branch, etc." However, Equation [9] of the paper takes care of the resistance of the branch pipe, whereas the "mixing-loss" formulas take care only of the additional "nonfrictional" losses. This is exactly the reason why it was necessary to separate the friction loss from the mixing loss.

Mr. Martin is also right in stating that many experimenters did not pay due attention to the great number of variables and thus received contradictory results. However, it is difficult to see how such experiments could be made satisfactory by introducing new variables.

The reports quoted by Mr. Bergman were not known to the author.

[6] Assistant Naval Architect, Navy Yard, Brooklyn, N. Y. Jun. A.S.M.E.

[7] Technical Advisor, C. F. Braun & Co., Alhambra, Calif. Mem. A.S.M.E.

Laboratory and Field Tests on Coal-in-Oil Fuels[1]

By J. F. BARKLEY,[2] A. B. HERSBERGER,[3] and L. R. BURDICK[4]

It has been determined from extensive laboratory and field tests on the stability of coal suspensions in fuel oil that each installation must be given individual consideration if satisfactory results are to be obtained. However, no particular difficulty in devising equipment or operating routine for efficient use need be experienced. A sufficiently stable suspension of coal in oil for practical purposes can be prepared by dispersing 40 per cent coal by weight ground from 98 to 99 per cent through a 230-mesh screen in heavy fuel oil. Careful control of storage and handling temperatures is required for most successful operation. Combustion efficiency is likely to be slightly less than that of fuel oil. The best opportunity for the successful use of coal-in-oil suspension is in the larger plants with better operating personnel and more readily accessible equipment.

Introduction

The following report covers laboratory studies on the stability of coal suspensions in oil, the mixing of pulverized coal and oil, and field tests on these mixtures in typical oil-burning equipment. This work was undertaken to obtain needed practicable information not found in technical literature, a review of which was given by Schroeder[5] in 1942. The work is the result of a joint effort by the Federal Bureau of Mines and The Atlantic Refining Company of Philadelphia, Pa. The laboratory work and preparation of the fuels were done by The Atlantic Refining Company. The main series of field tests made at The Atlantic Refining Company's Philadelphia Plant and additional tests made at the Cambria Silk Hosiery Company and the National Airoil Burner Company, both in Philadelphia, were conducted by the Bureau of Mines.

Laboratory Studies

In the laboratory, stabilities of coal-in-oil suspensions were studied by measuring the change in the center of gravity of the suspensions by means of a pendulum arrangement. This method was first used by Manning and Taylor.[6] The results are reported as the percentage of the coal settled, based on 100 per cent settled, as determined in an alcohol-water mixture. Other methods were also used to check the results obtained in this manner. Details of the procedures are given in the Appendix.

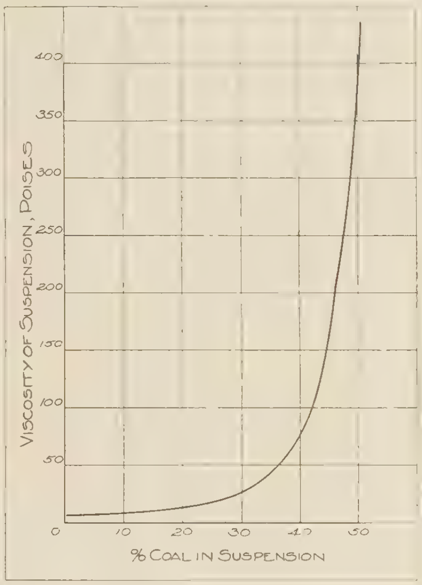

Fig. 1 Viscosity of Suspensions: Curve Showing Effect of Percentage of Coal in Mixture on Viscosity at 77 F
(Coal No. SC-2, minus 230 mesh; No. 6 fuel oil.)

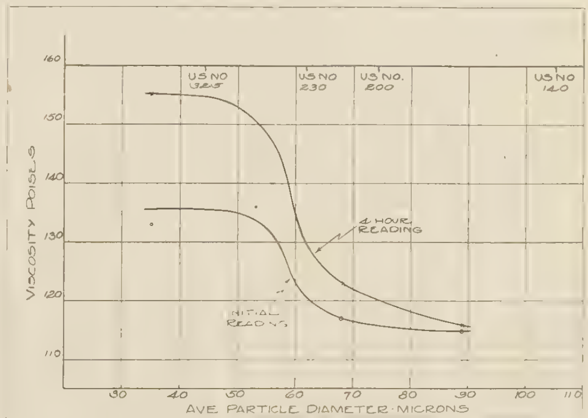

Fig. 2 Viscosity of Suspensions: Curves Showing Effect of Particle Size of Coal in Mixture on Viscosity at 77 F
(Coal No. SC-1, 40 per cent by weight; No. 6 fuel oil, 60 per cent by weight.)

The majority of the samples were mixed by vigorous stirring under about 10-mm vacuum. This was done to avoid including air in the samples which would eventually interfere with the pendulum readings.

The absolute viscosities, reported in Figs. 1 and 2, were obtained by means of a Brookfield Synchro-lectric viscometer which had been calibrated for average rates of shear and shearing stresses. The coal used in most of the experimental work was

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[1] Published by permission of the Director, Bureau of Mines, U. S. Department of the Interior.

[2] Chief, Division of Solid Fuels Utilization for War, Bureau of Mines, Washington, D. C. Mem. A.S.M.E.

[3] Research and Development Department, The Atlantic Refining Company, Philadelphia, Pa.

[4] Senior Fuel Engineer, Division of Solid Fuels Utilization for War, Bureau of Mines, Washington, D. C. Mem. A. S. M. E.

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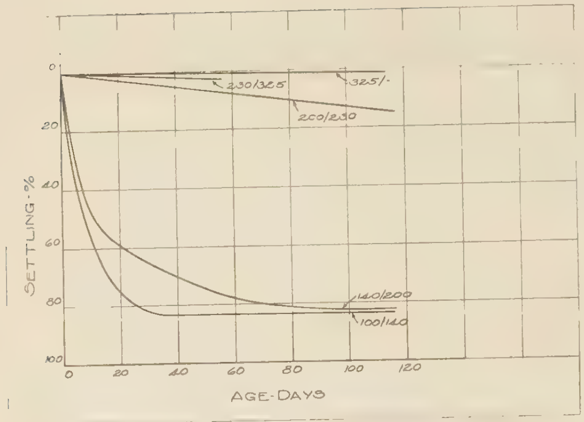

Fig. 3 Stability of Suspensions: Curves Showing Effect of Particle Size of Coal in Mixture on Stability at 77 F
(Coal No. SC-1, 40 per cent by weight; No. 6 fuel oil, 60 per cent by weight.)

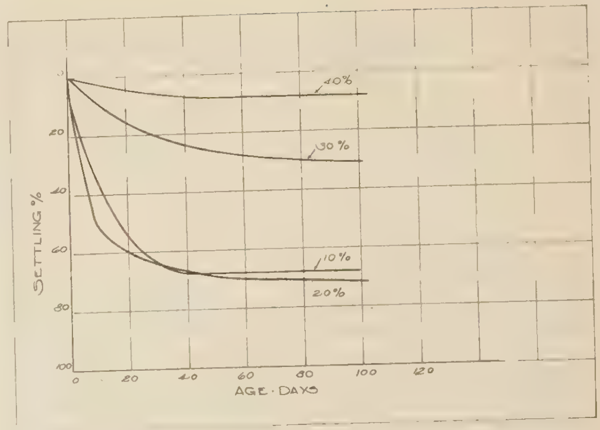

Fig. 6 Stability of Suspensions: Curves Showing Effect of Percentage of Coal in Mixture on Stability at 77 F
(Coal No. SC-2, minus 230 mesh; No. 6 fuel oil.)

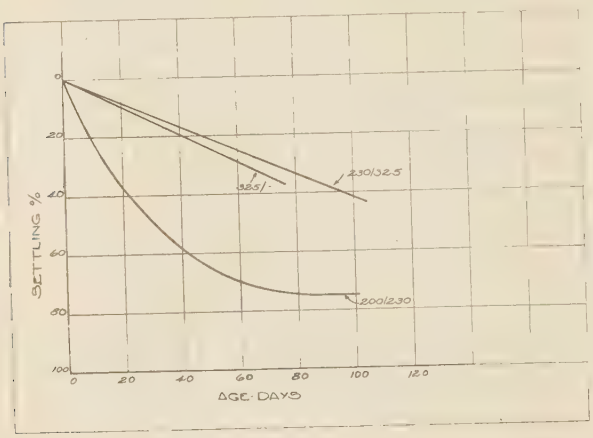

Fig. 4 Stability of Suspensions: Curves Showing Effect of Particle Size of Coal in Mixture on Stability at 120 F
(Coal No. SC-1, 40 per cent by weight; No. 6 fuel oil, 60 per cent by weight.)

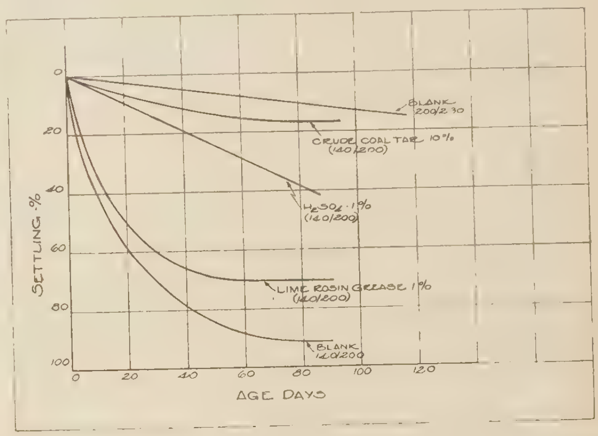

Fig. 7 Stability of Suspensions: Curves Showing Comparisons as to Stability at 77 F, When Various Stabilizing Materials Were Added to Coal-in-Oil Mixtures
(Coal No. SC-2, 40 per cent by weight; No. 6 fuel oil, 60 per cent by weight.)

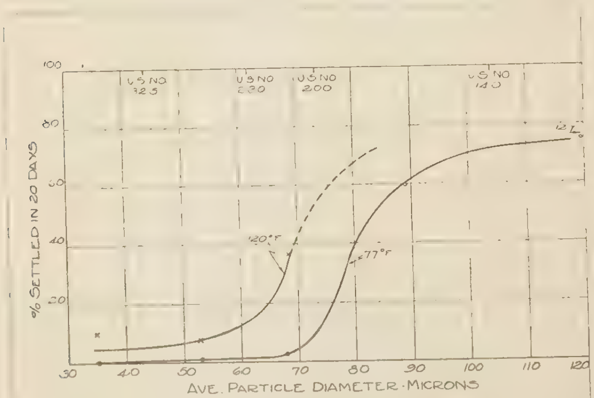

Fig. 5 Stability of Suspensions: Curves Showing Effect of Particle Size of Coal in Mixture on Amount of Settling in 20 Days
(Coal No. SC-1, 40 per cent by weight; No. 6 fuel oil, 60 per cent by weight.)

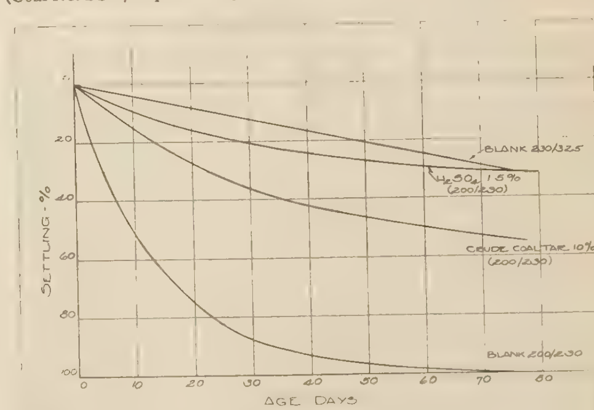

Fig. 8 Stability of Suspensions: Curves Showing Comparisons as to Stability at 120 F, When Various Stabilizing Materials Were Added to Coal-in-Oil Mixtures
(Coal No. SC-2, 40 per cent by weight; No. 6 fuel oil, 60 per cent by weight.)

screened to narrow size ranges. The data presented in Figs. 1 to 8, inclusive, are representative of a large number of experiments. The properties of the bituminous coal and the oil used in this particular set of experiments are given in Table 1. The oil was a typical refinery product containing both straight-run and cracked components.

From consideration of elementary formulas, relating settling

TABLE 1 PHYSICAL PROPERTIES OF COALS AND OIL USED IN LABORATORY EXPERIMENTAL STABILITY STUDIES

Coal	No. SC-1	No. SC-2	No. 6 Fuel oil	
Specific gravity at 60 F	1.29	1.34	A.P.I. gravity at 60 F	14.6
Volatile, per cent	33.1	22.1	Viscosity at 122 F	56 Furol
Ash, per cent	3.5	4.5	Viscosity at 210 F	86 Saybolt
Moisture, per cent	0.7	0.4		
Fixed carbon, per cent	62.7	73.4		
Sulphur, per cent	0.78	1.19		

velocities to particle size, gravity differential between the particles, and the liquid medium and its viscosity, it was evident that the best results would be obtained by using bituminous coals (low specific gravity), and oils of high specific gravity and high viscosity. It was also evident that suspensions of high coal content would be more stable than those of low content because of hindered settling.

The actual conditions for settling in concentrated suspensions are complex. Such suspensions develop rigidity for high solids content. Coal-in-oil suspensions have been shown to develop this rigidity[7] in the neighborhood of 38 volume per cent of coal.

A number of experiments were run to determine the effect of coal concentration on the viscosity of the coal-in-oil suspension. For finely divided coals in heavy fuel oils, the viscosities of the suspension were found to rise markedly in the region of 40 to 50 per cent of coal by weight, such results being in general agreement with those reported by others. This rapid rise in viscosity in the range of 40 to 50 per cent of coal was found to be relatively independent of the viscosity of the oil and the temperature of the measurement. These suspensions were also found to be non-Newtonian, i.e., the viscosity varied with the rate of shear, this deviation from viscous flow resulting from the development of structure in the suspension. Fig. 1 shows a typical viscosity-per cent coal curve.

If it is assumed that the rigidity in concentrated coal-in-oil suspensions results from the particles touching each other, it would appear that the smaller the diameter of the particle, i.e., the larger the surface area per unit concentration of coal, the greater the interference between particles. This should be evidenced by the effect of particle size on viscosity for concentrated suspensions, other variables being constant. Fig. 2 shows the effect of particle size of the coal on the viscosity of a 40 per cent by weight suspension of coal in oil. It is seen that a rapid increase in viscosity occurs between the coal sizes of 200/230 and 230/325 mesh (U. S. Standard screen). This rapid rise in viscosity and then a flattening out may be regarded as the result of a critical increase in surface area where the interference between particles has become large. The viscosity of these suspensions was measured fresh and after aging. The positions of the two curves show that the suspensions are thixotropic, i.e., the viscosities decrease with agitation. This property is more pronounced with the smaller-size particles.

Determinations of the stability of 40 per cent by weight suspensions of bituminous coal in oil were made in which the particle size of the coal and the temperature were varied. The effect of these variables is shown graphically in Figs. 3 and 4. The suspensions are relatively stable at 77 F when the maximum particle size is not larger than 200 mesh. At 120 F a satisfactory degree of stability is obtained if the maximum particle size does not exceed 230 mesh. Fig. 5 is a plot of percentage settled in 20 days against average particle size.

It is to be noted that the marked improvement in stability occurring with decreasing particle size falls in approximately the same particle-size range as the rapid rise in viscosity occurs, as shown in Fig. 2.

[7] "Laminar Flow of Oil-Coal Suspensions," by F. J. Gradishar, W. L. Faith, and J. E. Hedrick, Trans. American Institute of Chemical Engineers, vol. 39, April 25, 1943, pp. 201–222.

The relation of the stability of the suspension to the coal concentration was determined for minus-230-mesh bituminous coal. These results are shown in Fig. 6. No worth-while degree of stability is obtained until a coal concentration of approximately 40 per cent by weight is used.

The effect of physical properties of the oil and coal on stability of suspensions was also found to be important. Oils with either low viscosity or low specific gravity, in general, gave less stable suspensions. No marked differences in the stability of suspension of bituminous coals of slightly different properties were observed, although the range of bituminous coals studied was narrow. One experiment was run on a sample of anthracite which proved to be considerably less stable than a similar suspension containing a bituminous coal. Data illustrating these facts appear in Table 2.

TABLE 2 EFFECT OF PHYSICAL PROPERTIES OF COAL AND OIL ON STABILITY OF SUSPENSION[a]

Coal: Type	Coal: Specific gravity at 60 F	Coal: Mesh	Oil: A.P.I. gravity at 60 F	Oil: Furol viscosity at 122 F	Stability, 10 per cent settled at 120 F, days
Bituminous	1.29	230/325	14.6	56	23
Bituminous	1.29	230/325	..	29	3
Anthracite	1.56	minus 230	12.7	64	5
Bituminous	1.32	99 per cent minus 230	14.4	66	12
Bituminous	1.32	99 per cent minus 230	17.5	89	6

[a] Coal by weight, 40 per cent.

TABLE 3 EFFECT OF OVERSIZE PARTICLES ON STABILITY OF 40 PER CENT SUSPENSION OF COAL IN OIL BY WEIGHT

Size of oversize coal	Oversize coal, per cent (based on coal)	Stability 10 per cent settled at 120 F, days
Blank—230/325-mesh bituminous coal in No. 6 fuel oil	..	23
100/140	5	14
6/100	5	5

Laboratory samples were mixed by vigorous stirring under about 10-mm vacuum; other methods of mixing were also tried, such as passing the mixture through a homogenizer and ball-milling. Neither of these methods gave suspensions of better stability than those obtained by vigorous agitation of the mixture.

The coal used in the laboratory experiments was screened to a narrow range of sizes. From a practical point of view the effect of the maximum-size coal particles on the stability of the suspensions is important. The presence of small quantities of oversize particles was found to decrease the stability of the suspensions; data are shown in Table 3.

Figs. 7 and 8 show some stability tests on coal-in-oil suspensions containing stabilizers. Tests were conducted on 140/200 U. S. Standard-mesh coal at 77 F, and on 200/230-mesh coal at 120 F. Lime-resin soap (1 per cent), claimed to be an excellent stabilizer, was found to be very poor. Of all the stabilizers studied, the addition of 1 to 1.5 per cent of sulphuric acid to the coal surfaces before mixing with the oil was found to give the best results. Coal tar when used as 10 per cent of the oil mixture showed a stabilizing action. However, even these two materials, which were considerably better than any others of a large number tried, were not as effective as additional grinding of the coal, say, from minus-200 to minus-230 mesh. Also, both sulphuric acid and 10 per cent coal tar have practical disadvantages—sulphuric acid because of its probable corrosive action and coal tar because of the large quantity required.

If the criteria, from a stability and handling viewpoint, of a satisfactory coal-in-oil fuel are (1) high degree of stability at normal temperature, (2) sufficient stability at elevated temperatures to permit handling in conventional oil-burning equipment and (3) reasonable viscosity, the results of this work point

toward a 40 per cent by weight suspension of minus-230-mesh bituminous coal dispersed in No. 6 fuel oil. For practical purposes, the coal-specification size could be set at 98–99 per cent passing a 230-mesh screen and the limiting oil properties at 12 to 14.8 A.P.I. gravity at 60 F, and 55–70 Furol viscosity at 122 F. The laboratory results also indicate that the coal can be dispersed in the oil by relatively simple methods.

Tests in Typical Oil-Burning Plants

To obtain information on the operating problems involved in handling and burning coal-in-oil mixtures in typical oil-burning plants, tests were made at three different plants. From the laboratory information, it was decided to experiment with fuel of about 40 per cent coal by weight. Four batches of fuel were prepared, one batch having coal of a fineness of about 88 per cent through a 230-mesh A.S.T.M. (U. S.) Standard screen, one of about 95 per cent, and two of about 99 per cent. Coal could be ground to this fineness in present standard commercial equipment. As far as the authors know, no equipment is available at present that will, from a practical standpoint, continuously and satisfactorily produce coal of, say, the extreme fineness of minus 325 mesh. Table 4 gives data on the coal and oil used in the mixtures. In addition, the Appendix gives the distillation results, Bureau of Mines, Hempel method, obtained with the oil used in batch 3.

TABLE 4 DATA ON COAL, OIL, AND RESULTING COAL-IN-OIL FUELS

Batch no.	1	2	3	4
Coal, approximate percentage by weight in mixture	39.4	39.7	39.5	40.7
Proximate analysis:				
Moisture, per cent	1.7	1.5	1.3	1.4
Volatile, dry basis, per cent	35.5	35.9	37.0	36.6
Fixed carbon, dry basis, per cent	58.4	56.3	58.6	58.9
Ash, dry basis, per cent	6.1	7.8	4.4	4.5
Sulphur, dry basis, per cent	0.8	0.8	0.7	0.8
Btu, dry basis	14310	13990	14540	14520
Ash, initial deformation, deg F	2300	2230	2540	2570
Ash, softening temperature, deg F	2410	2300	2660	2690
Ash, fluid temperature, deg F	2720	2770	2910	2850
Ultimate analysis, dry basis:				
Carbon, per cent	79.9	78.0	81.5	81.1
Hydrogen, per cent	5.3	5.1	5.4	5.4
Oxygen, per cent	6.3	6.9	6.5	6.7
Nitrogen, per cent	1.6	1.4	1.5	1.5
Sulphur, per cent	0.8	0.8	0.7	0.8
Ash, per cent	6.1	7.8	4.4	4.5
Fineness, approximate, U. S. Std sieves:				
Retained on No. 100 sieve, per cent	0.3	0.02	0	0
Retained on No. 200 sieve, passing No. 100 sieve, per cent	6.5	2.68	0.3	0.2
Retained on No. 230 sieve, passing No. 200 sieve, per cent	5.7	2.8	0.3	0.2
Retained on No. 325 sieve, passing No. 230 sieve, per cent	12.1	8.4	6.4	6.4
Passing No. 325 sieve, per cent	75.4	86.1	93.0	93.2
Total, per cent	100.0	100.00	100.0	100.0
Oil, approximate percentage by weight in mixture	60.6	60.3	60.5	59.3
Ultimate analysis:				
Carbon, per cent	86.6	86.4	86.9	87.0
Hydrogen, per cent	10.6	10.6	10.7	10.8
Oxygen, per cent	1.0	0.9	0.5	0.4
Nitrogen, per cent	0.2	0.2	0.2	0.3
Sulphur, per cent	1.5	1.8	1.6	1.2
Ash, per cent	0.1	0.1	0.1	0.3
Btu	18320	18290	18440	18360
Specific gravity	0.980	0.980	0.979	0.977
Water content, per cent	0.20	0.18	Trace	0.2
Sediment by extraction, per cent	0.60	0.12	0.21	2.3
Pour point, deg F	30	40	35	50
Flash point,[a] deg F	270	265	275	275
Aniline point	Too dark	Too dark	Too dark	Too dark
Viscosity:				
Saybolt Furol at 70 F, sec	585.4	760.5	580.2	1119.0
Saybolt Furol at 100 F, sec	128.7	154.3	121.1	189.3
Saybolt Furol at 130 F, sec	46.0	51.9	44.7	56.7
Saybolt Universal at 170 F, sec	165.2 (17)	173.5 (17)	151.4 (15)	189.7 (19)
Saybolt Universal at 210 F, sec	82.6 (8)	88.3 (9)	81.3 (8)	93.1 (9)
Coal-in-oil:				
Viscosity				
Saybolt Furol at 70 F, sec	3368	5854		7181
Saybolt Furol at 100 F, sec	642	1015		867
Saybolt Furol at 130 F, sec	211	343		273
Saybolt Universal at 170 F, sec	734 (73)	1074 (107)		934 (93)
Saybolt Universal at 210 F, sec	327 (33)	477 (48)		426 (43)
Pour point, deg F				60

[a] Cleveland open cup.

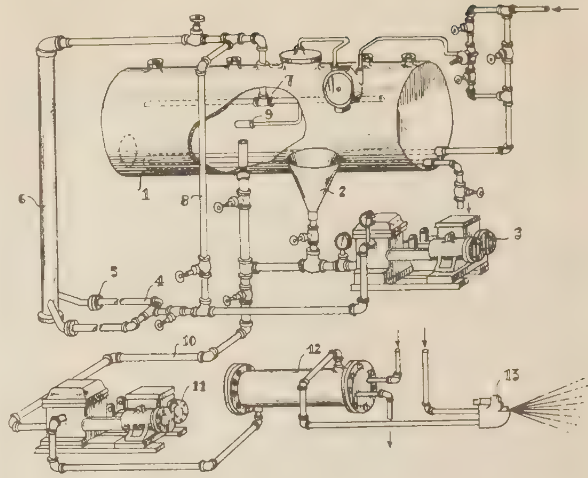

Fig. 9 Test Assembly Showing General Arrangement of Mixing Device and Oil-Burning Equipment

1 Steam coil
2 Hopper, powdered-coal intake (approx 2.5 ft diam)
3 Duplex double-acting pump:
Steam end, 7.5 in. ID
Liquid end, 7.5 in. ID
Stroke, 10 in.
4 Split flow
5 Knothole mixer (1 in.)
6 Riser
7 Spreader
8 By-pass
9 Temperature controller
10 Feed line to burner system
11 Reciprocating feed pump
12 Preheater
13 Steam-atomizing burner (1/2 in. Best)

As indicated by the laboratory results, it was found in the field that the mixing of the coal and oil needed for the tests did not present much difficulty. A simple batch-mixing device shown in the upper part of Fig. 9 was used. The fuel oil required for a batch, (about 5000 to 7000 gal), was put in the storage tank and then pumped from the tank suction line by an ordinary reciprocating pump. The oil temperatures used varied from about 75 to 120 F, the chosen temperature being maintained by steam coils in the tank. In the suction line, mounted on a tee and valve, was a funnel kept filled with pulverized coal. The coal under hand valve control was allowed to flow slowly into the stream of oil on its way to the pump. The mixture was discharged by the pump through two horizontal pipes, each having an orifice near the end. After passing the orifices, the two streams of fuel met at the base of a vertical pipe and then passed on to the storage tank. The fuel was spilled into the tank from two outlets, one at each end of the tank. The coal was fed into the oil at the rate of about 1500 lb per hr, and the fuel was continually circulated at a rate of about 9000 gph.

After the required amount of coal was fed for the batch, the fuel was circulated for about 12 hr. Laboratory tests of samples of fuel taken at various places throughout the tank showed that the mixture was uniform. No further recirculating of the fuel was done during burning tests, the mixing device being valved off from the suction line.

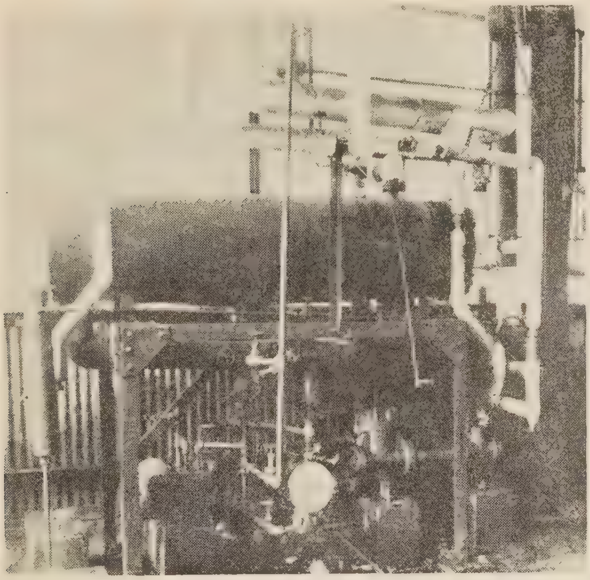

FIG. 10 HEATER-PUMP SET USED

Putting coal into oil produces a fuel having characteristics different from those of oil. The mixture used was about 10 per cent heavier in weight per unit volume than oil; it was about 4 to 8 times as viscous depending upon the temperature; it was much more abrasive; any settling out of the coal resulted in paste-like accumulations in various parts of the equipment, such as tanks, heaters, and valves; the ash from the coal was set free in the furnace.

TESTS AT ATLANTIC REFINING COMPANY BOILER PLANT

The first and main series of tests was made at one of the boiler plants of the Atlantic Refining Company. The boiler-plant equipment was typical of that of many oil-burning plants. The suction line from the outside above-ground oil storage tank ran underground about 100 ft to the heater-pump set. A steam line, that helped maintain fuel temperatures to the pump, paralleled the oil suction line. Fig. 10 shows the heater-pump set. One pump was a reciprocating and the other a rotary. The fuel was heated to about 160 to 200 F at the set and was pumped about 110 ft to the boiler. A standard Best steam-atomizing burner was used, as shown in Fig. 11. The fuel reaches a horizontal slot at the burner tip through the lower pipe; the steam reaches a vertical slot through the upper pipe and sweeps the fuel into the furnace. The fuel and steam were under hand control.

The boiler was a 151-hp, long-drum, straight water-tube Bab-

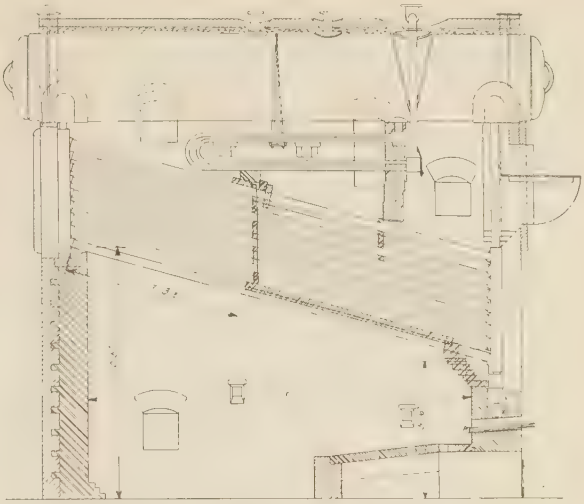

FIG. 12 GENERAL LAYOUT OF BOILER AND FURNACE

cock & Wilcox boiler, 8 tubes wide and 9 tubes high. Fig. 12 shows the boiler layout. The boiler was first operated on oil to establish normal operating characteristics. It was then operated continuously on coal-in-oil fuel. Table 5 shows the average results when fuel oil and coal-in-oil fuels were used. These tests were not specially run by test operators but were run as part of the regular plant steam production with the usual company plant operators. As shown in the tabulation, these operators obtained slightly lower boiler efficiencies with the coal-in-oil fuels than with the oil; about 79.5 to 80.5 per cent was obtained with the coal-in-oil fuels against 81.2 per cent with the oil; the difference is small enough to be within test errors. There was a small loss, item 69, of unburned combustible in the fly ash for the coal-in-oil fuels that was measurable.

No difficulties arose in burning the coal-in-oil fuels in the furnace with the steam-atomizing burner used. The flame responded smoothly to changes of boiler load and to variations of the quantity of air used for combustion. It could be controlled with the same ease as fuel oil. The CO_2 content of the products of combustion could be carried as chosen up to about 16 per cent with no CO. During operation no difficulties were experienced in the production of smoke; smoke could be controlled the same as with oil. However, in starting up a cold furnace with a coal-in-oil fuel, considerable smoke was produced. Fig. 13 shows conditions when starting. The boiler was operated at about rating during most of the testing, this load fitting best into the plant steam requirements from this boilerhouse. However, ratings from about 40 per cent to over 200 per cent were carried for several hours without difficulty.

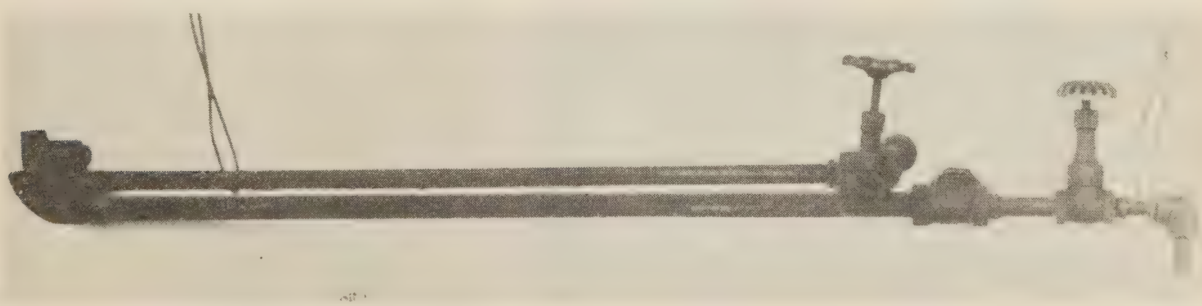

FIG. 11 STANDARD BEST STEAM-ATOMIZING BURNER USED

(Fuel reaches a horizontal slot at burner tip through lower pipe; steam reaches a vertical slot through upper pipe and sweeps fuel into furnace.)

TABLE 5 TESTS OF OIL AND COAL-IN-OIL FUELS, ATLANTIC REFINING COMPANY BOILER UNIT

		1	2	3	4
1	Test no.	1	2	3	4
2	Date of test, 1943	March 3–11	April 1–7	March 22–27	April 14 21
3	Maker and type of boiler	The Babcock & Wilcox Company; longitudinal drum			
4	Maker and type of superheater	The Babcock & Wilcox Company; convection type			
5	Maker and type of fuel-burning equipment	Best outside-mix steam-atomizing burner			
6	Boiler heating surface, sq ft	1510			
7	Superheater surface, sq ft	282			
8	Total heating surface, sq ft	1792			
9	Number of burners	One			
10	Method of producing draft	Natural-draft chimney			
11	Fuel: Type	No. 6 fuel oil	Coal-in-oil	Coal-in-oil	Coal-in-oil
	Batch no.		Batch No. 1	Batch No. 2	Batch No. 3
	Coal through 230 U. S. Std mesh screen, per cent		87.5	94.5	99.4
12	Area of furnace floor, 5′-0″ wide × 16′ 1 1/2″ deep, sq ft	80.6			
13	Height of furnace, mean, floor to nearest boiler heating surface, ft	8.0			
14	Total furnace volume (approximate), cu ft	550			
15	Furnace volume per sq ft boiler heating surface, cu ft	0.36			
	Fuel (Moisture-Free)				
16	Moisture (as fired), per cent	0.1–	0.9	1.5	0.9
17	Heating value per lb, Btu	18380	16820	16670	16960
18	Flash point (open cup), deg F				260
19	Carbon, per cent	87.4	84.5	83.8	85.5
20	Hydrogen, per cent	10.6	8.6	8.4	8.6
21	Oxygen, per cent	0.5	2.5	2.6	2.2
22	Nitrogen, per cent	0.2	0.7	0.7	0.7
23	Specific gravity at 60/60 F	0.982	1.082	1.094	1.083
24	Sulphur, per cent	1.3	1.3	1.4	1.2
25	Ash, per cent		2.4	3.1	1.8
	Products of Combustion				
26	Gas analysis, boiler outlet:				
	CO_2, per cent	12.8	12.2	13.9	14.7
	O_2, per cent	4.5	5.8	3.6	2.7
	CO, per cent	0.0	0.0	0.0	0.0
	N_2, per cent	82.7	82.0	82.5	82.6
27	Dry gas per lb dry fuel, boiler outlet, lb	17.2	17.3	15.1	14.7
28	Dry gas per lb dry fuel (no excess oxygen), lb	13.8	12.9	12.8	13.1
	Pressures and Drafts				
29	Steam pressure by gage, superheater outlet, psi	122	125	124	126
30	Pressure of fuel at burners, psi	0.3	1.5	3.0	2.0
31	Pressure of fuel at pump inlet, psi	+1.3 to +1.8	—2.2 to +0.1	—3.1 to +0.5	—3.7 to —0.4
32	Draft in furnace, in. of water	—0.01	—0.04	—0.04	—0.02
33	Draft at boiler outlet, in. of water	+0.04	+0.01	+0.02	+0.03
	Temperatures				
34	Steam temperature at superheater outlet, deg F	435	457	451	438
35	Superheat, deg F	84	104	99	85
36	Temperature of air for combustion, deg F	59	62	69	64
37	Temperature of gases leaving boiler, deg F	457	480	476	455
38	Temperature of feedwater entering boiler, deg F	222	192	215	212
39	Temperature of fuel in storage tank, deg F	75	81	88	84
40	Temperature of fuel at pump, deg F	60 to 70	85 to 125	75 to 110	70 to 95
41	Temperature of fuel leaving heater, deg F	180 to 200	160 to 180	180 to 195	150 to 170
42	Temperature of fuel at burner, deg F	163	154	163	142
	Hourly Quantities				
43	Duration of test, hr	192	144	120	120
44	Fuel per hour (as fired), lb	347	383	395	387
45	Fuel per hour (moisture-free), lb	347	380	389	384
46	Fuel per cu ft furnace volume per hour (as fired), lb	0.63	0.70	0.72	0.70
47	Fuel per burner per hour (as fired), lb	347	383	395	387
48	Actual water per hour, lb	4930	4670	4870	4950
49	Atomizing steam per hour, lb	90	150	130	160
50	Atomizing steam per lb of fuel (as fired), lb	0.26	0.39	0.33	0.41
51	Atomizing steam per cent of steam produced, per cent	1.8	3.2	2.7	3.2
	Unit Quantities				
52	Heat absorbed by water and steam, Btu per lb	1050	1092	1066	1061
	Refuse				
53	Dust in breeching, per cent of fuel (moisture-free), per cent		0.8	0.9	0.8
54	Refuse in furnace, per cent of fuel (moisture-free), per cent		0.4	0.8	0.3
55	Dust entering stack, per cent of fuel (moisture-free) (estimated), per cent		1.9	2.3	1.2
56	Percentage of combustible in breeching dust, per cent		32	36	28
57	Percentage of combustible in furnace refuse, per cent		0	0	0
58	Ash in furnace and breeching, per cent of ash in fuel, per cent		40	45	50
	Evaporation				
59	Rate of heat absorption per lb fuel (moisture-free), kB	14.93	13.42	13.34	13.67
60	Rate of heat absorption per sq ft of heating surface per hour, kB	2.89	2.85	2.90	2.93
61	Rating developed based on boiler heating surface only (10 sq ft per boiler horsepower), per cent	103	101	103	104
	Efficiency				
62	Efficiency, per cent	81.2	79.8	80.0	80.6

	Heat Balance, Based on Dry Fuel	Test No. 1 Btu	Test No. 1 Per cent	Test No. 2 Btu	Test No. 2 Per cent	Test No. 3 Btu	Test No. 3 Per cent	Test No. 4 Btu	Test No. 4 Per cent
63	Heating value of fuel, Btu per lb	18380	100.0	16820	100.0	16670	100.0	16960	100.0
64	Heat absorbed by boiler and superheater	14930	81.2	13420	79.8	13340	80.0	13670	80.6
65	Heat loss due to moisture in fuel	0	0.0	11	0.1	19	0.1	11	0.1
66	Heat loss due to water from combustion of hydrogen	1183	6.4	966	5.7	937	5.6	955	5.6
67	Heat loss due to dry chimney gases	1643	9.0	1736	10.3	1475	8.9	1379	8.1
68	Heat loss due to carbon monoxide	0	0.0	0	0.0	0	0.0	0	0.0
69	Heat loss due to combustible in dust leaving furnace (estimated)			105	0.6	131	0.8	67	0.4
70	Heat loss due to unconsumed hydrogen and hydrocarbons, radiation, and unaccounted for	624	3.4	582	3.5	768	4.6	878	5.2

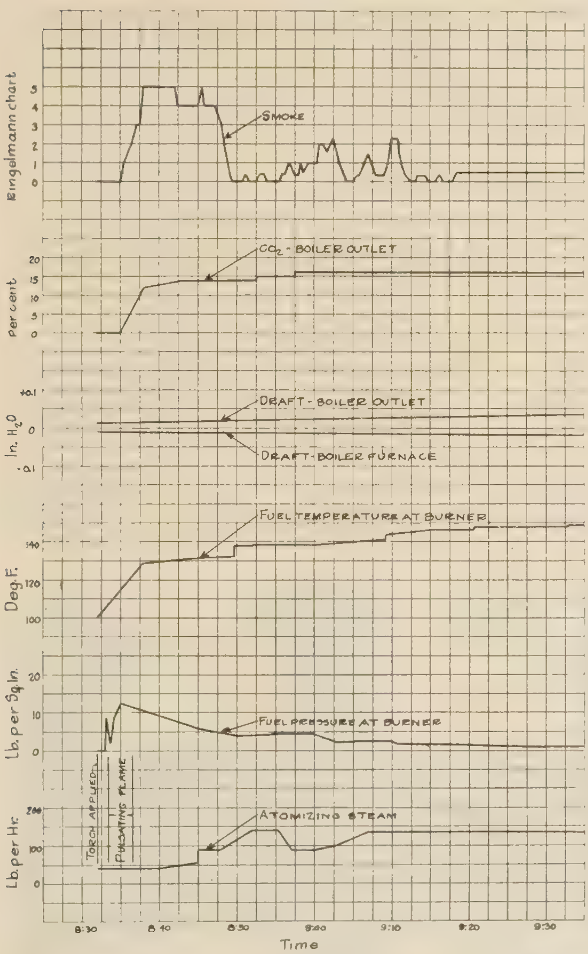

Fig. 13 Curves Showing Operating Conditions When Starting With Coal-in-Oil Fuel in Cold Boiler Furnace

As shown in Table 5, about two thirds more atomizing steam was required for the coal-in-oil fuels than for the oil. Both saturated steam and superheated steam were tried. Essentially the same amount of each was required to produce the same combustion results, as far as could be measured. It was found, however, that superheated steam gave a little less difficulty from pulsations and blowing of flame from the burner when higher amounts of steam per pound of fuel were used.

Troubles from the ash of the coal in the furnace were less than had been anticipated. The use of pulverized coal has supplied a vast amount of information on the action of ash in boiler furnaces and passes. In general, much less ash troubles should be expected with coal-in-oil fuels. It is not desirable to re-explore this field to any great extent in connection with an emergency fuel.

Fig. 14 is a view of the furnace with the ash accumulation after one week of operation. There was no dripping or slagging. The fly ash on the tubes was cleaned without difficulty every 8 hr during the test by the boiler soot blowers. Fig. 15 shows the effects of soot-blowing on the flue-gas temperatures when running at about rating with a CO_2 range from about 14.7 to 15.9 per cent. As shown in Table 5, about 45 per cent of the ash in the fuel accumulated in the furnace and breeching of the boiler.

It was found that with batch No. 1, for which coal had been ground to a fineness of about 88 per cent through a 230-mesh screen, there was considerable sludge accumulation at the bottom of the tank after about 7 days. The average temperature maintained in the tank was 81 F; no movement was given the fuel. With batch No. 2, for which coal had been ground to 95 per cent through a 230-mesh screen, only a small amount of sludge had accumulated at the end of 5 days. The average temperature in this case was 85 F. With batch No. 3, for which coal had been ground to 99 per cent through a 230-mesh screen, no discernible settling occurred after 9 days with the temperature in the tank at about 82 F. When the temperature was raised to 120 F, however, sludge accumulation was definitely discernible at the end of a few days. About 1500 gal of batch No. 4, for which coal had been ground to 99 per cent through a 230-mesh screen, was loaded in a typical delivery tank truck at about 115 F. The truck was then driven on the highway for about 9 hr covering about 150 miles; it was then allowed to stand for 32 hr. No settling occurred. The fuel was allowed to stay in the standing truck about 8 weeks. No settling was found at the end of this period. These field findings check with the laboratory findings.

No settling or clogging was experienced in the pipe lines from tank to pump to burner, although the lines were relatively long and the fuel was at elevated temperatures. For a given temperature, the coal in a coal-in-oil fuel separates much more readily when quiet than when the fuel is in motion.

There was appreciable accumulation of coal-oil paste in the heater when both batch No. 1 and batch No. 2 were used. It was unnecessary, however, to clean the heater during either test run. With the 88 per cent coal, there was an accumulation of 113 lb of paste after about 7 days; with the 95 per cent coal, 84 lb of paste after about 6 days. With the 99 per cent coal, the accumulation after about 9 days was practically negligible, being so

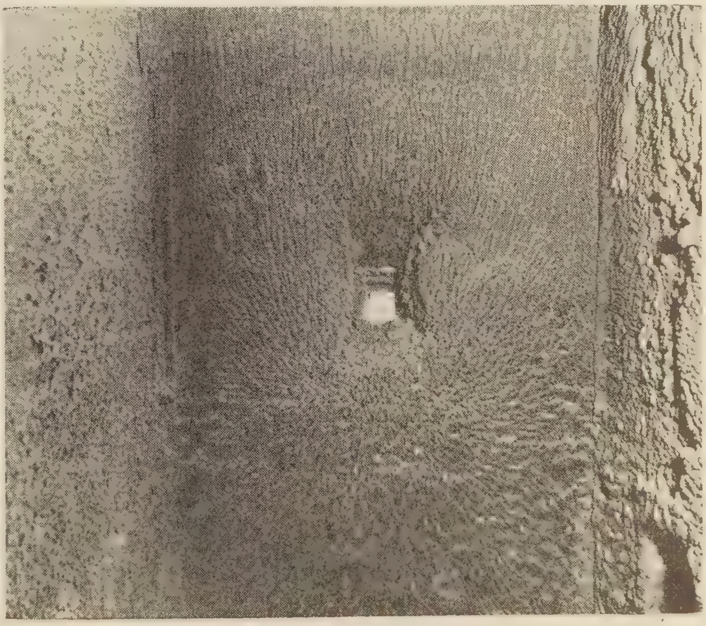

Fig. 14 Interior of Boiler Furnace Showing Ash Accumulation After One Week of Operation

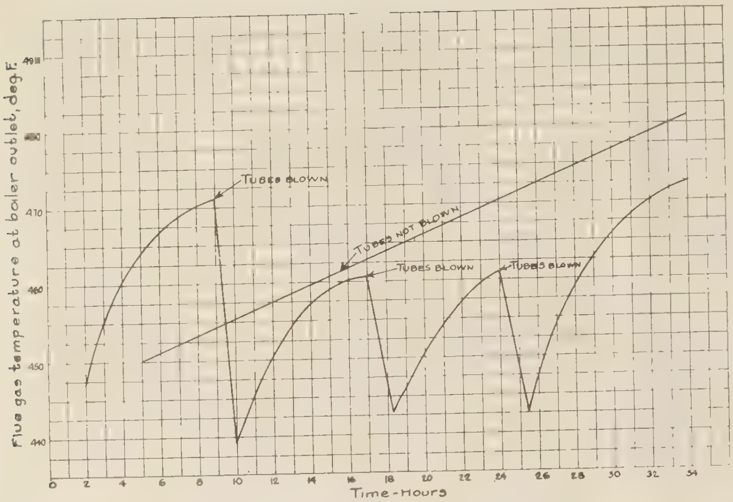

Fig. 15 Curves Showing Effect of Use of Boiler Soot Blowers on Exit Temperatures of Flue Gases

somewhat higher amount of steam, which could readily be done when superheated steam was used, appeared to help prevent this trouble.

Coal-in-oil fuels with their higher specific gravity and viscosity present new pumping problems when used in oil equipment. Studies of the flow of oil-coal suspensions have been reported by Gradishar, Faith, and Hedrick;[7] they supply formulas for determining pressure differentials. Table 1 gives viscosity comparisons of the coal-in-oil fuels, and the oil from which they were made.

The main pumping problem was from the storage tank to the pump; after the fuel was heated in the heater it was not so hard to pump. Coal-in-oil suspensions would have to be heated to temperatures of 140 to 170 F to give about the same viscosity as that of oil at about 100 F. These higher temperatures were avoided in the storage tank in order to eliminate settling of coal oil paste. At The Atlantic Refining Company boiler plant, the storage tank was above ground, and when oil was pumped at about 70 F at the pump the pressure at the oil-pump inlet was about 1.5 psi; when coal-in-oil fuels were pumped at about that temperature the inlet pressure was about 7 in. of vacuum or —3.4 psi. In starting up on coal-in-oil fuel, with no fuel in the lines about 1 hr pumping with the reciprocating pump was required to bring the fuel, at a temperature of about 80 F, from the tank to the burner; the rotary pump was not successful in starting the fuel but was quite satisfactory after the flow was established. When the fuel in the tank was 120 F, about $^1/_4$ hr of pumping was required to start.

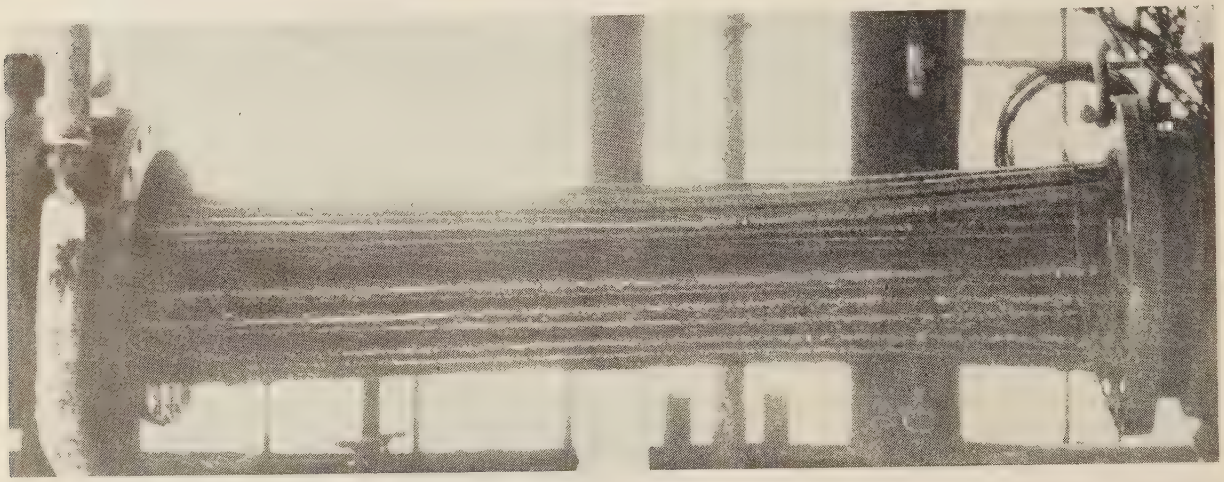

Fig. 16 View of Heater Coils After About 9 Days of Use in Heating to About 150 to 170 F Coal-in-Oil Fuel Having 99 Per Cent of Coal Ground to Minus 230 Mesh

(Accumulation was so slight as not to warrant cleaning for the next run.)

slight as not to warrant cleaning the heater. Fig. 16 shows the heater coils after the 99 per cent coal was used. In a steam heater of this type, some of the fuel is heated much hotter than the average temperature shown at the heater outlet. Fuel in contact with the steam coil approaches the temperature of the steam which was of the order of 350 F entering the coil. An offsetting factor, however, is the movement of the fuel which helps keep the coal in suspension. When coal-in-oil fuel is used, it is best not to heat it any hotter than necessary at any point in the system up to the furnace. Operating routine can be arranged to clean the heater when found necessary. If there are two heaters, they could be used alternately.

There was some clogging in the meters and screens with all of the coal-in-oil fuels. By-passes prevented loss of load during cleaning. Occasionally throughout the day the fuel control valve tended to clog, but it could be readily cleared by opening it wide quickly. Control-valve clogging would be apt to create some difficulties under automatic control. The burner tip occasionally clogged a little and had to be blown clear with a shot of steam piped to blow through the oil slot. Carrying continuously a

About 30 lb pressure was in general maintained ahead of the burner oil-control valve for both oil and coal-in-oil fuels. Table 5 shows the pressures at the burner; much higher relative pressures were required for the coal-in-oil fuels. The electric inputs to pump motors were increased. Under one set of operating conditions this input was about doubled. The increase in motor input depends upon the percentage of the total pump load which comes from moving the fuel itself.

For the rotary pump used the capacity dropped about 20 per cent. No satisfactory measurement was obtained in the loss of capacity of the reciprocating pump. The higher-viscosity fuel

DIMENSIONS OF COLLOIDAL FUEL PUMP AT NO. 14 BOILER HOUSE

	DATE INSPECTED	3-3-43	3-30-43	4-8-43	4-21-43	5-4-43
A	OD OF ROTOR	2.4995"	2.4985"	2.496"	2.495	2.495
B	WIDTH OF ROTOR	.9985"	.9965"	.992"	.990	.989
C	OD OF SHAFT	.625"	.622"	.620"	.620	.619
D	THICKNESS OF ROTOR WALL	.334"	.334"	.332"	.332	.330
E	THICKNESS OF ROTOR TEETH	.2765"	.276"	.275"	.275	.274
F	I.D. OF CASING	2.513"	2.513"	2.513"	2.514	2.515
G	OD. OF IDLER	1.929"	1.929"	1.929"	1.929	1.924
H	THICKNESS OF IDLER	.660"	.6585"	.655"	.655	.654
I	ID OF IDLER BUSHING	.627"	.634"	.640"	.647	.654
J	DIAMETER OF PIN	.625"	.618"	.616"	.610	.608
K	THICKNESS OF CRESCENT	.281"	.271"	.268"	.267	.266
	HRS.		75	139	103	73

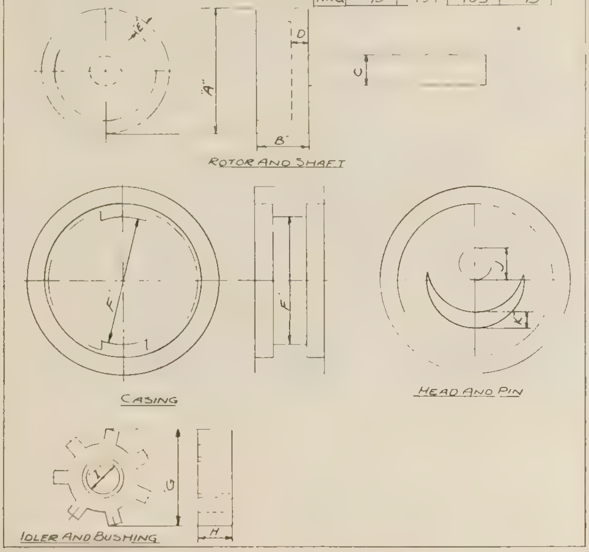

Fig. 17 Table and Sketch Showing Wear on Rotary Pump From Pumping Coal-in-Oil Fuel

causes the valves of reciprocating pumps to seat at a slower speed; stiffer valve springs helped to overcome this difficulty.

Coal-in-oil fuel was found to be rather abrasive. It acted something like a lapping powder. Fig. 17 gives a record of the wear on the rotary pump shown in Fig. 18. Appreciable wear occurred at the spindle; no repairing, however, was found

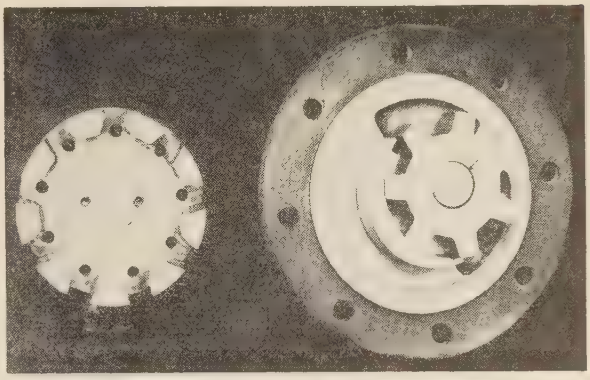

Fig. 18 Rotary Type of Oil Pump Used as Part of Heater-Pump Set
(Most of the wear was at spindle.)

necessary during the entire series of tests. There was also some cutting of the burner tip. Fig. 19 shows this erosion at both outer edges of the tiptable. It was unnecessary, however, to change the tip during the tests, good combustion being obtained. It is doubtful if this abrasive characteristic would necessarily be a serious handicap to successful operation. Equipment affected would have to be watched and repaired as found necessary.

Tests at Cambria Silk Hosiery Company Boiler Plant

A part of the batch No. 4 coal-in-oil fuel was burned in the boiler plant of the Cambria Silk Hosiery Company, Philadelphia, Pa. This plant was typical of the smaller industrial plant burning No. 6 fuel oil. The boiler used on the trial was a typical brickset 72-in. × 18-ft 150-hp H.R.T. boiler, equipped with one burner. The burner was a standard Johnson type 30-H automatic on-and-off rotary-cup burner having a maximum capacity of about 90 gph of oil. The oil pump, rated at 150 gph, an integral part of the burner assembly, drew oil from an underground storage tank through a 16-wire-mesh strainer and discharged it to a steam heater; from the heater the oil took two paths, one through a manually set relief valve back to the storage tank, and one to a 24-wire-mesh strainer to an electric heater, through a control orifice, and then to the burner cup. Primary air was supplied by a fan on the burner; additional air was dampered through a checkered furnace floor. Automatic ignition was effected by electric spark and gas pilot which were cut off after ignition.

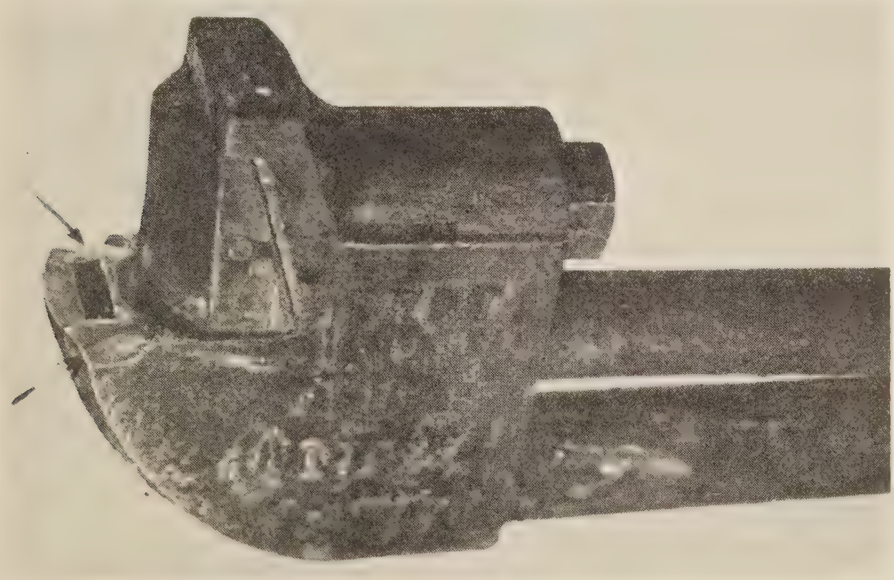

Fig. 19 View Showing Erosion at Outer Edges of the Tip-Table of Steam-Atomizing Burner From Use of Coal-in-Oil Fuel

In switching this plant from oil, the coal-in-oil fuel was not put into the underground storage tank. It was supplied to the pump through an auxiliary suction line of the same size as the main suction line direct from the delivery tank truck stationed about over the underground tank. This eliminated the suction lift. With the fuel at about 85 F in the truck, it was found that insufficient fuel could be pumped to maintain an even fire—perhaps

30 per cent of burner capacity—although 18 in. of vacuum was registered at the strainer just ahead of the pump. Under normal operation with No. 6 fuel oil at about 90 F from the underground tank, only 6 in. vacuum was required. A special heater was installed near the suction outlet in the tank and the relief hot-oil return flow was also piped closer to this outlet. This was done to heat the fuel at the suction outlet, where there was considerable movement; it was considered that the main body of fuel in the tank should be kept cool to avoid settling of coal-oil paste. When the fuel temperature at the suction was increased to about 95 F, the burner could be fired near capacity with a small amount of return oil to the tank. At 120 F very satisfactory pumping conditions resulted, a vacuum of 7 in. being carried.

On the delivery side of the pump it was found necessary to change the size of the orifice from 0.0935 in. diam as regularly used, to a larger orifice 0.161 in. diam to lessen restriction of flow. Under normal operating conditions with oil and with the smaller orifice, an oil pressure of about 15 to 18 lb was carried; with coal-in-oil fuel and with the larger orifice, a pressure of about 20 to 23 lb was required for the same load. The 24-mesh screen ahead of the electric heater became clogged after about 3 hr; a larger-mesh screen would be desirable for coal-in-oil fuels. The final temperature of the fuel to the burner ranged in general from about 140 to 160 F.

When starting with a comparatively cold furnace, it was found that the automatic ignition would not ignite coal-in-oil fuel; a hand torch had to be used. A larger gas pilot was needed. With a hot furnace, however, the automatic ignition operated quite satisfactorily.

The revolving cup of the burner for use with oil fuel was tapered outward somewhat. The flame of the heavier coal-in-oil fuel flared out from the cup more than the flame of the oil. This caused carbon masses to accumulate on the circular brickwork throat of the burner, which had to be cleaned about every 15 min. The tapered cup of the burner was changed to a straight-line cup; the brickwork of the throat was flared out a little farther. These changes resulted in satisfactory operation. The burner was readily adjusted and the CO_2 could be carried as chosen. Somewhat improved operation could be had by heating the fuel to about 160 to 170 F ahead of the burner. The efficiency when coal-in-oil fuel was used was about the same as when oil was used. The maximum load on the boiler with coal-in-oil fuel on this installation was approximately 10 per cent less than with oil. This was caused primarily by the difficulty of supplying enough secondary air in the right spots. The fly-ash problem was of little moment.

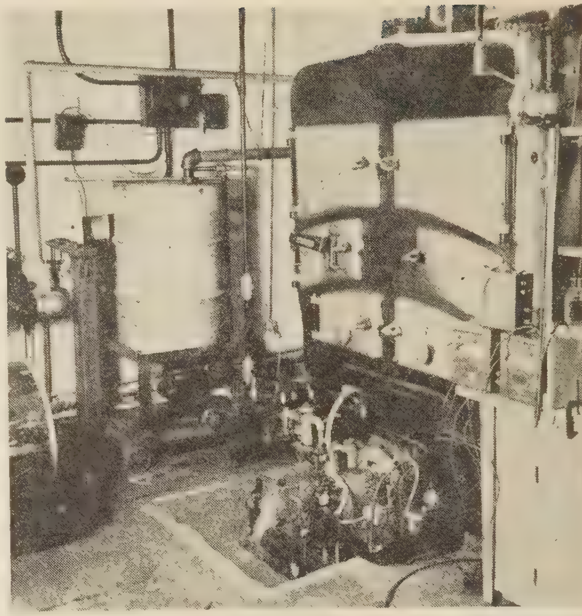

Fig. 21 View of Boiler Front and Burner at Heating Plant of National Airoil Burner Company

Tests at National Airoil Burner Company Heating Plant

In co-operation with the National Airoil Burner Company of Philadelphia, Pa., part of batch No. 4 of coal-in-oil fuel was tried in its office headquarters' heating boiler. This boiler was a cast-iron sectional boiler having an output of about 3,060,000 Btu (approximately 90 hp). It was equipped with a small rotary-cup burner of a capacity of 20 gph of oil. Figs. 20 and 21 show the general layout. This plant was typical of the small heavy-oil-burning heating plant.

Normally the plant uses No. 6 oil pumped from an underground storage tank. Since it was believed that the regular burner setup would not be able to pump coal-in-oil fuel from the underground tank, a 275-gal tank was installed above ground as shown in Fig. 21. The suction and return lines, A and B, were connected to the existing suction and return piping properly valved so that fuel from either tank could be used.

The oil burner was started with oil, and normal operation was established. The equipment was switched to coal-in-oil fuel having a temperature of about 68 F. The pump suction registered 20 in. of vacuum and was held at this point for 1 hr, but no fuel reached the burner. The equipment was switched back to oil, and the vacuum dropped to 15 in., giving a good oil flow. It was then decided to install a 52-gal tank above and to the side of the burner, as shown in Fig. 21. An electric fuel heater was placed directly under the tank. The usual suction and return lines were installed.

The burner was started on oil from the underground tank at 65 F, and a burner pressure of about 26 lb was used. The pump suction was about 13 in. of

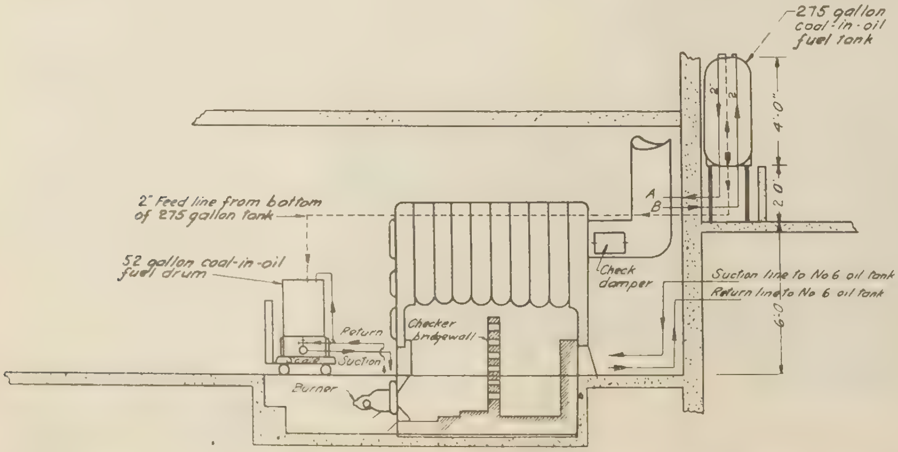

Fig. 20 General Layout of Test Setup at Heating Plant of National Airoil Burner Company

vacuum; the CO_2 ranged from about 11.5 to 13.5 per cent; the stack temperature was about 570 F. The equipment was switched to coal-in-oil fuel ranging in temperature at the tank from about 85 to 110 F. The pump pressure was about 26 lb; the pump suction was about 0; the CO_2 ranged from about 10.5 to 13.5 per cent; the stack temperature was about 625 F. The combustion was not satisfactory, as the flame was smoky at the outer edges. Carbon deposits accumulated on the side walls.

The equipment was shut down and allowed to cool to determine possibilities of cold starting with this type of equipment. After the fuel cooled in the pipe lines, it was very difficult to pump. The burner pressure was raised to about 40 lb to move it. The gas pilot would not ignite it. Ignition was finally obtained with a hand torch, and after furnace and fuel had reached higher temperatures, the flame was fairly well established. The carbon deposits continued to build up on the brickwork around the burner and in front of the cup until the opening was choked and it was necessary to shut down after about 20 min. The wider angle of the flame of the coal-in-oil fuel was the main cause of this difficulty.

Conclusions

Coal-in-oil fuel has its own peculiar characteristics which must be given thorough consideration for each job if satisfaction is to be obtained. However, this fuel does not present a particularly difficult problem in devising equipment and operating routine for efficient use. It appears that in some instances premixing of the fuel might be eliminated. The direct introduction of the coal into the flowing stream of oil to the heater-pump set offers possibilities. The information obtained from this work on bituminous coal-in-oil fuels leads to the following conclusions:

1 A sufficiently stable suspension of coal in oil for practical purposes can be prepared by dispersing 40 per cent coal by weight, ground from 98 to 99 per cent through a 230-mesh screen, in heavy fuel oil. The coal must be ground to this fineness to avoid most of the difficulty arising from unstable suspensions.

2 Mixing the coal into the oil does not present a particularly difficult problem.

3 Careful control of storage and handling temperatures is required for most successful operation.

4 Careful attention must be given the pumping of coal-in-oil-fuels, particularly in cases where underground storage tanks are used.

5 The combustion efficiency of coal-in-oil suspensions is apt to be slightly less than that of oil.

6 The load-carrying flexibilities of oil and coal-in-oil fuels are about the same.

7 The best opportunity for the successful use of coal-in-oil suspension is in the larger plants with the better operating personnel and more readily accessible equipment.

Acknowledgments

Acknowledgment should be given to the following persons for their part in carrying out this work: of The Atlantic Refining Company, C. J. Cutting, W. A. Myers, S. W. Ferris, W. A. Schmidheiser (for his part in getting the first large-scale test under way), J. C. Reid, and K. M. Thompson (the last two for performing most of the laboratory work); of the Bureau of Mines, Dr. W. C. Schroeder, R. Wiggers, and J. G. Mingle (the last two for their general field-testing work).

Appendix

Description of Pendulum Apparatus and Procedure for Its Use in Studying Stability of Coal-in-Oil Suspensions

Since the opacity of the oil used in coal-in-oil suspensions precludes visual observation of settling phenomena, the evaluation of stability presents a problem. A modification of a method, described by Manning and Taylor[6] and again by McMillen, Stutzman, and Hedrick,[8] has proved satisfactory for this study. The apparatus used, Fig. 22, consists of a carriage which is supported on knife-edges and is so designed that a tube containing a suspension of coal in oil may be fitted into it and held rigidly in place. The periods of this system (acting as a compound pendulum, observed first with the tube in its normal position in the carriage, and then with the tube raised through a known distance referred to its support by the insertion of a block of known dimensions), provide a means of calculating the location of the center of gravity of any tube and its contained suspension.

Period T of a compound pendulum may be expressed

$$T^2 = \frac{4\pi^2 I}{Mg\bar{r}}$$

where I is the moment of inertia of the pendulum about the axis of support, M is the mass of the pendulum, g is the acceleration due to gravity and $\bar{r}$ is the radius of the center of gravity. Combining this equation with the theorem of mechanics relating the moment of inertia I about any axis to the moment of inertia I_0 about an axis through the center of gravity

$$I = I_0 + M\bar{r}^2$$

and separating the various parts of the system, the location of the center of gravity of the tube and its contained suspension is given by

$$\bar{r}_t = \frac{\bar{r}_c M_c g T_1^2 + [gdM_t - (\bar{x}_c M_c g + \bar{r}_b M_b g)]\, T_2^2 + 4\pi^2 I_b + 4\pi^2 M_t d^2}{gM_t\,(T_2^2 - T_1^2) + 8\pi^2 M_t d}$$

[8] "A Pendulum Method for Measuring Settling Velocities," by J. H. McMillen, L. F. Stutzman, and J. E. Hedrick, *Industrial and Engineering Chemistry*, Analytical Edition, vol. 13, 1941, pp. 475–478.

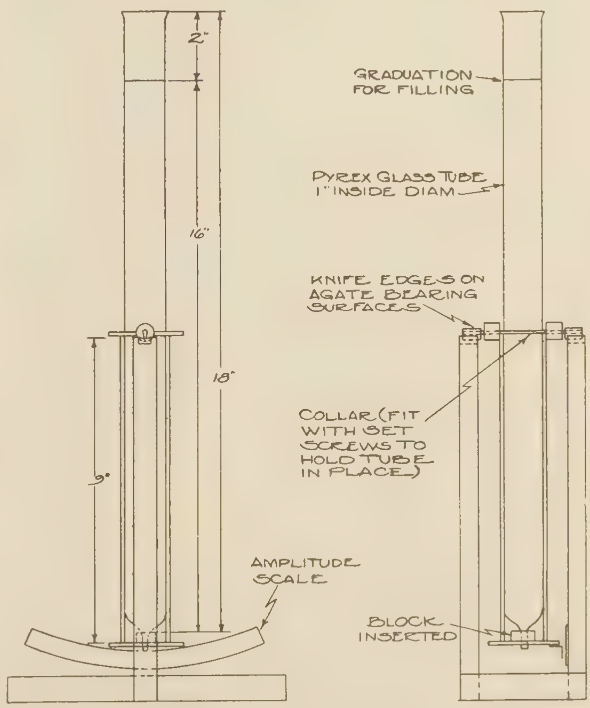

Fig. 22 Sketch of Pendulum Apparatus Used in Determining Settling in Coal-in-Oil Suspensions

where

$\bar{r}_t$ = radius of center of gravity of tube and contained suspension in its initial position, cm
$\bar{r}_c$ = radius of center of gravity of carriage, cm
M_c = mass of carriage, g
g = acceleration due to gravity, cm per sec per sec
d = distance through which tube is raised, cm
M_t = mass of tube and contained suspension, g
$\bar{r}_b$ = radius of center of gravity of inserted block, cm
M_b = mass of inserted block, g
I_b = moment of inertia of inserted block about axis of support, g $\times$ cm^2
T_1 = period of compound pendulum without block, sec
T_2 = period of compound pendulum with block inserted, sec

Experimental results for the pendulum used in these experiments indicate an average deviation of ± 0.025 cm in $\bar{r}_t$ calculated by this equation. For the best accuracy the carriage should be as light as possible and the axis of support should be only a short distance above the center of gravity of the pendulum. Temperature fluctuations affect the accuracy of this work and must be compensated for unless the tests are run at constant temperature. Suitable compensation may be effected by plotting the period against the age of the sample, thereby deriving an average curve from which corrected values may be taken.

The change in the position of the center of gravity may be correlated with the percentage of settling that has occurred. For example, complete settling, determined for a 40 per cent suspension of coal in an ethanol-water solution of the same specific gravity as the oil, results in a net change of 0.48 cm (± 0.02 cm) in the position of the center of gravity of a tube and its contained suspension. Assuming that the net effect of the sedimentation is a gradual concentration of the slurry, a linear relationship between the change in the center of gravity and percentage of settling may be used. Thus with the sensitivity of the pendulum method previously noted, a single determination of the percentage settled is accurate to within about 5 per cent. Through the use of the compensation method described, large errors may be detected readily and the accuracy of the determinations considerably improved.

As a check on the pendulum test, samples that had shown no change in the position of the center of gravity on long standing were analyzed top and bottom by quantitative methods. The results obtained showed the same concentration of coal at top and bottom. The stability was checked further by visual examination of beaker mixes; a removable disk was placed at the bottom of the beaker and the material adhering to the disk was examined periodically. Tests in such studies confirmed the findings of the pendulum tests. Visual results were proved by quantitative methods.

Extreme care is required in the preparation of mixes for the evaluation of stability by pendulum test. Air carried into the mixture by the coal is lost very slowly, and by resulting in reduced volume, changes the period of the pendulum in a manner that may easily be misinterpreted as settling. This difficulty may be overcome by heating the mixture to 160 F with hand mixing and then stirring the warm mixture for 15 min at an absolute pressure of 10 mm Hg.

The coal used throughout the laboratory tests was ground in a hammer mill as received. Various sizes were separated carefully by screening. No effort was made to dry the pulverized coal beyond its equilibrium value with the atmosphere.

TABLE 6 OIL USED IN BATCH NO. 3[a]

Distillation at atmospheric pressure, 742 mm. First drop, 185 C (365 F):

Fraction number	Cut at deg C	Cut at deg F	Per cent	Sum, per cent	Specific gravity, 60/60 F	Deg A.P.I., 60 F	C.I.	S.U. viscosity at 100 F., sec	Cloud test, deg F	Aniline point, deg C
1	50	122								
2	75	167								
3	100	212								
4	125	257								
5	150	302								
6	175	347								
7	200	392								
8	225	437								
9	250	482	4.0	4.0	0.868	31.5	..			28.1
10	275	527	5.8	9.8	0.909	24.2	66			29.0
Distillation continued at 40 mm:										
11	200	392	1.7	11.5	0.921	22.1	67	41	5	..
12	225	437	5.8	17.3	0.928	21.0	67	47	25	46.5
13	250	482	8.6	25.9	0.935	19.8	67	64	50	53.2
14	275	527	9.3	35.2	0.942	18.7	67	110	70	59.9
15	300	572	9.5	44.7	0.946	18.1	66	230	85	65.7
Residuum			55.2	99.9	1.016	7.8				

Carbon residue of residuum, 19.3 per cent; carbon residue of crude, 10.9 per cent.

[a] Distillation, Bureau of Mines, Hempel method.

Discussion

E. G. Bailey.[9] This paper gives some interesting data covering important factors in relation to the burning of mixtures of pulverized coal and oil. The physical factors are one thing, the economic factors quite another.

It is the writer's opinion that with respect to the so-called emergencies we have already had, and those which we may yet expect, it is still better to continue the use of oil in those plants, which are not readily convertible 100 per cent to coal, and make those conversions which should be made to balance the fuel supply, by converting the installations which can readily be operated 100 per cent with coal.

It would seem to take substantially as much pulverizing and special classifying equipment to prepare 40 per cent coal for the mixture, as it would for 100 per cent coal with proper fineness, to be burned in those places where it can be done. Therefore, more oil could be saved with the same equipment or the same saving with less equipment if conversion to 100 per cent coal were the chosen method.

A higher degree of fineness in burning pulverized coal enables its use in many places where it is considered unadaptable with the coarser coals which many people have tried with unsatisfactory results. Many so-called expert opinions on this subject are hand-me-downs from experiences with coarser coal and poor burners of by-gone days, and such opinions and decisions should not prevail over the later experience with finer pulverization and modern equipment.

W. C. Schroeder.[10] The tests reported in this paper give a complete picture of the preparation and burning of colloidal fuel in a relatively small boiler which is typical of a number of oil-burning installations. The authors' work parallels, in several respects, an investigation that has been carried out by W. I. Jones of Powell Duffryn Associated Collieries in Wales.[11] This discussion will show that the results from the two studies are in substantial agreement and will also present additional information available from the work abroad.

Stability of Oil-Coal Suspensions. From the tests which were made in

[9] Vice-President, The Babcock & Wilcox Company, New York, N. Y. Fellow A.S.M.E.

[10] Assistant Chief, Fuels and Explosives Service, Bureau of Mines, Washington, D. C. Mem. A.S.M.E. This discussion is published by permission of the Director, Bureau of Mines, U. S. Department of the Interior.

[11] The information for this discussion was provided by W. Idris Jones, of Powell Duffryn Associated Collieries. The writer is indebted to him for this material and for permission to publish it.

Wales the following statements were made concerning stability:

"This is a relative term and no coal-in-oil suspension is perfectly stable. The stability decreases with a decrease in coal content or fineness and with increasing temperature. Stability also depends on the nature of the oil. Cracked residual fuels of the Bunker C type have adequate inherent stabilising characteristics to yield a stable suspension with 99% through 200-I.M.M. (230-U. S. Standard) mesh, coal in concentrations preferably of 40–45%. Regular laboratory tests and careful inspection of tanks and pipe lines have shown that these suspensions are adequately stable for all normal commercial requirements, provided there is no intense local overheating. In this connection, our storage tank at Aberaman was fitted with hot-water coils instead of steam coils. Coal-in-oil suspensions should not be kept in a static condition in the presence of steam coils more than is strictly necessary to ensure the mobility of the fuel. Turning full steam pressure on the coils is to be avoided unless the fuel is being circulated.

"Reduced stability at elevated temperatures is not accompanied by the precipitation of coal as such. What actually occurs is a gradation in coal concentration increasing from the top layer to a maximum of about 58% in the bottom 20% layer.

"In the case of 40% suspensions of 200-I.M.M. dry steam coal in Bunker C oil numerous stability tests since 1933 have enabled us to draw up the following table (Table 7) which sets out the various times taken to attain a concentration of 45% and 58% of coal in the bottom 20% layer of the "Hurrelled"[12] fuels at various temperatures:

TABLE 7

Temperature, C	Deg F	Time to reach a coal concentration in the bottom layer of— 45%	58%
15	59	More than 150 days	More than 500 days
30	86	Approximately 65–70 days	Approximately 240 days
50	122	Approximately 28 days	Approximately 85–90 days
70	158	Approximately 5 days	Approximately 17 days

"The figures in column 3 show the approximate maximum time for which it is safe to allow a 40% suspension (Hurrell) in Bunker C oil to stand at various temperatures to permit of its direct use from a heated tank. The figures in column 4 indicate the approximate times in which the mixture will break down completely at various temperatures, leaving a top layer of fuel oil.

"Summing up, it may be claimed that if due precautions are taken against overheating, the fuel suspensions containing 40–45% of coal in Bunker C oil are stable enough for all practical commercial requirements and can be handled by the normal oil-handling equipment. The coal, however, must be ground and classified to yield not less than 98.5% through a 200-I.M.M. mesh screen and then thoroughly wetted with the oil, first in a primary mixer and then in a final mixer of the Hurrell Homogeniser type, normally used for making emulsions."

The effect found for temperature, time, coal fineness, and coal concentration on stability are in good agreement with those reported by the authors of this paper. Jones, however, went to somewhat greater length to insure thorough mixing of the finely pulverized coal and oil.

TABLE 8

Coal in mixture, per cent	Viscosity, Saybolt Furol sec. at 100 F	150 F	200 F	250 F
55	..	14000	6000	3000
50	1800	570	250	140
45	660	190	79	37
40	380	110	46	24
35	290	88	42	19
30	190	46	17	...

[12] Run through Hurrell homogeniser (G. C. Hurrell, Limited, Old School Works, Woolwich Road, Charlton, England). In all of this work the coal was ground to the desired fineness, mixed with the oil, and then passed through the homogeniser.

Effect of Coal Concentration and Temperature on Viscosity. The data in Table 8 give the results Jones secured for the effect of coal concentration and temperature on viscosity. These data carry the viscosity values to higher temperatures than reported in the paper.

The figures were secured from logarithmic graphs compounded from a very large number of mixtures examined during the course of the work. In general the coal was about 98–99 per cent through 200-I.M.M. (230-U. S. Standard) sieve, while the Bunker C oils showed the following characteristics:

Specific gravity at 60 C	0.98–0.99
Flash point, deg C	156–208
Viscosity, Saybolt Furol, sec	
122 F	99–499
175 F	27–33
225 F	14–17
Pour point (A.S.T.M.), deg F	10 15

Combustion Tests. Jones carried out over 150 combustion tests with colloidal fuel on a boiler having the following characteristics:

Type	3 drum
Heating surface, sq ft	3947
Working pressure, psi	175
Furnace volume, cu ft	795
Evaporation rate, lb per hr	18000–20000
Preheated air, deg F (mean)	335
Air pressure at burners, in	2–3
Fuel pressure at burners, psi, avg	120
Fuel temperatures at burners, deg F	260–280

He reported as a result of this work that the boiler furnace, when using coal-in-oil suspensions, even when viewed by a skilled observer, was not different in appearance from the oil-fired furnace. The efficiency depended upon the fineness of the coal, the volatile content of the coal, the nature of the oil, and the type of burner used.

Tests were run with coals containing 12 to 40 per cent volatile matter. The mixtures containing high-volatile coal gave a somewhat higher efficiency than those containing low-volatile coal. Under the worst conditions, efficiencies dropped about 5 per cent below those for oil alone. Under the best they were within 1 per cent of oil.

Jones states, "One of the most pleasing features of the boiler tests was the almost complete freedom from burner stoppages. Ignition was hardly ever lost in the tests due to the burners going out. In fact, the whole equipment, comprising pumps, filters, heaters, and atomizers, gave a performance which was quite indistinguishable from that obtained when using fuel oil in the same plant." This seems in essential agreement with the work reported in the paper.

Jones ran his tests with a variety of pressure, pressure and air, or pressure and steam burners and he states, "The Babcock and Wilcox pressure oil burners were adopted for convenience for most of the tests, but other burners of this type would be equally suitable."

Manufacturing. During the period in which this work was carried out in Wales, a small plant was erected to make about 0.75 ton per hr of oil-coal suspension. From this pilot plant, estimates were made of capital and manufacturing costs. On this basis, the writer has estimated that the cost of preparing 60 per cent oil - 40 per cent coal suspensions in the United States today would be about $0.80 to $1 per ton for a plant of about 100 tons per day.

This does not, of course, include the cost of the coal or oil but it does include all preparation costs, i.e., services, power, manage-

ment and supervision, labor, maintenance, rent, interest on investment, taxes, depreciation at 10 per cent, and insurance. It must be recognized, however, that there is considerable chance for error in translating costs in one country to another, which could not be avoided with the data available.

Summary. From the work done in this country and abroad, sufficient technical information is now available to contemplate the erection of colloidal-fuel plants of considerable size with reasonable assurance that the product could be used in certain oil-burning installations with little modification of existing equipment. This procedure provides a means for saving oil without the large equipment alterations necessary to convert plants and furnaces completely from oil to coal. It might therefore be of value during the war period with the prevailing critical equipment shortages. In general, the Bureau of Mines has not looked on the oil-coal mixtures as having a permanent place as an industrial fuel. Under normal conditions, with the necessary equipment available for handling any fuel, a plant will use oil or solid fuel alone, according to the dictates of cost, convenience, or other demands of a particular installation. From the long-range point of view, it is difficult to visualize a large field for colloidal fuel.

During the period in which this work was carried out, even the short-range viewpoint has altered greatly. The shortage of coal in both the United States and Great Britain becomes steadily more acute, and the saving of oil at the expense of coal looks much less attractive than it did a year ago. Unless this situation changes, it is doubtful if there will be any extensive use of colloidal fuel during this war.

C. C. Wright.[13] The authors are to be congratulated for their presentation of a timely and much needed clarification of the technical factors concerning the production and use of coal-in oil fuels. From the viewpoint of the fuel technologist, a small amount of additional data on specific gravities would be most helpful in settling questions regarding the volume of fuel produced upon admixture of coal and oil. Specifically, the writer would like to see data included on the specific gravity of the coal-in-oil fuels reported in Table 2 of the paper, the specific gravity of the coal used in the preparation of the fuels reported in Table 4; also whether or not any change in specific gravity of the coal-in-oil fuel was observed with aging.

[13] Professor of Fuel Technology, The Pennsylvania State College, State College, Pa.

Authors' Closure

The authors agree thoroughly with the statement in Mr. Bailey's discussion that "the physical factors are one thing, the economic factors quite another." In this paper, attempt was made to supply new and needed information of a physical nature; such facts are required before economic factors can be soundly considered. Economic studies have been made both by The Atlantic Refining Company and the Bureau of Mines. Publication of these studies has not been made but they are available.

Dr. Schroeder supplies interesting information from unpublished work recently done in England which adds to the value of the paper.

In answer to Mr. Wright's inquiries, the following is submitted:

The specific gravities of the coal-in-oil fuels of Table 2 were not determined. The specific gravity of the coal of batches 1 and 2 of Table 4 was 1.30 and of batch 3 and 4, 1.32.

The following table gives typical specific gravities and viscosities of a coal-in-oil fuel over a period of eight days; other stable suspensions examined for gravity after much longer periods of storage showed no significant changes in gravity as the result of storage:

TABULATION OF SPECIFIC GRAVITIES

Batch No. 3

	Initial tank inspection	Daily analysis of fuel — April, 1943							
		14	15	16	17	18	19	20	21
Per cent coal in mixture		40.0	40.2	40.1	39.6	40.6	39.0	39.0	39.0
Sp. Gr. @ 80 F	1.085	1.089	1.079	1.090	1.085	Sample lost	1.077	1.078	1.087
Vis. @ 122 F (Furol)	331	412	380	397	368		361	324	328
Vis. @ 210 F (Furol)	40	52	48	49	47		46	42	44

Wear-Resisting Materials for Lathe Construction

By R. W. DAYTON,[1] C. H. LORIG,[1] and R. E. ADAMS,[1] COLUMBUS, OHIO

It was found, from tests of all the materials now in general use as the bearing surfaces of lathes, that the combination of hardened steel and alloy cast iron was far more wear-resistant than others and was adequately scoring-resistant.

A process referred to as "flame refining" was developed for improving the wear resistance of cast iron. An iron treated by this process and used with hardened steel formed the best combination tested, having greatly improved wear resistance and scoring resistance.

A wear test in simulated lathe service confirmed the earlier work and also showed that for minimum wear of a combination containing cast iron and hardened steel, the steel should form the carriage surface and the cast iron the bed surface.

GENERAL-purpose machine tools change but little in design from year to year and would remain useful almost indefinitely if they retained their initial accuracy. However, they do not, largely because of wear on the friction surfaces of the bed and the carriage which causes a gradual but eventually fatal loss of accuracy.

Machine tools vary in ability to resist deterioration of their ways. The ways of grinders, for example, remain true for many years because the conditions of service and the possibilities of design are such that the causes of wear are avoided. Thus materials of great wear resistance are not essential, so the original and most readily used material for machine-tool construction, plain cast iron, is still in use for all rubbing surfaces.

Other types of machine tools such as lathes operate under adverse, almost abusive, conditions. Since design can do little to make these conditions tolerable, wear-resistant materials must be used if the lathe is to have a long useful life. Accordingly, plain cast iron is being used less, and hardened steel and cast iron more, for the beds and carriages of lathes.

The number of materials which can be used for lathe-bed and carriage construction is limited. The body of both bed and carriage must be made of cast iron for economic reasons, so if different types of bearing surfaces are desired, they must be applied on the surfaces. Two methods of altering the bearing surfaces are now in use, flame-hardening the cast iron itself or fastening hardened-steel strips to the surface.

One other limitation in the choice of materials must be observed, namely, that either the carriage or the bed must be soft enough to be scraped to a fit by hand, for machine tools cannot economically machine the bed and carriage surfaces to a satisfactory fit. This prevents the use of fully hardened surfaces on both bed and carriage, a combination which might be the most wear-resistant of all.

As a result of the limitations of choice, only four combinations of materials for the wearing surfaces of lathe beds and carriages are in general use. These are the following:

1. Cast-iron bed, cast-iron carriage
2. Flame-hardened cast-iron bed, cast-iron carriage
3. Hardened-steel-faced bed, cast-iron carriage
4. Cast-iron bed, hardened-steel-faced carriage.

The manufacturing advantages and disadvantages of each of these types of construction are thoroughly understood, but the serviceability of each is not. The purpose of the study being reported was to appraise the serviceability of each of the combinations and then, if possible, to develop one better than any. The problem proved so complex that nearly four years of study were required to achieve the desired result.

Testing Methods

Lathe beds and carriages may experience two distinctly different types of damage. Either they may wear, losing material gradually from their rubbing surfaces until the accuracy of the lathe is lost, or they may gall, material being torn out of one surface in small pieces and welding to the other causing deep gouging and scratching. It can be expected that proper materials for the ways of lathes will not gall and will wear only slowly.

Wear of lathes is generally due to either abrasive or lack of lubrication. With adequate lubrication and abrasive-free conditions, no wear occurs as shown both by laboratory tests and by long-time running of a full-sized lathe under these conditions. If lubrication is lacking because of carelessness, very rapid wear will occur; so unlubricated wear was studied. However, since only a minute amount of oil suffices to prevent unlubricated wear, and since the necessary amount is likely to be present on the bed in all but rare instances, the results for unlubricated wear are considered less significant than those for abrasive wear.

Practically all lathe-bed wear is abrasive in character. If abrasive could be prevented from working between the rubbing surfaces, lathe beds would be as long-lived as grinder beds. Unfortunately, design considerations do not permit adequate protection of beds as long as those of some lathes.

Two types of abrasive cause most lathe-bed wear; scale from heat-treated steel and dust from cast iron. Both of these are brittle enough to crush small particles that get trapped between the rubbing surfaces, and hard enough to cause wear. Tests for abrasive wear were made in both of these abrasives. Some description of the wear tests is given in the caption of Fig. 1.

The galling, or scoring, which infrequently occurs between bed and carriage, is an acute form of failure. It is induced by "short-stroke" work in which the carriage moves back and forth many times over a short distance on the bed. The action appears somewhat similar to the wringing together of gage blocks, the continued slight reciprocation bringing the surfaces closer and closer together until molecular attraction causes welding. The only certain method of resisting this form of failure is to lift the surfaces out of contact at intervals, as by an oiling system which forces oil between the surfaces periodically.

No metals seem capable of resisting this type of failure indefinitely, even at minute loads, when the stroke is short. However, metals vary in their ability to resist scoring for a longer

[1] Battelle Memorial Institute, Columbus, Ohio.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

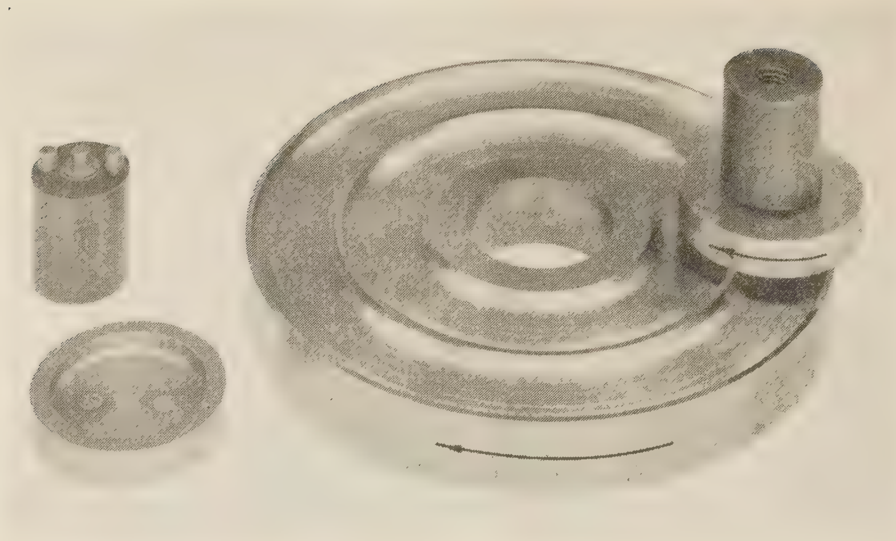

Fig. 1 The Wear-Test Specimens Used in the Early Investigation

(The large plate is made of one of the materials under test; the two small specimens like the one inverted in the lower left hand of the photograph are made of the other. The holder behind the small specimen centers and drives it, allowing rocking for alignment. During a test, both small specimens rest on the plate, one in the position shown, the other diametrically opposite. All three specimens are driven at the same angular speed in the direction indicated by the arrows. Unlubricated tests were made in both dry and humid air, for humidity was found to affect wear. The conditions under which each type of test listed in Table 1 is run are as follows:

Scale. The specimens are submerged in an agitated mixture of 25 per cent of —200 mesh heat-treating scale and oil. They are run at 148 ft per min and 210 psi of pressure. Three successive two-hour tests are run and an average taken.

Modified Scale. This test is used to distinguish between slow-wearing combinations. The conditions are the same as for the test in scale, except that the three two-hour tests are preceded by a 16-hour wearing-in period, during which the specimens are roughened and charged with abrasive so that rapid wear occurs.

Cast Iron. The same as the test in scale, except that the specimens are submerged in a mixture of 25 per cent of —200-mesh cast-iron dust and oil.

Dry Air. The specimens are surrounded with thoroughly dried air and run unlubricated, at a speed of 16 ft per min and a load of 75 psi. Four two-hour tests are run.

Moist Air. The same conditions as the test in dry air, except that the air is brought to about 100 per cent humidity and the load is 510 psi.)

stroke, and it is desirable that the best be selected for lathe-bed service. For sorting out the various materials, a scoring test was used which oscillated two specimens together, as discussed in the caption of Fig. 2.

Preliminary Test Results

The results of the preliminary wear and scoring tests are summarized in Table 1. Each of the values reported is an average of three or more tests of each sample, generally of at least three samples and occasionally of seven or eight. In the cases where only a single material of a type was tested, the hardness is listed as a single number instead of a range.

A "relative-wear index" appears in next to the last column of Table 1. It was introduced to facilitate rough comparisons between the various combinations. The index was obtained by taking a weighted average of all the tests which were first reduced to a ratio, using combination IV-*a* as standard. A relative weight of 4 was assigned to the abrasive tests and of 1 to the unlubricated tests. This weighting takes cognizance of the fact that most lathe-bed wear is of the abrasive type and assumes that four times as much wear results from abrasive as from lack of lubrication.

Groups I, II, III-*a*, and IV-*a* of Table 1 are the combinations of materials in use in lathes at the present time. A study of the data for these four combinations shows that the two combinations in which cast-iron forms, both rubbing surfaces wear far faster, particularly in abrasive, than the two combinations in which fully hardened steel is one component. This is attributed to the fact that a cast-iron surface embeds abrasive in its pores which causes wear of the opposing surface. When both surfaces are cast iron, both will wear rapidly because the hardness of the abrasives exceeds even that of flame-hardened cast iron. When fully hardened steel is one component, there is little wear because it does not embed abrasive to wear the opposing cast-iron surface and is too hard to be worn much by the abrasive embedded in the cast iron. Thus if cast iron were run against softer steel, which could be worn by the abrasive embedded in the cast iron, rapid wear, particularly of the steel, would be expected to occur. The data for groups IV-*c* and IV-*d* show that this does happen.

The unlubricated wear of the two cast iron - cast iron combinations was also appreciably higher than that of the two hardened steel - cast iron combinations, though this fact is considered less serious.

The scoring resistance of the flame-hardened cast iron - plain cast iron combination was the highest. It was not, however, so

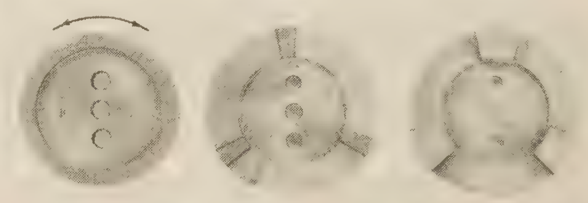

Fig. 2 The Specimens Used in the Scoring Tests

(The one on the left oscillates as indicated by the arrow, with the annular face in contact with the three narrow (10 deg and 30 deg) raised segments of either of the two specimens on the right. The bearing faces are lapped optically flat for test. For the tests reported in Table 1, the angular motion was 25 deg, and the segmental specimen was like the one on the right with three 30-deg segments. Tests were run in an oil bath at 100 F for 15 min at 570 cycles per min. No scoring starts after this time.)

Fig. 3 The Typical Microstructure of a Chilled-Cast-Iron Bed; ×100

(The matrix is fully pearlitic and the graphite is in spidery clusters. This iron is neither as wear-resistant nor as scoring-resistant as the alloy iron in Fig. 4.)

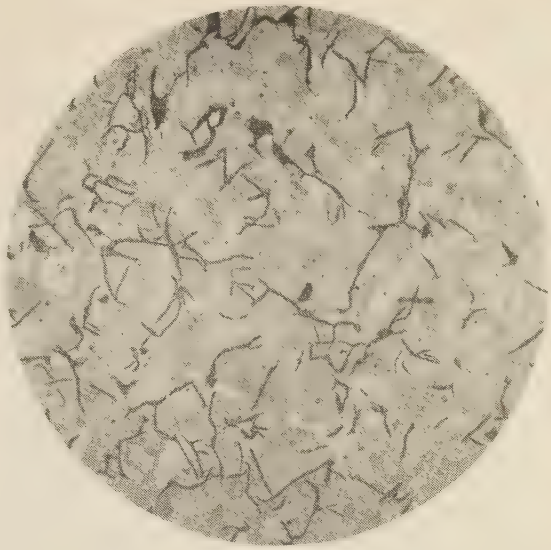

Fig. 4 The Typical Structure of an Alloy-Cast-Iron Bed

(The graphite is in small flakes randomly distributed. This iron is quite wear- and scoring-resistant.)

much higher than the scoring load of combination IV-*a* as the data would indicate at first glance, for the plain-cast-iron sample tested against all the flame-hardened irons was highly scoring-resistant, and when it was tested against hardened steel, scored at the highest load of all the iron tested in combination IV-*a*, 360 lb. Thus it seems that the combination of flame-hardened cast iron and plain cast iron is inherently only about 50 per cent

TABLE 1 AVERAGE RESULTS OF WEAR AND SCORING TESTS ON VARIOUS LATHE BED-CARRIAGE COMBINATIONS

Combination no.	Combination of materials	Vickers hardness	Average wear rate —0.001″ per hour: Abrasive wear: Scale	Mod. scale	Cast iron	Unlubricated wear: Dry air	Moist air	Relative wear index	Scoring load, lb
I	Cast iron	200–230	0.640		0.060	0.025	0.009		
	Cast iron	217	1.13		0.064	0.033	0.013		
			1.77		0.124	0.058	0.022	44.6	
II	Flame-hardened cast iron	450–610	0.049		0.009	0.008	0.007		
	Cast iron	198–217	0.234		0.029	0.044	0.019		
			0.283		0.038	0.052	0.026	8.9	490–590*
III-*a*	Chilled cast iron	225–230	0.013		0.005	0.005	0.035		
	Fully hardened steel	765–820	0.009		0.003	0.001	0.005		
			0.022		0.008	0.006	0.040	1.4	170–210
III-*b*	Ferritic chilled cast iron	195	0.028		0.018	0.046	0.072		
	Fully hardened steel	800	0.012		0.006	0.018	0.002		
			0.040		0.024	0.064	0.074	3.8	70–100
IV-*a*	Alloy cast iron	205–280	0.011	0.080	0.004	0.004	0.005		
	Fully hardened steel	765–820	0.008	0.072	0.004	0.003	0.004		
			0.019	0.152	0.008	0.007	0.009	1.0	216–360
IV-*b*	Ferritic alloy cast iron	140	0.027		0.017	0.022	0.066		
	Fully hardened steel	800	0.014		0.016	0.049	0.002		
			0.041		0.033	0.071	0.068	4.3	
IV-*c*	Alloy cast iron	270	0.115		0.005	0.032	0.002		
	Tempered steel	600	0.20		0.009	0.033	0.012		
			0.315		0.014	0.065	0.014	8.4	
IV-*d*	Alloy cast iron	270	0.222		0.016	0.026	0.005		
	Tempered steel	435	0.72		0.013	0.053	0.004		
			0.942		0.029	0.079	0.009	22.5	
V-*a*	Flame-refined cast iron	230–300	0.007	0.056	0.002	0.004	0.004		
	Fully hardened steel	765–820	0.006	0.052	0.007	0.002	0.007		
			0.013	0.108	0.009	0.006	0.011	0.8	316–440**
V-*b*	Flame-refined ferritic irons	240–260	0.012		0.006	0.004	0.009		
	Fully hardened steel	765–820	0.006		0.004	0.002	0.002		
			0.018		0.010	0.006	0.011	1.1	

* Fifty per cent higher scoring load than the same piece of plain cast iron against hardened steel.
** Each of the hardened irons scored at a 50 per cent higher load than the same iron not flame-treated.

more resistant to scoring than hardened steel and plain cast iron, instead of 100 per cent better as the data of Table 1 would indicate.

These results indicated that combinations of hardened steel and cast iron were inherently more wear-resistant than combinations of cast iron and cast iron, so the former were investigated more thoroughly. A comparison was made between two types of beds, chilled and alloy iron, being made for the manufacturer sponsoring this work. The former is an iron containing approximately 3.3 per cent carbon, 0.40 per cent chromium, and up to 0.50 per cent copper cast against chills on the wearing surface to produce a fine-grained iron; the latter contains approximately 3.0 per cent carbon, 0.20 per cent chromium, 0.75 per cent copper, and 0.30 per cent molybdenum, cast without chills but just as fine-grained as the chilled iron because of its composition. The photomicrographs of Figs. 3 and 4 show that there is far more difference in microstructure than the slight composition differences would indicate. The chilled iron has what is called a "modified" graphite flake, the alloy iron a normal graphite flake.

Wear and scoring tests showed that chilled beds wore faster and scored at lower loads than unchilled beds. Chilled beds have another disadvantage in that foundries making chilled beds are more likely to produce an occasional ferritic bed than a foundry making alloy-iron beds, and as a comparison of combinations III-*b* and IV-*b* with III-*a* and IV-*a* will show, ferritic irons are extremely poor bearing surfaces. The manufacturer discontinued the use of chilled beds when this information was available and has obtained the indicated improvement in performance.

Investigation of the effect of composition of the cast iron on the performance of combination IV-*a* showed that the composition given was as good as any. Increase of the carbon content or small changes in the alloy content do not greatly change the rate of wear or the scoring load, so long as the iron is fully pearlitic. Ferritic irons like combinations III-*b* or IV-*b* must be avoided or rapid wear and scoring will be experienced. Low-carbon irons too must be avoided, for one fully pearlitic iron with only 2.53 per cent carbon scored at an extremely low load, 140 lb. High-alloy irons, containing considerable amounts of free carbide, wore a little more slowly and scored at somewhat higher loads, but they were expensive and extremely difficult to scrape because the many carbide particles dulled scrapers with extreme rapidity.

Effects of Heat-Treatment

The results of the composition study provided little hope that any considerable improvements could be obtained from modifications of composition alone. Heat-treatment was then tried as a means of obtaining the desired improvement. From furnace heat-treating experiments, it was found that better results were

Fig. 6 The Microstructure of the Unaffected Zone of the Bar of Iron, Fig. 5; $\times 500$

(The iron is comparatively soft, 190 Brinell, as the coarseness of the pearlite would indicate.)

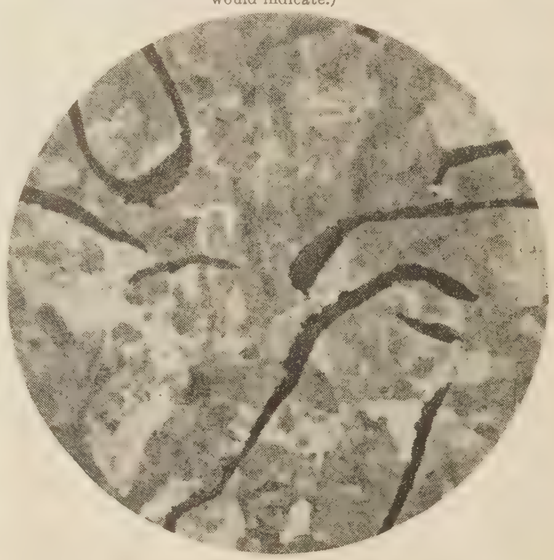

Fig. 7 The Microstructure of the Flame-Refined Part of the Bar of Iron, Fig. 5; $\times 500$

(The pearlite lamellae are much finer, the hardness has been increased to 280 Brinell, and the iron is more wear- and scoring-resistant.)

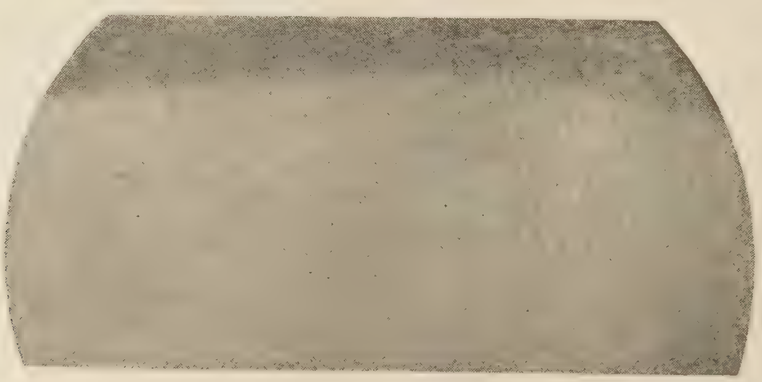

Fig. 5 The Cross Section of a Flame-Refined Bar of Cast Iron, Showing the Depth of the Hardened Zone; $\times 2$

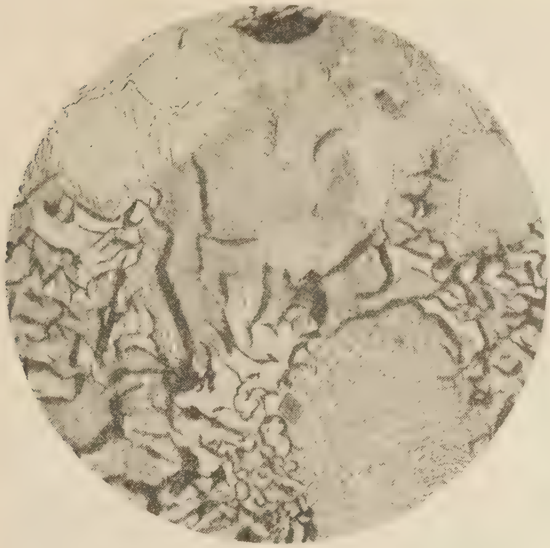

Fig. 8 The Microstructure of a Modified Ferritic Type of Iron; ×500

(The white patches of free ferrite in dispersed grains and between the clusters of graphite flakes form approximately 25 per cent of the structure. This type of iron scores readily and wears rapidly.)

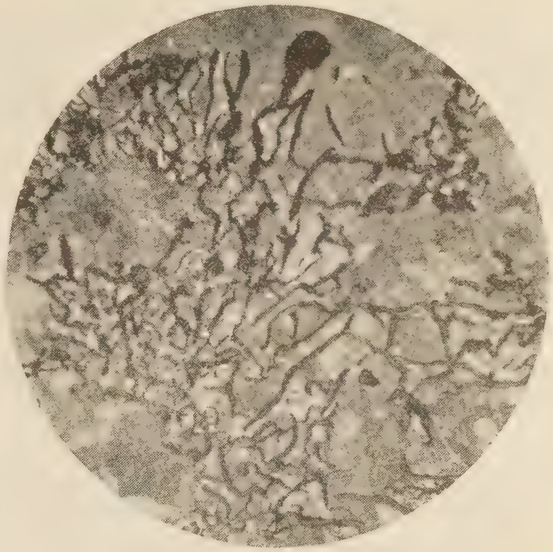

Fig. 9 The Same Iron as in Fig. 8 After Flame-Refining; ×500

(The ferrite has been almost entirely converted to pearlite, and the pearlite has a finer grain size. The hardness of the iron has been increased from 180 to 225 Brinell. The wear and scoring resistance is almost as good as for a flame-refined originally pearlitic iron.)

obtained by cooling comparatively rapidly from above the critical temperature, to produce an extremely fine pearlitic structure, than by quenching and drawing to the same hardness to produce a sorbitic structure.

The experiments were extended to sections of beds, and it was found that the desired structure and wear resistance could be obtained by heating the surface with a flame-hardening torch to above the critical temperature and depending on the heat drain from the cooler interior of the bed to produce the proper rate of cooling. It was found that the temperatures of the underlying bed must be somewhat elevated, by making a preheating run with the torches, to prevent too rapid cooling and excessive hardness. This process is called "flame refining"[2] to distinguish it from flame hardening where a quench is essential and full hardness is desired. By experiment, it was found that these heat-treated irons could be scraped at hardnesses up to 30–35 Rockwell C if carbide-tipped scrapers were used. The structure of an iron before and after treatment is shown in Figs. 6 and 7 where it is evident that the pearlite is greatly refined by the process. The depth of hardening is apparent in the cross section illustrated in Fig. 5.

Wear and scoring tests of these heat-treated irons showed that wear in abrasive was only 70 per cent as great as that of the same irons before heat-treatment, and that the scoring load was 50 per cent greater than for the same irons not hardened. The data for the tests are given under combination V-*a* in Table 1. The heat-treating process performs even more strikingly on ferritic irons, as a comparison of the data for combination V-*b* with III-*b* and IV-*b* shows. It is to be expected that the flame-refining process would have such an effect, for it eliminates free ferrite from the structure, as a comparison of Figs. 8 and 9 shows.

Simulated Service Tests

The results up to this point indicated that a combination of fully hardened steel and flame-refined cast iron was the most wear-

[2] Patent applied for.

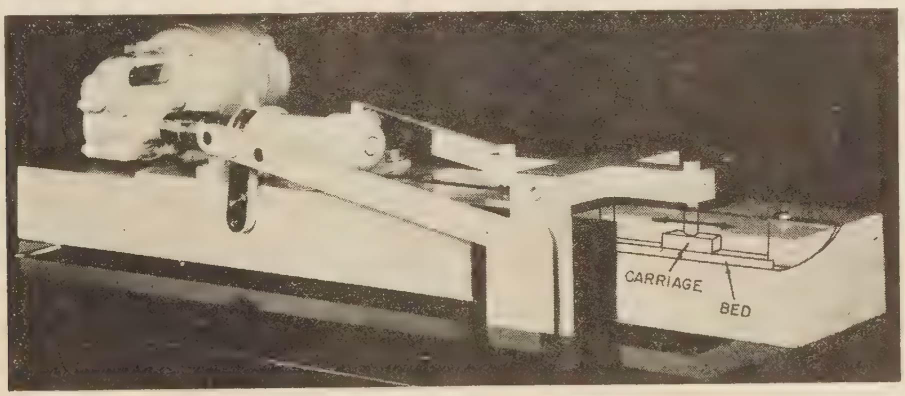

Fig. 10 A Reciprocating Wear-Testing Machine Used for Simulated Lathe-Bed Service Wear Tests

(The form of the specimens is shown in phantom outline. The specimens were run in the same slurry of heat-treating scale as the preliminary wear tests. The machine operated at $11^1/_2$ cycles per min with a 4-in. stroke. The bearing load was 15 psi.)

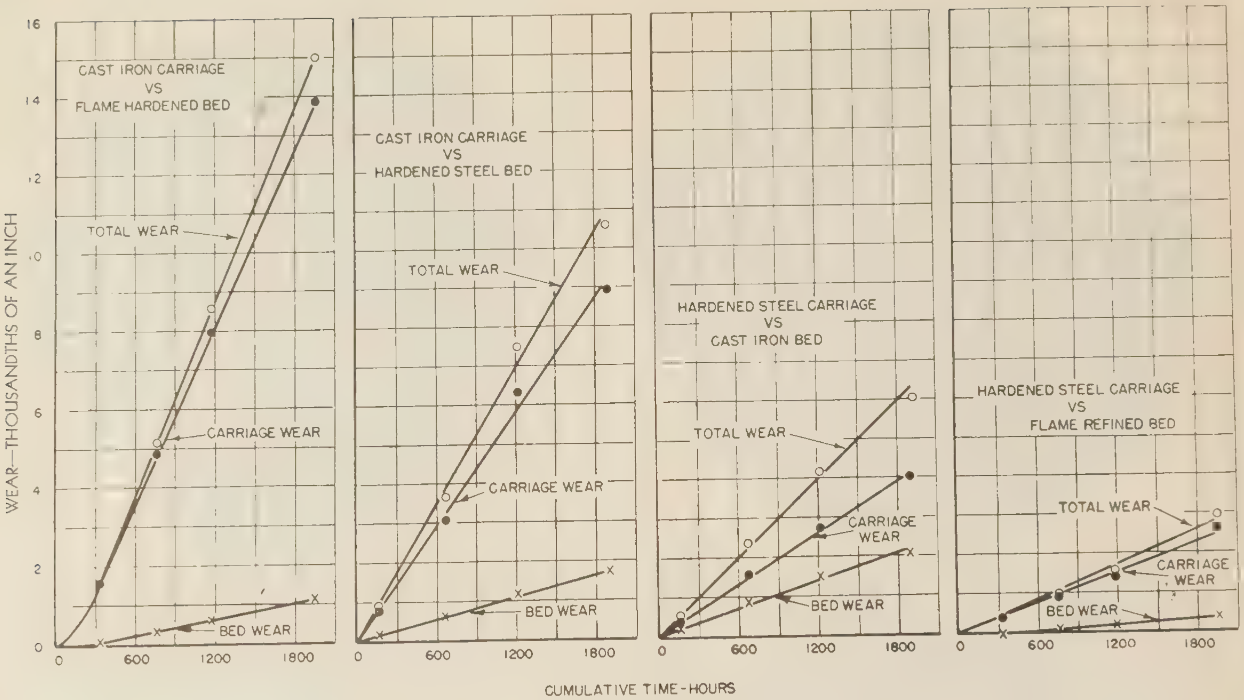

Fig. 11 Wear of Materials in Simulated Service Tests

resistant, but did not show which of the two materials should be the carriage and which the bed. To determine this, as well as to obtain additional confirmation of the results up to this point, long-time abrasive wear tests were made in an apparatus which simulated the wearing action of a lathe. The apparatus is illustrated and described in Fig. 10. The results for four combinations of materials are given in Fig. 11. The materials used in this test were those which had been proved by the earlier work to be the best of their types.

The first combination of a cast-iron carriage and a flame-hardened bed shows little wear of the bed, but relatively enormous wear of the carriage. After test, the originally square ends of the carriage were rounded off, indicating that they had acted as wedges, permitting a great deal of abrasive to work between the surfaces.

The second combination, again of a cast-iron carriage but with a fully-hardened-steel bed, showed slightly more bed wear but considerably less carriage wear. However, the carriage wore at the ends the same as against a flame-hardened bed.

The third combination of an alloy-cast-iron bed and a fully-hardened-steel carriage showed slightly more wear of the bed but far less wear of the carriage than the other two. The carriage remained square at the ends.

The fourth combination of a flame-refined cast-iron bed and a fully-hardened-steel carriage was superior in every respect. Carriage wear was only 60 per cent as great, bed wear only 25 per cent as great, and total wear only 45 per cent as great as on the best of the other combinations in these individual respects.

Conclusions

The results indicate that the combination of a carriage faced with fully-hardened-steel strips[3] and a cast-iron bed flame-refined to the maximum allowable hardness for scraping will wear far less than any other combination. The combination appears to be about as scoring-resistant as any other, and 50 per cent better than unhardened cast iron against hardened steel. Since service records show no scoring difficulties with this latter combination, except for infrequent ferritic beds, the scoring resistance appears more than adequate.

Acknowledgment

This investigation was made for the Lodge and Shipley Machine Tool Company of Cincinnati, Ohio. The assistance of Mr. W. L. Dolle and Mr. Fred Schoeffler of that organization, and of Dr. H. W. Gillett and Dr. H. W. Russell of Battelle has been greatly appreciated.

[3] U. S. Patent No. 2,167,609.

Porous Chromium in Engine Cylinders

By RUSSELL PYLES,[1] OLEAN, N. Y.

In recent years it has been possible to deposit thick coatings of chromium electrolytically on machined parts, where lubrication is a vital factor. Diesel-engine cylinders and piston rings may be included among the most critical engine parts from a wear standpoint, and in this application, porous chromium has demonstrated its ability to increase the life span of an engine. The author outlines the properties inherent in porous chromium which contribute to its low wear rate in engine cylinders, i.e., hardness, exceeding that of any other cylinder-lining material; high thermal conductivity; low friction coefficient, and low affinity for other metals; corrosion resistance, preventing surface deterioration.

THE element chromium was discovered in 1798, but not until 1859 was it isolated by F. Wohler who obtained small quantities by reduction of the trichloride. As early as 1848–1854 (1)[2] efforts were being made to electrodeposit chromium. Decorative chromium as thin coatings of 0.00001–0.00002 in. became a commercial fact about 1924 (2). It is interesting to note, too, that the metallurgical advantage of chromium as a steel-alloying agent was put to use in 1870, when Midvale produced 2400 tons of 1 per cent chrome steel for the Eads Bridge over the Mississippi River at St. Louis (3).

Only in the last few years, however, have thick coatings of porous chromium been developed for machine applications where lubrication is a vital factor (4). Cylinders and piston rings are among the most critical engine parts from a wear standpoint. Here porous chromium has demonstrated that it will increase the life span of an engine.

Properties of Electrodeposited Chromium

Hardness. Hardness alone is not accepted as being a criterion for low wear but it is true, that for the same material, wear is lessened when hardness is increased. This has been indicated by many tests on cast-iron engine cylinders and heat-treated steels. Hardness, though, may be a dominant factor where erosion is a problem. Chromium is hard but some variations in hardness are obtainable by control of plating factors, such as bath composition, current density, and temperature. However, within the usual limits of these plating factors the hardness variation is slight.

TABLE 1 HARDNESS TABLE

	Brinell hardness (load 3000 kg; 10-mm ball) or equivalent
Chromium	800–1000
Cast-iron cylinder compositions	150 275
Cast-iron cylinder compositions, heat-treated	Max 400
Steel 4140 heat-treated	250–375
Steel, carburized	625
Nitralloy in cylinders after removal of surface stock	650–750

Chromium is harder than any other of the metals in general use for engine cylinders, and this hardness is maintained throughout the thickness of the case while the hardness of nitralloy or carburized steels decreases with depth. The hardness of the base metal under the chromium is not such an important factor in cylinders, although it may be in other special applications. Crushing loads, due to piston-ring and gas pressures, are relatively light in engine cylinders and porous chromium performs just as well on soft steels and cast irons as it does on nitralloy. Thus, the engine designer may choose his basic cylinder material without any regard for its wear characteristics. Alloy irons are not necessary, except as determined by structural-strength requirements. Cylinders may be of low-carbon steel for good welding where the cylinder assemblies are fabricated for lightness. These economic features are important in normal times and even more so now, when alloys are so difficult to obtain.

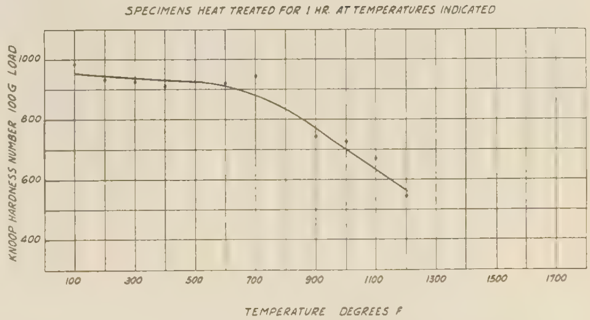

FIG. 1 KNOOP HARDNESS, POROUS-CHROME CYLINDER
(0.0045 in. chromium on cast iron.)

The hardness numbers in Table 1 are given for comparison in Brinell numbers, but it should be noted that the Brinell test is not used for the higher hardnesses and particularly for determining hardness of thin coatings. Because usual hardness tests, such as Rockwell, Shore, and Brinell, cover relatively large areas and are influenced by an appreciable depth of the underlying metal, they do not give a true picture where thin hard coatings are to be tested. The Knoop indentor (5) overcomes this difficulty by using a small diamond pyramid with shallow indentation. This machine is well adapted to use on porous chrome because the short length of the impression (approximately 0.004 in.) readily fits into the plane areas between the pores.

Contrary to many metals, chromium maintains its hardness very well within the range of cylinder-wall operating temperatures, Fig. 1.

TABLE 2 EXPANSION COEFFICIENTS AND THERMAL CONDUCTIVITY

Metal	Linear thermal expansion at 68 F × 10^{-6}	Thermal conductivity, cal/cm²/cm/deg C/sec at 20 C	Melting point, deg F
Chromium, electrolytic	4.5	0.165	3325+
Cast iron	6.6	0.12	2500
Steels	6.2–6.6	0.11	2700–2800
Aluminum	12–13	0.52	1216
Copper	9.1	0.92	1981

The linear-expansion coefficient of chromium is lower than that of cast iron or steel, yet no cases of bond failure due to this have been noted. The surface temperature in a cylinder is higher than that of the underlying metal, there being a steep

[1] Chief Engineer, Van der Horst Corporation of America.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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temperature gradient through the cylinder wall. This condition, for minimum heat stresses, theoretically calls for an expansion coefficient varying from a lower value at the inner wall to a higher value at the outer cylinder wall. An inside layer of chromium partially fulfills this ideal requirement. Chromium has been plated on the aluminum piston crown of a Lauson engine $1^3/_4$ in. bore $\times$ $1^7/_8$ in. stroke and run for 54 hr without any signs of bond failure or change in surface appearance under the microscope. Many cases of record in high-output air-cooled airplane engines, with 700 to 1000 hr, show no tendency of the chromium to loosen due to differential expansion between the chromium and the steel. These engines are running with cylinder-wall temperatures much higher than those encountered in stationary-engine service. This evidence indicates that the low expansion of chromium may be a desirable feature for cylinder walls.

It should be noted that the thermal conductivity of chromium is approximately 40 per cent higher than for steel or cast iron. Maximum metal temperatures of the cylinder are at the inner cylinder surface particularly in the combustion zone. Any improvement in heat transfer results in a lower wall temperature and will improve piston and ring lubrication. Heat actually gets away faster through chromium. Tests on air-cooled porous-chrome cylinders have shown that more heat passes to the barrel fins because a greater quantity of cooling air on the barrel is required to maintain a standard thermocouple temperature. Other tests have shown a higher than normal temperature at the cylinder flange, indicating greater heat transfer to the barrel. Specific fuel consumption was normal so there is no reason to believe that there was any increase in total heat loss. This additional cylinder heat then must have been subtracted from heat loss to the cylinder head, piston, and lubricating oil. It is a fact that in some engine types the bottleneck for heat dissipation lies in the cylinder head. The heat distribution for an air-cooled airplane-cylinder assembly is of the order of 60 per cent to the head and 40 per cent to the barrel. Any diversion of heat to the barrel is evidently very desirable and chromium seems to accomplish this.

The question naturally arises concerning heat transfer through the bond where two dissimilar metals are joined. Inspection of photomicrographs of porous chromium applied directly to iron or steel shows no space or void between the two metals. Of course, the basic metal must be absolutely clean before plating but this is readily accomplished by a short electrolytic etch, which, at the same time, produces some mechanical keying and interlocking on the basic-metal surface.

In some plating the metal is transferred from the anode to the cathode, but in chromium plating, the chromium comes directly out of the bath solution which is essentially chromic acid (CrO_3 H_2O). It is deposited as molecular chromium on the cathode, the molecules being of insignificant size dimensionally. Thus the chromium fills in the smallest irregularities of the surface being plated and a bond approaching fusion such as in welding is obtained. Chromium under the microscope even at high magnification exhibits no crystalline structure such as we see in steels, but the crystalline structure has been indicated by X-ray diffraction. Because of the fineness and intimacy of this bonding good heat transfer results. Mechanical failure at the bond is the least of our worries, calling only for meticulous care in the cleaning of the basic-metal surface.

Friction. Chromium is conceded to have the lowest coefficient of friction of any of the generally used structural metals. The value of the sliding coefficient for chromium on chromium has been given as 0.12 and for chromium on steel as 0.16, while steel on steel is 0.20 (6). It should be pointed out here that our experience with chromium running on chromium under boundary lubrication has been unsatisfactory.

For rotating shafts chromium was found to have the lowest friction of any of the metals tested (7). Friction in an engine cylinder can hardly be estimated from any simple friction tests, and no actual engine tests have yet been run by the author's company to check comparative friction losses with porous-chrome cylinders.

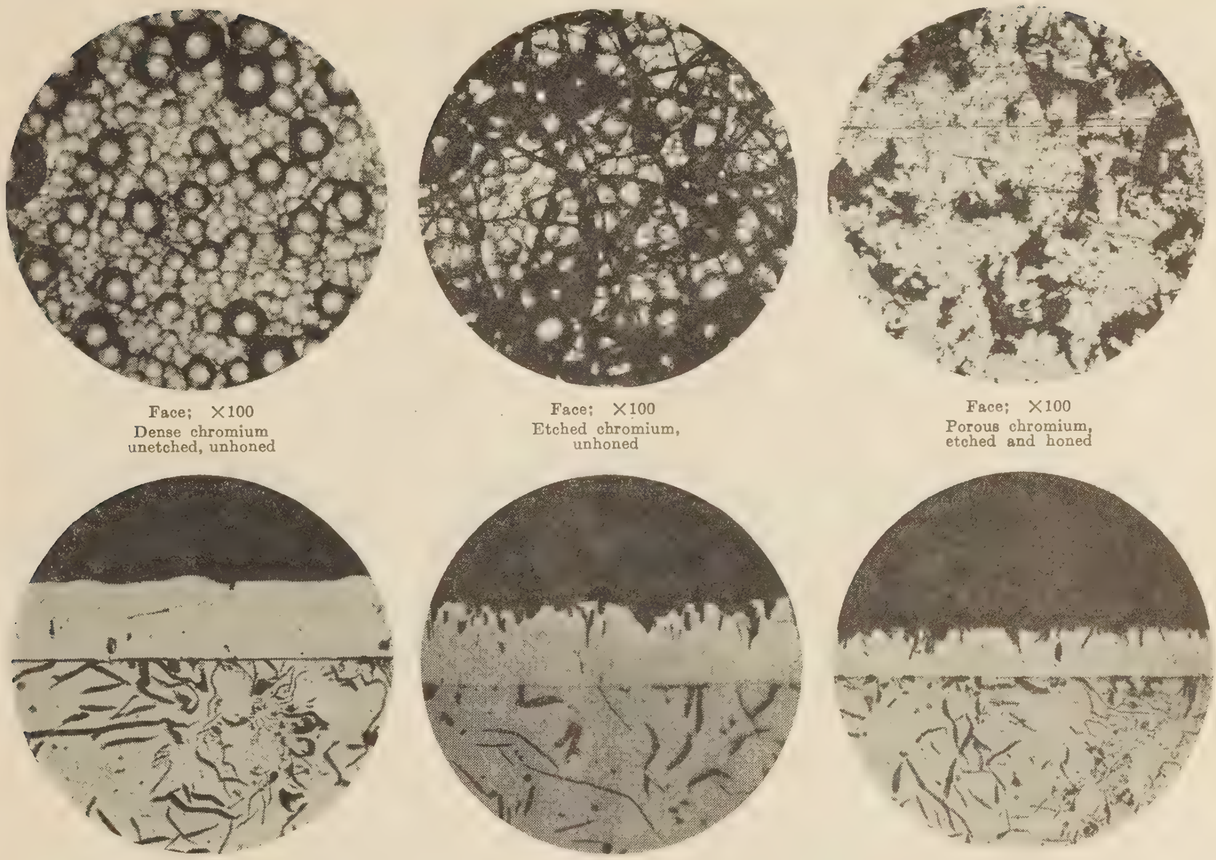
Corrosion Resistance. The corrosion resistance of chromium is high and is an important contributor to its low wear rate in engines. Sulphur is always present to some degree in Diesel fuels ranging from $^1/_2$ to 3 per cent by weight. At the high temperature and pressure existing in the cylinder, sulphur may combine with oxygen to form sulphur trioxide (SO_3) which, with water as a product of combustion, forms dilute sulphuric acid. Chromium is only very slightly attacked by sulphuric acid but ferrous metals are quickly and vigorously attacked. Another sulphur compound, hydrogen sulphide, may be encountered as a corrosive agent, particularly in gas engines running on sour gas.

A steel rod plated over a part of its length was subjected to moist hydrogen sulphide at temperatures ranging from 120 F to 220 F for 252 hr. After this exposure the unplated portion of the rod was blackened and badly corroded, but the chromium-covered portion was unattacked. This resistance to hydrogen-sulphide corrosion has been confirmed in service by excellent performance in gas engines operating on natural gas with an H_2S content as high as 1 per cent by volume. In spite of the resistance of chromium to corrosion from sulphur compounds they should always, in so far as possible, be removed from gaseous fuels because their tendency to increase fuel knock adversely affects combustion. Chromium is unaffected by nitric acid and saturated solutions of ammonia, but it is dissolved by hydrochloric and bromic acid.

Chromium resistance to atmospheric corrosion is well recognized. Porous-chrome cylinders may be stored indefinitely with little or no protection to the chromium surface and no detrimental effect results.

Having mentioned some of the good characteristics of chromium, it is well to note also some of the deficiencies so that its limitations may be recognized. Its tensile strength seems to be negligible. We have made rather crude tests for tensile strength of electrodeposited chromium and obtained a value of around 2000 psi. Chromium was deposited on a copper strip to a thickness of 0.005 in. and the copper was then dissolved by nitric acid, leaving 0.005 in. of chromium. This was very fragile and brittle but with care was clamped and pulled. Search of existing literature fails to reveal any values for tensile, shear, or compressive strength, yet this information is desirable when chromium is used as an engine material. However, chromium, in common with other brittle materials such as cast iron, must have appreciable compressive and shear strength as evidenced by its performance in engine cylinders and on piston rings.

Porous Chromium. Porous chromium has all of the characteristics peculiar to chromium but differs principally in its surface condition. This is responsible for its ease of lubrication and ability to carry high bearing loads. Fig. 2 shows face and cross sections of chromium as used on a cylinder wall. When chromium is electrodeposited to a thickness of 0.004 to 0.015 in., it becomes nodular and exhibits a network of cracks. Thin deposits of the order of 0.00001 to 0.00005 in., as used in decorative work also invariably show checking, except that they may be smeared over by cold working in buffing or polishing operations. These cracks are due to tensile stresses beyond the ultimate tensile strength of the chromium. They are shrinkage cracks probably resulting from the evolution of hydrogen (8). In the plating process, hydrogen is generated at the cathode (cylinder wall), and the chromium as deposited may contain large volumes of occluded hydrogen.



Face; ×100
Dense chromium
unetched, unhoned

Face; ×100
Etched chromium,
unhoned

Face; ×100
Porous chromium,
etched and honed

Cross Section; ×100

Cross Section; ×100

Cross Section; ×100

FIG. 2 FACE AND CROSS SECTIONS OF CHROMIUM AS USED ON A CYLINDER WALL

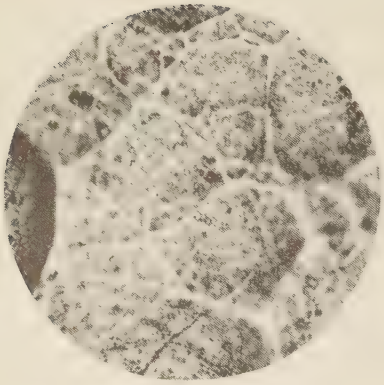
When this hydrogen diffuses out, a change in volume occurs, the result being, for analogy, a condition similar to dried mud from which water has evaporated. These cracks are not the grain boundaries as commonly seen in etched steel. Because tensile failure has occurred along these lines it cannot be assumed that the chromium is stress-relieved. Appreciable stresses still exist throughout the plate, but they are below the ultimate strength. Electrolytic action preferentially attacks along stress lines or planes and the current-reversing process, which is the treatment for producing porous chromium, does exactly this (9). As the reversing process proceeds, the existing cracks become wider and deeper at the same time they tree or sprout branches until the whole surface is completely cut up, appearing under the microscope like a coal pile. Fig. 3 shows well the effect of progressive electrolytic etching or reversal. This porosity treatment effects a great reduction in residual stresses and may be responsible for the slight reduction in endurance limit found on porous-chrome-plated steel specimens (10–15 per cent, while dense unetched chromium may show a reduction of as much as 50 per cent).

Strips of 0.005-in-thick shim stock were plated with the results shown in Fig. 4. Upon plating the strip (*a*) on one side only, it assumed the curvature shown at (*b*), yet the chromium surface on microscopic inspection showed no cracks. Shrinkage of the chromium occurred, but due to the thinness of the steel base strip, it conformed, and the stresses in the chromium did not exceed its ultimate strength. However, the application of reverse current (*c*) produced a reduction in curvature and also many surface cracks, demonstrating that residual stresses were relieved. Strip (*d*) was plated on both sides with the same thickness of chromium. It remained substantially straight, but the chromium surface on each side was cracked exhibiting tensile surface failures because the stresses of deposition were not relieved by conformation or current reversal.

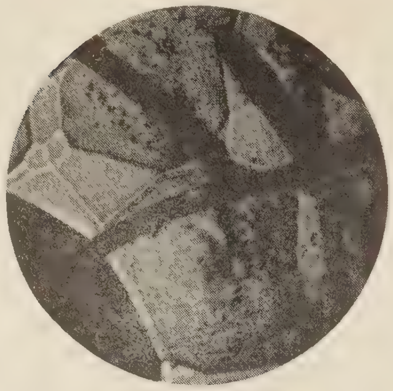
The current-reversing process may be carried to the point of complete removal or stripping of all of the deposited chromium, but in the production of cylinders it is stopped short of any appreciable dimensional change in the bore diameter. Honing of the bore then results in a smooth surface containing microscopic indentations, recesses, or pits.

Wettability. The term "wettability" has been applied to surfaces and oils to define that property of either which affects the rate of oil dispersion. It is a function of both the oil and the surface. Oils vary in wetting characteristics. Surface geometry and profile also play a major part. The porous-chromium surface after honing, with its flats dotted with microscopic pits, depressions, or crevices is readily wetted with oil, which spreads quickly and covers. It has the further advantage of oil retention. Oil may be scraped from the flat surfaces but some remains in the pores, and this will creep up and rewet.

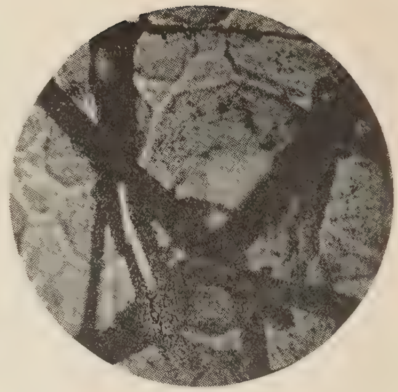
The mechanism of wettability, which has been ably discussed by J. T. Burwell (10), is such that an oil drop on a plane smooth surface will reach a state of equilibrium, where it will neither spread nor recede. At this condition the dome-shaped drop makes a definite contact angle with the plane surface, this contact angle depending upon inherent characteristics of the oil.



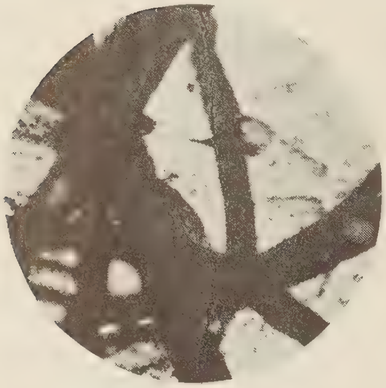
1 Face; ×500



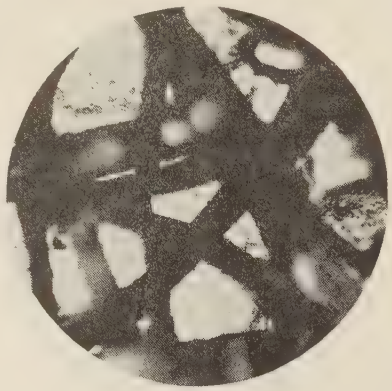
3 Face; ×500



6 Face; ×500



9 Face; ×500



12 Face; ×500

15 Face; ×500

Fig. 3 Effect of Progressive Electrolytic Etching on Chromium

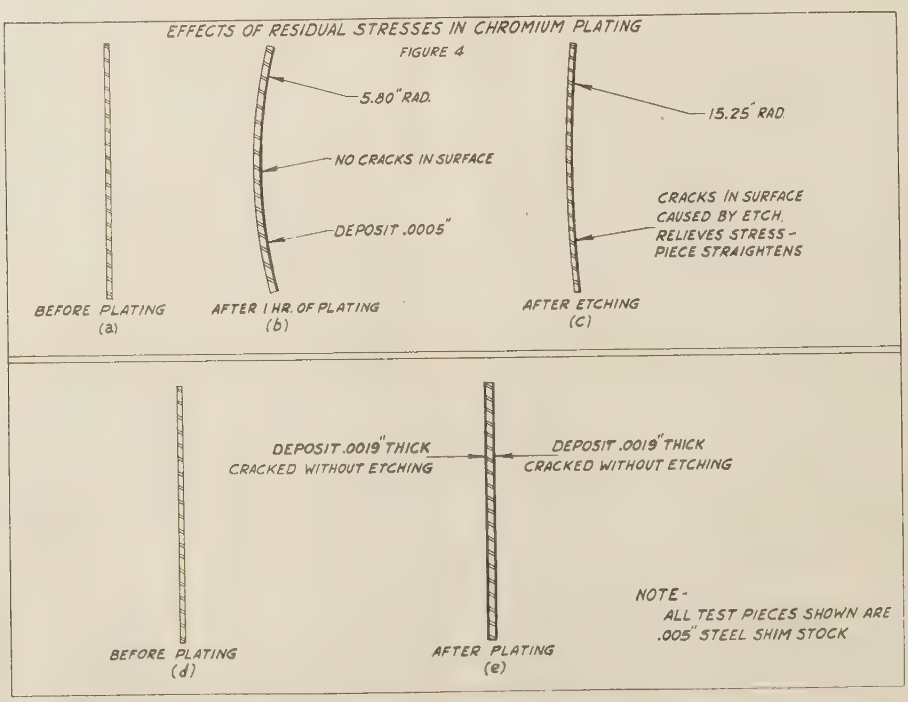


Fig. 4 Effects of Residual Stresses in Chromium Plating

itself. The energy of adhesion of the oil W is given by $W = \alpha (1 + \cos \theta)$ (10), where α is the surface tension and $\cos \theta$ is the contact angle.

A $^1/_4$-in-diam area on porous chromium may contain fifty to several hundred pores or crevices depending upon the "porosity." Taking a section through the diameter of such a surface with an oil drop on it, we will find the edge of the drop may be on a flat surface and at this point it will stop spreading when the equilibrium contact angle is reached. However, there is an infinite number of diameters that may be taken through the oil drop and every one will show a different geometrical condition of the porous-chrome surface at the edge of the drop. Circumscribing a circle on a photomicrograph of the porous-chrome surface, it will cut through flats, the downslope of pores, and the upsloping sides. As a result, the condition at the periphery of drop is unstable, because the contact angle of equilibrium can never be reached at all points. The oil thus proceeds to execute enveloping tactics around the flat areas, and to spread over the whole surface.

Our experience indicates that on any metal surface which has roughness, for instance, scratches running in all directions, oil will spread faster than when the surface is smooth. On a specimen polished in one direction, the oil will also spread faster in the direction of the polish scratches than it will at 90 deg to them. In this case the drop becomes elliptical, the major axis being in line with the scratches. This action is explainable by the variation of the contact angle at the edge of the oil drop.

Load Capacity of Porous Surfaces

With full oil-film lubrication the capacity of a bearing is a function of bearing area if other factors such as viscosity and speed are constant. Crankshaft bearings operate, at least during a large part of the cycle, under conditions approaching full film lubrication. An increase of bearing area here results in an increase in load capacity generally. Under conditions of boundary-layer lubrication such as exists between piston rings, piston, and cylinder wall, the case is reversed and load capacity increases with a decrease in bearing area. This is entirely at variance with accepted theories of hydrodynamic lubrication, but this type of lubrication cannot be maintained on cylinder walls. Oil control requires removal of oil from the cylinder wall, what remains being subjected to high temperature and combustion flame as well as dilution from fuel or condensation of combustion products. The dynamic factor of velocity is a requisite for hydrodynamic oil-film maintenance, yet at the top center position of the piston, where all of the adverse conditions accumulate, the piston-ring velocity is zero. Boundary-layer lubrication is the best that can be expected in the upper ring travel.

Notable investigations of boundary-layer conditions have been made by F. P. Bowden (11) and collaborators. Macy O. Teetor (12) has analyzed and experimented on these conditions for the specific case of piston rings and cylinders. The results are startling and establish some novel concepts of factors involved in boundary-layer lubrication. Evidence seems to indicate that scuffing or scoring results when the surface temperatures of the metals in contact reach the melting points, then fusion or welding occurs. These surface temperatures, even under conditions of smooth running and low friction, have been found to exceed 1100 F, yet the body of the test specimen remained at room temperature (11 *b*).

That these temperatures can sometimes exceed the melting point of cast iron is evidenced by inspection of seized cast-iron pistons where we may find some of the cylinder-wall material welded on the piston or the reverse condition of piston surface welded on the cylinder wall. The factors producing heat at the surface are speed, load, and frictional characteristics of the metals in contact, but friction depends to a large degree upon the surface geometry (rough or smooth), and the actual net area of contact. The larger the contact area the greater will be the amount of heat generated. Cooling of the surface is accomplished partially by oil but mainly by heat transfer into the base material. Teetor's work (12) demonstrates that reducing contact area by interrupting the surface results in a smaller amount of generated heat because by so doing he was able to carry higher bearing loads. The low limit for contact area may be dependent upon the physical characteristics of the contacting metals, particularly their shear and compressive strength at operating temperature. The amount of heat that can be carried away from the surface depends upon the thermal conductivity, and the limiting operating surface temperature is fixed by the melting points of the contacting materials. Briefly, the following steps are indicated for increasing load capacity under boundary-lubrication conditions:

1 Reduce the net actual contact area.
2 Choose materials with high thermal conductivity.
3 Choose materials with high melting points.

It is interesting to note how porous chromium fits into these requirements:

1 Inspection of face and cross sections of Fig. 2 shows that contact area is reduced by the pores or crevices. This we call porosity and have come to speak of it on a percentage basis; thus 30 per cent porosity signifies 30 per cent of the total area in depressions and 70 per cent of the area as flats, plateaus, or contact surface. This percentage porosity is to some extent controllable in the electrolytic processing and in the finish-honing.

2 The thermal conductivity of chromium is 40 per cent higher than steel or cast iron, permitting rapid heat flow for the maintenance of lower surface temperature.

3 The melting point of electrolytic chromium is approximately 800 F higher than cast iron and 500 F higher than steel, thus raising the limit for operating-surface temperature.

The exceptional performance of porous-chromium cylinders and piston rings in severe service is a fact and lends credence to these explanations of why it works.

Processing. Before applying porous chromium to a cylinder it should be ground or bored oversize and then honed to a smooth finish of 10 μ in. rms or less. A fine finish is desirable for the following reasons:

1 It gives a more definite calculable actual surface area for plating. Control must be maintained over the thickness of deposited chromium and with a given plating solution this depends upon area, current density, and time. The latter two are controllable but the actual net area is inherent in the part to be plated.

2 Deposition of chromium tends to exaggerate initial surface irregularities; a rough surface becomes rougher as the thickness of deposition builds up. As a finished cylinder must always come within both dimensional and surface-finish tolerances, it is desirable to have a uniform thickness of chromium throughout the bore.

Cracks, inclusions, and spongy spots in the cylinder wall are accentuated by plating; in fact the process displaces the necessity for magnaflux checking. Cracks which are invisible to the eye are clearly shown up after plating.

The cylinders are then assembled in fixtures with accurately centered anodes, given a short anodic treatment, immersed in a plating tank, and plated with the required thickness of chromium. The fixture-and-cylinder assembly is then removed as a unit to an etch tank where it receives a porosity treatment. Upon

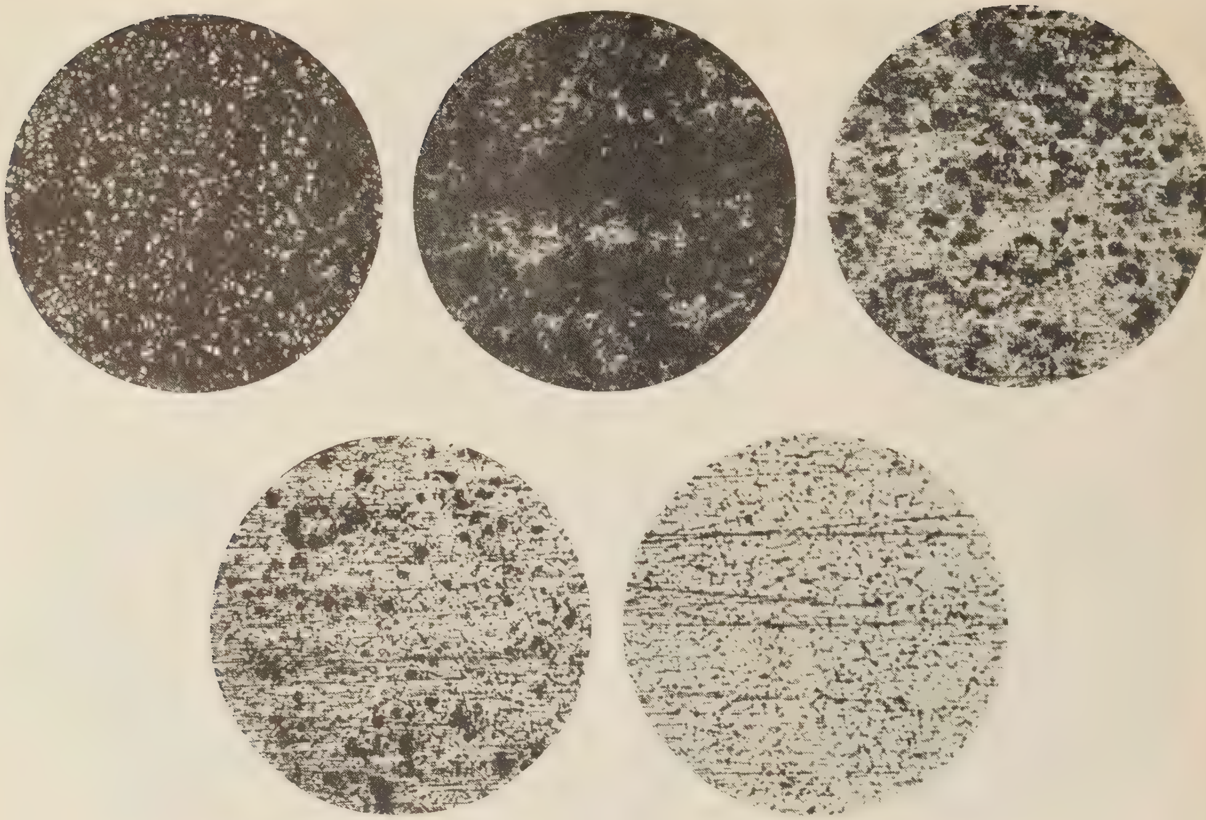


Fig. 5 Effect of Progressive Removal of Metal on the Porosity of Chromium; ×50
(No. 0 emery paper used.)

disassembly the cylinders are ready for finishing. The porous chromium is readily honed with proper abrasives, but here finishing differs from the usual procedure of honing to a bore dimension only. Surface finish or porosity must also come within limits, because porosity has an influence on oil wettability. Fig. 5 shows the influence of surface stock removal on porosity. The factors affecting the finished cylinder bore are as follows:

1 Initial bore dimensions: the bore should be straight and round, as the ideal plating can do no better than deposit chromium to a uniform thickness throughout the cylinder.

2 Plating control of solution, composition, and temperature, current density, and time to obtain exactly the desired thickness of chromium.

3 Removal of correct amount of stock in finishing to obtain the desired surface finish.

Because of these additional requirements final dimensional tolerances on cylinder bores must be wider than those usually held where the only limiting factor is diameter.

For some time the profilometer was used as an inspection instrument in checking porous chromium for surface finish. However, the profilometer fails to give a true indication of surface condition in this case. Fig. 6 shows clearly that the standard profilometer needle does not trace the surface of porous chromium. Engine tests have subsequently established that surface porosity is an important characteristic and accordingly this property is considered in final inspection. While there is a reasonable correlation between the μ in. reading rms and percentage porosity, Fig. 7, it has been found more accurate and expedient to check porosity directly by microscopic inspection. Photomicrographs of various degrees of surface finishing have been planimetered to determine the percentage of flat and below-surface areas. Inspectors using a 50-power right-angle microscope can then search the bore and by comparison with the calibrated photomicrographs, make a reasonably accurate estimate of porosity.

In order to speed up inspection and minimize human errors of judgment we are now adapting the photoelectric cell to this work. The pores, depressions, or pits in the surface will reflect very little light with vertical illumination, while the smooth flats will have high reflectivity. Thus with a constant light source a photoelectric cell can be made to measure reflected light quantity. This, with a properly calibrated meter, can be a measure of the percentage of flat area or, inversely, of porosity. Such a "porosity meter" is shown diagrammatically in Fig. 8. Experiments to date indicate that this may be a satisfactory shop-inspection instrument, but more actual experience with it will be necessary to establish its utility definitely.

A wealth of experience under service conditions is being accumulated with porous-chrome cylinders on all types of engines, but for military reasons no details can now be given. However, other test data of interest are available, particularly accelerated wear tests on $^3/_4$-hp Lauson engines. This is a $1^3/_4$-in. bore × $1^7/_8$-in. stroke air-cooled 4-cycle gasoline engine. Two engines were run for 9 hr under load, one with the standard cast-iron cylinder and one with a porous-chrome cylinder. Allis-Chalmers type B dust (86 per cent below 5 microns, 100 per cent below 15 microns) was added to the initial charge of crankcase oil, the proportion of dust being 8.6 per cent by weight to accelerate wear. Tables 3 and 4, and Fig. 9, show the cylinder, piston, and ring wear.

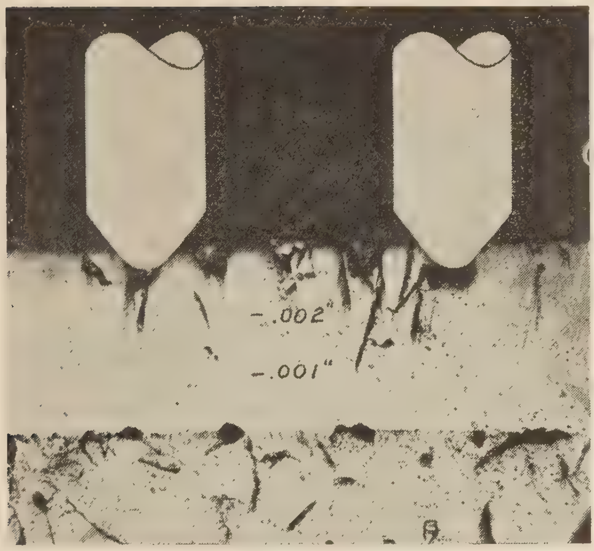


FIG. 6 SHOWING RELATION OF STANDARD 0.0005-IN-RADIUS PROFILOMETER NEEDLE TO PORES
(Honed porous chromium; ×500.)

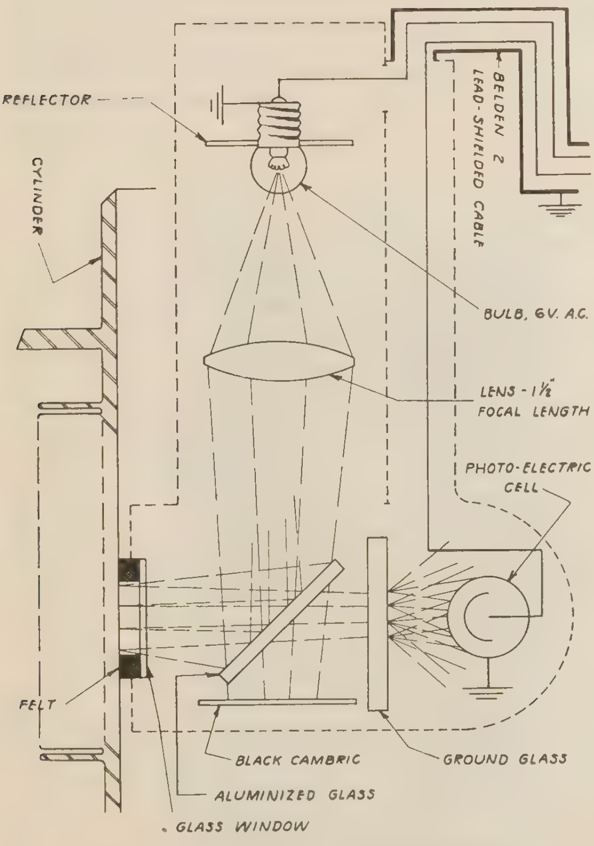


FIG. 8 SCHEMATIC DETAILS OF "POROSITY METER"

The relative wear of the rings is illustrated graphically in Fig. 10 which shows the sections of the worn rings compared with new rings. This test under such severe abrasive conditions, while not quantitatively applicable to full-scale engines,

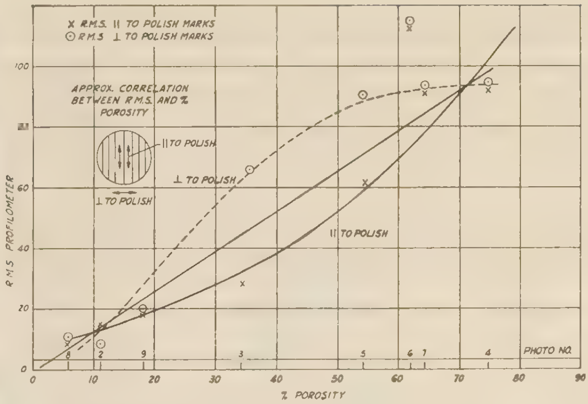


FIG. 7 APPROXIMATE CORRELATION BETWEEN RMS AND PERCENTAGE OF POROSITY

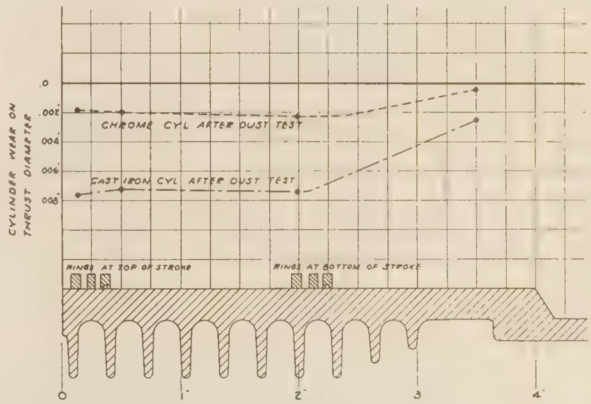


FIG. 9 CYLINDER WEAR ON CHROME AND CAST-IRON CYLINDERS

does indicate greatly lower cylinder wear with porous chromium and at the same time a great reduction in ring wear. Both results have been confirmed in service engines where records have been established of $^1/_{24}$ normal cylinder wear and $^1/_5$ normal

TABLE 3 CYLINDER WEAR

Location	Diametral wear, in. Cast-iron cylinder	Porous-chrome cylinder	Wear ratio, cast-iron cyl., porous-chrome cyl.
$^1/_8$ in. below head end....	0.0077	0.0018	4.3–1
$^1/_2$ in. below head end....	0.0071	0.0021	3.4–1
2 in. below head end.....	0.0083	0.0022	3.8–1

TABLE 4 PISTON AND RING WEAR

	In cast-iron cylinder		In porous chrome cylinder		
	Grams	Per cent of original weight	Grams	Per cent of original weight	Wear ratio, cast-iron cyl., porous-chrome cyl.
Aluminum piston loss in weight......	0.589	0.90	0.316	0.45	1.9:1
Top ring loss in weight......	2.506	56.	0.849	18.9	3.0:1
Second ring loss in weight....	2.586	57.8	0.972	21.8	2.6:1
Oil ring loss in weight......	1.260	46.5	0.865	31.9	1.5:1

ring wear. Of course engines differ, and the improvement resulting from the use of porous chromium will vary with the particular engine.

Another manner in which porous chromium in cylinders immediately shows up to advantage is in the initial test or green

run of an engine. Because of its relative immunity to scuffing and ability to take abuse, ring scuffing and piston seizure are

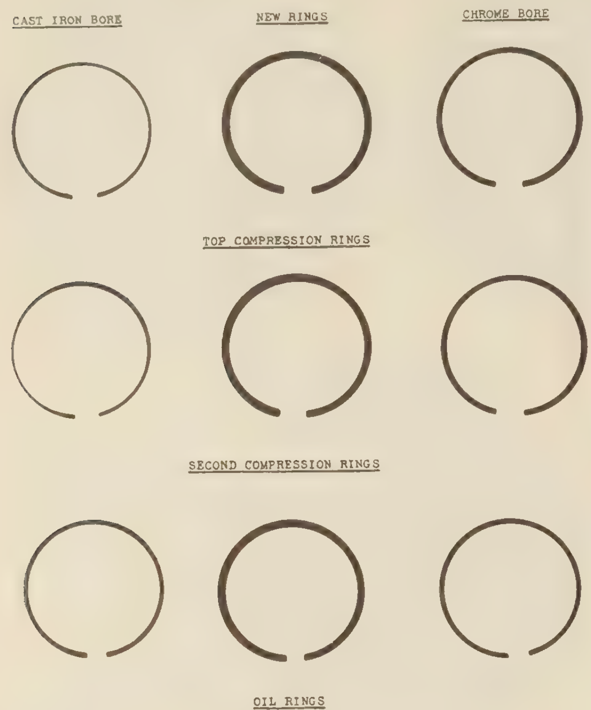


Fig. 10 Comparison of Worn Rings With New Rings

eliminated, and engine production is accelerated. Fig. 11 illustrates two identical pistons from a 10-in. × 12-in. two-cylinder gas test engine, one of the cylinders being cast iron and the other cylinder porous chromium. Both cylinders were carrying the same load and exhaust temperatures, yet the piston in the cast-iron cylinder seized, ruining both piston and cylinder. The piston in the porous-chrome cylinder was only polished where it had rubbed hard and the cylinder was unharmed.

Summary

The inherent properties of porous chromium which contribute to its low wear rate in engine cylinders are as follows:

Hardness, which exceeds that of any other available material for cylinder liners; hardness throughout the case and at high temperatures.

High thermal conductivity, which aids in maintaining low internal cylinder-surface temperatures, improving lubrication and heat transfer.

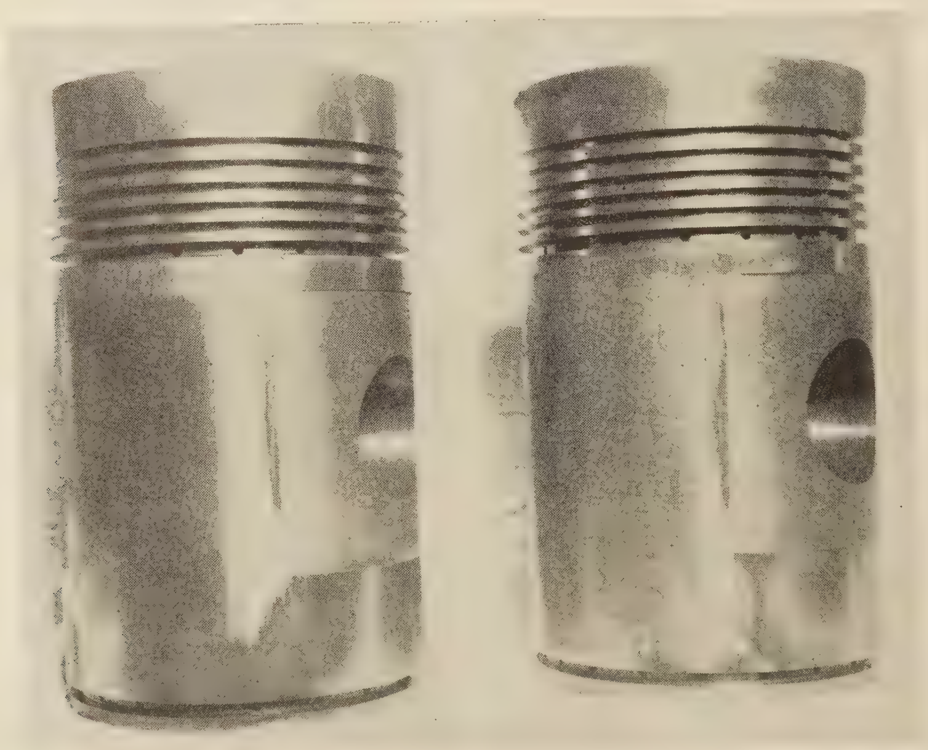
Low friction and low affinity for other metals, minimizing heat generated at the surface under boundary-lubrication conditions.

Corrosion resistance, which eliminates surface deterioration from action of the products of combustion.

In the light of recent theories concerning boundary lubrication an attempt has been made to explain how and why the peculiar surface geometry of porous chromium fits into the theories resulting in high load capacity and wettability.

The processing, finishing, and inspection procedures have been briefly described. The most important factors here are control of solutions, current density, temperature, and time, as well as a final surface finish.

Much has yet to be learned, but the need for intensive re-



(a) (b)

Fig. 11 Results of Seizure Tests on Pistons in Cast-Iron Cylinder (a), and in Porous-Chrome Cylinder (b)

search on the potential possibilities of porous chromium is recognized and such a program is under way.

ACKNOWLEDGMENTS

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Discussion

K. W. Schwartz.[3] Our laboratory measurements on the hardness of chromium indicate that all bright chromium deposits from a chromic-acid bath have about the same hardness, and that holds regardless of the bath composition, temperature, current density, or other plating conditions. However, if the deposit obtained is dull in appearance, the hardness is considerably lower than that of a bright deposit. Our figures indicate a hardness of about 1000 Bhn for bright chromium and a hardness of about 650 Bhn for dull chromium. So-called "milky" chromium has a hardness intermediate between these values, that is, about 800.

As to methods of determining hardness, we have found that practically all of the methods referred to in the literature involve crushing of the chromium deposit to get a reading. This may not be entirely reliable, particularly when the indentation that is made might be influenced by the hardness of the underlying base.

We experimented some years ago with methods of determining so-called hardness of chromium and found that we could get quite consistent results by means of a scratch hardness test. The apparatus which we used was a Bierbaum microcharacter, and the principle on which the apparatus functioned was quite simple. It merely uses a diamond point of definite shape, to which is applied a given weight. The diamond point is drawn across the surface of the chromium to be tested at a regulated speed. The width of the scratch is then measured under the microscope at a magnification of 1000. The results obtained are quite consistent and reproducible. We found that by means of this method the hardness measurements made on chromium deposits, even as thin as 0.001 in., were entirely independent of the base metal on which the chromium was plated.

In connection with comments on the plating operations, it should be stated that pure chromic acid alone, when dissolved in water, will not give a plating bath. The current can be passed through such a solution indefinitely, but no metallic chromium will be plated out. It is quite interesting, and rather unique as an electrochemical phenomenon, that by adding a catalyst to the chromic-acid solution a plating bath results. The catalysts that can achieve this result, interestingly enough, are acid radicals and, as we electrochemists refer to them, they are anions. They have a negative charge and are attracted to the anode, yet they perform their function at the cathode where the metal is deposited.

It is essential that any chromic-acid solution have present in it a certain amount of acid-radical catalyst to enable it to plate; commercially that acid radical is usually sulphate. Whether we add it as sulphuric acid, sodium sulphate, or any other soluble sulphate makes no difference. The point is that the sulphate radical or an equivalent radical must be present in the solution to enable it to plate.

The author states that chromium fills in the smallest irregularities of the surface being plated. From experience, we have found that chromium deposits tend to follow the irregularities in the underlying surface and to reproduce them. For example, let us take the surface of a cylinder liner that had been honed. The reader will be familiar with the honing marks or scratches that are present on the surface. A thickness of chromium as high as 0.008 in. on the radius can be deposited, for example, and the honing marks will tend to be reproduced in the deposit. Furthermore, and this is quite an important precaution, pits in the base metal do not tend to be bridged over by the chromium deposits. Such other things as nonmetallic inclusions in the base metal will not be covered by the chromium deposit, because quite obviously they are not conductors of current, and as such do not take the deposit. Therefore it is quite important that the surface on which we plate the chromium be as free from nonmetallic inclusions as possible.

Incidentally, any uncombined carbon or graphite in cast iron should be kept at the lowest possible minimum, because the con-

[3] United Chromium, Incorporated, New York, N. Y.

ductivity of carbon is very much lower than that of iron, and that will tend to produce undesirable irregularities in the plate.

In connection with chromium deposits on the bore of cylinders, in addition to the degree of porosity, that is to say, the relationship between the so-called table tops or plateaus and depressions, the depth of these depressions is quite important. While we know that electrodeposited chromium is extremely hard and that it functions very well on the bores of liners, it nevertheless does wear after long service. In order to get the maximum serviceable life out of the chromium deposit, these depressions or pores should be as deep as it is practicable to make them. In plating so-called "porous" or oil-retaining deposits of chromium, careful control of the plating conditions and final finishing operations is necessary. This should be performed so that the plated liner when ready to install in the engine has a maximum depth of porosity in addition to the other properties required for satisfactory results.

Hans Bohuslav.[4] We have made a few applications of chrome-plated steel liners in high-speed gasoline engines, and one of the drawbacks has been the length of time of break-in of such liners. In our first tests, it took 25 to 30 hr break-in time which, on engines of this type, is too long for efficient production. Even after that, our oil consumption was erratic. So we tried different methods such as match-lapping and individual honing, and found that apparently it was not practical at least on steel liners. Recently we have had some hope of overcoming the difficulty by using taper-face rings. We have found that such rings allow for more irregularities in the liner, at least in plain-steel liners. To try this, we used some of the liners which had been tested some months previously and which had been somewhat misused. Despite the fact that these liners were quite irregular, having been put in a new block which did not allow the same position for the surfaces, the oil consumption came down very rapidly. In about 2 hr running, we were within the normal oil consumption; in another 2 hr, we were considerably below our average.

The author's comments on the following questions would be appreciated: With all the cracks and porosity in the plating, it is the writer's understanding that considerable honing is done after the plating. What happens to the fine particles left during and after the honing operation?

The second question relates to the control of the areas of the metal so they will not fall out. In other words, if two cracks are too close together, or are not exactly perpendicular to the surface, there would normally be some areas in which the cracks finally merged on the steel or on the base surface.

[4] Sterling Engine Company, Buffalo, N. Y.

H. Van der Horst.[5] It has been shown by tests on porous-chromium cylinders, conducted at the Naval Experiment Station at Annapolis, that the use of chromium plating in a well-known high-speed high-output Diesel engine which had been run for 6000 hr resulted in a reduction in wear of 1 to 14. That is a large figure, and what is more surprising, a reduction of ring wear to 22 per cent was made, which means only 22 per cent of what is known to occur when running-in cast-iron cylinders.

Author's Closure

The catalytic effect of the sulphate radical in chromium deposition is known to all concerned in plating, but the author did not wish to burden mechanical engineers with the electrochemical phase of the work. It is true, in general, that chromium deposition tends to follow the base-metal surface. However, the author has observed photomicrographs of edge sections where the base metal has been selectively etched, and the chromium has deposited into the pits and recesses.

Concerning Mr. Bohuslav's question about excessive oil consumption, the only comment is that every engine is a separate personality and often requires some special treatment of its own. As he says, very fortunately now, the taper-faced rings have done the job. The abrasives in the finishing or honing operation, of course, are washed down to some extent by the copious flow of coolant in the honing. It is easily conceivable that some abrasive may lodge in the pores and crevices. However, in the normal procedure, these are thoroughly washed before they ever go into service, and it is doubtful if there has ever been any failure which could be attributed to retaining the abrasive from the honing operation itself.

Regarding the cracks and their proximity, it may be explained by the fact that edge sections through the porous chromium show the table top and the sides coming down from the table top at an angle. The table top has a good firm base. It is not as though it were undercut and standing up like a column. It is rather a truncated cone, if we want to think of it that way, so that the structure is pretty well cantilevered. We have had no breakage or failure of the chromium due to the geometry of the metal structure.

[5] Van der Horst Corp. of America, Olean, N. Y.

High-Pressure Pipe-Line Research

By F. W. LAVERTY and F. M. MCNALL,[1] OLEAN, N. Y.

In estimating gas-pipe-line costs prior to undertaking the detailed design, it has been the practice generally to utilize graphical determinations of such variables as pipe-line diameter, compressor-station spacing, operating pressures, and the like. The graphical method is a time-consuming process, however, entailing the preparation and consideration of a large number of curves. The purpose of this paper is to show the derivation of an empirical equation containing all of the design variables, to which the methods of the differential calculus may be applied to yield a general series of expressions for the most economical design of any pipe line.

Nomenclature

The following nomenclature is used in the paper:

a = compressor brake horsepower per million cubic feet per day of gas compressed, 18.5 for curve data
A = cross-sectional area of pipe, sq ft
avg = average condition
B = fixed charge on pipe, fraction of investment (taken as 0.12 for curve data)
$C = \left(1 - \frac{1}{C_r^2}\right)$
C_I = total investment, dollars per mile
Cy = total yearly cost, dollars per year per mile
Cf = fuel cost, dollars per 1000 standard cu ft (taken as 0.10 for curve data)
Cr = compression ratio
d = inside diameter of pipe, ft
D = inside diameter of pipe, in.
E = experience or efficiency factor for flow equation
f = friction factor, defined by Fanning's equation
F = friction, feet of gas; load factor expressed as a decimal
g = acceleration due to gravity, fpsps (taken as 32.2 for illustrative data)
G = constant in pipe-laying cost, dollars per ton per mile, 12 in data for curves
H = constant in pipe-laying cost, dollars per mile, 3000 in data for curves
K = constant = $(\alpha X + 86.4Cf + 6.79)$ = 19.43 for curves
K_1 = constant = $(B)\,(Y + G)\,(28.2)$ = 328.3 for curves
K_2 = constant = $\frac{80.8}{\rho_0\mu^{0.0814}}\left[\frac{M}{T}\right]^{0.541}$ = 765.3 for curves
l = distance, interval of pipe between compressor stations, ft
L = distance interval of pipe between stations, miles
m, n = coefficients of optimum diameter equation
M = molecular weight, lb (16 for methane)
N = constant in pipe-laying cost, dollars per in. of diam per mile (270 for illustrative data)
P = absolute pressure, psi
p = absolute pressure, psf
P_0 = base pressure, absolute
P_1 = compressor discharge pressure, psia
P_2 = compression suction pressure, psia
q = quantity of gas transported, cfs, measured at P_0 and T_0
Q = quantity of gas transported, cu ft per day, measured at P_0 and T_0
R = gas constant = 1546 for illustrative data
S = allowable hoop stress, psi (20,000 psi for illustrative data)
T = flowing temperature deg R (520-deg R for illustrative data)
T_0 = base temperature, deg R
t = thickness of pipe wall, in.
u = linear velocity, fps
V = specific volume, cu ft per lb
W = weight of pipe, tons per mile
X = cost of compressors, dollars per bhp, installed (80 for illustrative data)
Y = cost of pipe, dollars per ton (85 for illustrative data)
Z = constant in gas equation, "supercompressibility," dimensionless
α = fixed charge on compressors, fraction (0.05 for illustrative data)
ρ_0 = density of gas, lb per cu ft at T_0 and P_0 (0.0422 for illustrative data)
ρ = density of gas, lb per cu ft at T and P
μ = absolute viscosity of gas, lb per ft-sec (7×10^{-6} for methane)
$\bar{d}$ = symbol for differential, e.g. ($\bar{d}p$ differential of pressure)

General Considerations for Pipe-Line Cost Estimates

The detailed design of a gas pipe line is so laborious and time-consuming that it is generally desirable first to estimate rather closely the optimum values of the design variables for the case at hand. In general, these include pipe-line diameter, compressor-station spacing and operating pressures as functions of material cost, labor cost, and the character and quantity of gas to be handled.

Previous methods for estimations of this type (1, 3, 4, 6, 7)[2] have for the most part been graphical in nature, entailing the preparation and consideration of a large number of curves. In addition, the effects of deviations from the perfect-gas laws have been in some cases neglected.

If the total cost of a gas pipe line can be represented by an empirical equation containing all of the design variables, the methods of the differential calculus may be applied to yield a general series of expressions for the most economical design of any pipe line.

The first requirement for the development of a generalized cost equation is a gas-flow equation which will be dependable over a wide pressure and composition range. An equation which corresponds fairly well with known pipe-line conditions may be developed from Bernoulli's theorem combined with the Fanning equation. Based on the assumption that the flow is isothermal and at constant elevation, the differential form of Bernoulli's theorem

$$-V\bar{d}p = \frac{\mu}{g}\,du + \bar{d}F \quad [1]$$

[1] Clark Brothers Company.

Contributed by the Petroleum Division and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

may be combined with the differential form of the Fanning equation

$$\bar{d}F = 2f\left(\frac{\mu^2}{g}\right)\frac{dl}{d} \qquad [2]$$

to yield the expression

$$-V\bar{d}p = \frac{\mu}{g}\bar{d}u + 2f\left(\frac{\mu^2}{g}\right)\frac{dl}{d} \qquad [3]$$

Now

$$u = \frac{q\rho_0 V}{A} \qquad [4]$$

and

$$V = \frac{ZRT}{pM} \qquad [5]$$

hence the differential form of the flow equation becomes

$$\frac{-M}{ZRT}\,p\bar{d}p = \frac{q^2\rho^2}{gA^2}\left(2f\frac{\bar{d}l}{d} - \frac{\bar{d}p}{p}\right) \qquad [6]$$

Integrating between limits leads to

$$q = \frac{A}{\rho_0}\left[\frac{\dfrac{gM}{2Z_{avg}RT}(P_1^2 - P_2^2)}{\dfrac{2fl}{d} + \ln p_1/p_2}\right]^{1/2} \qquad [7]$$

and neglecting ($\ln p_1/p_2$) in comparison to $\frac{2fl}{d}$, it simplifies to

$$q = \frac{A}{\rho_0}\left[\frac{gMd(p_1^2 - p_2^2)}{4flZ_{avg}RT}\right]^{1/2} \qquad [8]$$

For the Reynolds numbers between 5×10^4 and 5×10^7, which covers practically all gas-transmission practice, an average straight line can be drawn through the logarithmic plot of the Drew and Genereaux data (2)

$$f = 0.03015\left(\frac{du\rho}{\mu}\right)^{-0.1505} \qquad [9]$$

Substituting the value of f in Equation [9] in Equation [8], and replacing A by $\pi d^2/4$

$$q = \frac{2.467}{\rho_0}\left[\frac{d^{4.85}gM(p_1^2 - p_2^2)}{lZ_{avg}RT\mu^{0.1505}}\right]^{0.541} \qquad [10]$$

Using the common engineering units and introducing the necessary conversion constants, Equation [10] takes the form

$$Q = \frac{(2.467)(144)^{1.081}(24)(3600)(32.2)^{0.541}}{(12)^{2.622}(5280)^{0.541}(1546)^{0.541}}\,\frac{1}{\rho_0}\left[\frac{D^{4.85}M(P_1^2 - P_2^2)}{LZ_{avg}T\mu^{0.1505}}\right]^{0.541} \qquad [11]$$

$$Q = \frac{80.8}{\rho_0\,\mu^{0.0814}}\left[\frac{D^{4.85}M(P_1^2 - P_2^2)}{LZ_{avg}T}\right]^{0.541}E \qquad [12]$$

where E is an experience factor ordinarily having a value of from 1 to 1.03 for natural gas.

For simplicity, let

$$K_2 = \frac{80.8}{\rho_0\mu^{0.0814}}\left[\frac{M}{T}\right]^{0.541}E \qquad [13]$$

a constant for any particular case, then

$$Q = K_2\left[\frac{D^{4.85}(P_1^2 - P_2^2)}{LZ_{avg}}\right]^{0.541} \qquad [14]$$

Cost on a Minimum Yearly Basis

One of the possible cost bases for pipe-line computation is minimum yearly cost, or that which gives the most economical gas transportation. The total yearly cost may be divided into fixed charges and operating expense.

Compressor Cost. The fixed charges on gas compressors may be formulated as follows: The initial installed cost of compressors is X dollars per horsepower. To pump Q cubic feet of gas per day for a distance of L miles requires aZ_2 brake horsepower per million cubic feet per day, where Z_2 is the deviation of the gas from the perfect-gas laws at the suction pressure, and a is the reference brake horsepower per million cubic feet of gas per day, dependent only upon the nature of the gas and the ratio of compression. The investment in compressors is then

$$X\left(\frac{QaZ_2}{L10^6}\right) \text{ dollars per mile}$$

The fixed charge on the compressors α includes interest on the capital investment, depreciation, taxes, insurance, etc. The yearly fixed charge on the compressors is then

$$\left(\frac{\alpha XQ_aZ_2}{L10^6}\right) \text{ dollars per year-mile}$$

Pipe Cost. The fixed charges on the pipe may be found from the pipe cost, Y dollars per ton, and the necessary W tons of pipe per mile. For steel pipe, $W = 28.2\,(D + t)t$, where D is the inside diameter of the pipe in inches, and t is the pipe-wall thickness in inches. The Barlow expression for wall thickness of pipe in terms of maximum pressure P, pounds per square inch, and maximum allowable stress, S pounds per square inch, is $t = \frac{P_1D}{2(S - P_1)}$.

The direct pipe investment is then

$$YW = Y28.2\left[D + \frac{P_1D}{2(S - P_1)}\right]\left[\frac{P_1D}{2(S - P_1)}\right] \qquad [15]$$

dollars per mile.

According to the A. O. Smith report (4), a good division of pipe-laying cost is H dollars per mile for right of way, surveying, clearing, supervision engineering, undistributed costs, and a part of ditching, painting, and cleaning expenses; G dollars per ton-mile for unloading, hauling, aligning, placing, and welding or coupling; and N dollars per mile per inch of diameter for ditching, painting, backfilling, and laying equipment.

The total pipe cost is then ($YW + H + GW + ND$) dollars per mile. At $B \times 100$ per cent fixed charges on the installed pipe (depreciation, taxes, insurance, interest on capital, salaries of pipe-line personnel, and all operating expenses chargeable to line maintenance), the fixed yearly charges on the installed pipe are

$$B(Y + G)(28.2)\left[D + \frac{P_1D}{2(S - P_1)}\right]\left[\frac{P_1D}{2(S - P_1)}\right] + BND + BH \qquad [16]$$

Operating Expenses

The pipe-line operating expenses may be expressed as functions of the amount of gas transported, if average values of the individual items are considered. Labor, supervision, salaries, etc. amount to about \$1750 per month for a 4000-hp compressor

station. Assuming direct proportionality for other stations of approximately the same size, this expense is

$$\frac{1750 \times 12}{4000}\left(\frac{QaZ^2}{L10^6}\right) = 5.25\left(\frac{QaZ_2}{L10^6}\right) \text{ dollars per year per mile.}$$

The average value of maintenance is \$0.50 per hp per year; lift water approximately \$0.20 per hp per year; and oil about \$0.84 per hp per year. The yearly cost of these items is then \$1.54 per hp per year per mile. About 10 standard cu ft of 1000 Btu fuel gas per bhp-hr are required by the gas compressors at a cost of Cf dollars per 1000 standard cu ft. The yearly fuel cost is then

$$\frac{10 \times 24 \times 360}{1000} Cf \frac{QaZ_2}{L10^6}$$

or

$$86.4\, Cf \frac{QaZ_2}{L10^6} \text{ dollars per year per mile.}$$

An allowance for gas lost from the pipe line may also be made, and for present purposes it will be taken as \$100 per year per mile. Taken in this manner, the loss figure has no effect on the design variables.

Upon adding all of the fixed and operating costs, the total yearly cost per mile

$$Cy = (\alpha X + 86.4Cf + 6.79)\frac{QaZ_2}{L10^6} + B(Y+G)(28.2)D^2 \left[\frac{P_1}{2(S-P_1)} + \frac{P_1^2}{4(S-P_1)^2}\right] + BND + BH + 100 \quad [17]$$

For convenience in manipulating any specific problem, let

$$K = (\alpha X + 86.4Cf + 6.79) \quad [18]$$

and

$$K_1 = 28.2B(Y+G) \quad [19]$$

Then

$$Cy = \frac{KQaZ_2}{L10^6} + K_1D^2\left[\frac{P_1}{2(S-P_1)} + \frac{P_1^2}{4(S-P_1)^2}\right] + BND + BH + 100 \quad [20]$$

A set of values of compressor suction pressure P_2, compressor discharge pressure P_1, pipe diameter D, and compressor-station spacing L, which makes the total yearly cost a minimum, must satisfy the requirement that the partial derivatives of the total yearly cost with respect to each of these variables must simultaneously equal zero. Proof that a set of values so found corresponds to minimum-cost conditions may be found by substitution of the set in the cost equation, comparing the total yearly cost with that resulting from any other set.

The term L may be eliminated from the yearly cost equation on replacing it by its value determined from the flow Equation [14]

$$\left(\frac{K_2}{Q}\right)^{1.85}\frac{D^{4.85}(P_1^2 - P_2^2)}{Z_{avg}} = L \quad [21]$$

The yearly cost equation then becomes

$$Cy = \frac{KQ^{2.85}Z_2aZ_{avg}}{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)} + K_1D^2\left[\frac{P_1}{2(S-P_1)} + \frac{P_1^2}{4(S-P_1)^2}\right] + BND + BH + 100 \quad [22]$$

The partial derivatives of Cy with respect to D, P_2, and P_1 are

$$\frac{\partial Cy}{\partial D} = \frac{-KQ^{2.85}Z_2aZ_{avg}\,4.85}{10^6K_2^{1.85}D^{5.85}(P_1^2-P_2^2)} + 2K_1D\left[\frac{P_1}{2(S-P_1)} + \frac{P_1^2}{2(S-P_1)^2}\right] + BN \quad [23]$$

$$\frac{\partial Cy}{\partial P_2} = \frac{2KQ^{2.85}Z_2aZ_{avg}P_2}{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)^2} + \frac{KQ^{2.85}Z_2a}{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)}\frac{\partial Z_{avg}}{\partial P_2} + \frac{KQ^{2.85}Z_2Z_{avg}}{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)}\cdot\frac{\partial a}{\partial P_2} + \frac{KQ^{2.85}aZ_{avg}}{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)}\cdot\frac{\partial Z_2}{\partial P_2} \quad [24]$$

and

$$\frac{\partial Cy}{\partial P_1} = \frac{-2KQ^{2.85}Z_2aZ_{avg}P_1}{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)^2} + \frac{KQ^{2.85}Z_2Z_{avg}}{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)}\cdot\frac{\partial a}{\partial P_1} + \frac{KQ^{2.85}Z_2a}{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)}\frac{\partial Z_{avg}}{\partial P_1} + \frac{K_1D^2S^2}{2(S-P_1)^3} \quad [25]$$

Setting $\frac{\partial Cy}{\partial D}$ equal to zero, and multiplying all terms of the resulting equation by $\frac{D^{5.85}}{K_1\left[\frac{P_1}{S-P_1} + \frac{P_1^2}{2(S-P_1)^2}\right]}$ the first expression for minimum value of Cy is

$$D^{6.85} + \frac{BN}{K_1\left[\frac{P_1}{S-P_1} + \frac{P_1^2}{2(S-P_1)^2}\right]}D^{5.85} = \frac{4.85KQ^{2.85}Z_2aZ_{avg}}{10^6K_2^{1.85}(P_1^2-P_2^2)K_1\left[\frac{P_1}{S-P_1} + \frac{P_1^2}{2(S-P_1)^2}\right]} \quad [26]$$

Equating $\frac{\partial Cy}{\partial P_2}$ to zero, and multiplying all terms of the resulting equation by $\frac{10^6K_2^{1.85}D^{4.85}(P_1^2-P_2^2)}{KQ^{2.85}Z_2aZ_{avg}}$, the second expression for the minimum value of Cy is

$$\frac{2P_2}{P_1^2-P_2^2} + \frac{\partial \ln Z_{avg}}{\partial P_2} + \frac{\partial \ln a}{\partial P_2} + \frac{\partial \ln Z_2}{\partial P_2} = 0 \quad [27]$$

Using the empirical relationship for methane gas

$$a = 56.8 \ln P_1/P_2 + 5.1 \quad [28]$$

and finding that $\frac{\partial \ln Z_{avg}}{\partial P_2}$ and $\frac{\partial \ln Z_2}{\partial P_2}$ are very small numbers from $P_2 = 0$ to $P_2 = 2000$ psi, Equation [24] reduces to

$$\ln [P_1/P_2] = \frac{1}{2}(P_1^2/P_2^2) - 0.5898, \text{ or } P_1/P_2 = 1.33 = Cr \quad [29]$$

Following the same method of approach, Equation [25] reduces to

$$\frac{-2}{P_1C} + \frac{56.8}{P_1a} + \frac{10^6K_2^{1.85}D^{6.85}S^2P_1^2CK_1}{2(S-P_1)^3KQ^{2.85}Z_2aZ_{avg}} = 0 \quad [30]$$

from which

$$P_1 = \frac{S}{1 + D^{2.283}\left(\dfrac{K_2^{1.85}S^2K_1}{KQ^{2.85}Z_2Z_{\text{avg}}}\right)^{1/3}} \quad (17.40) \qquad \ldots\ldots [31]$$

Equations [26] and [31] may be solved simultaneously for P_1 and D by plotting P_1 versus D for both equations. The intersection of the two curves in the positive quadrant then yields the desired values of P_1 and D. As a convenience in evaluating D in Equation [20] for trial values of P_1 it may be considered to be of the form

$$D^{6.85} + mD^{5.85} = n$$

or preferably

$$D^{5.85} = \frac{n}{m + D}$$

Two or three trial values of D should then be sufficient to obtain the proper value of D.

The station spacing L may then be found from Equation [21], and the minimum total yearly cost then computed from the cost Equation [17].

It is obvious that there is but one such set of positive values of P_1, P_2, and D. The total cost Cy is either a maximum or minimum, and that this set corresponds to a minimum value of Cy may be seen from the total yearly cost Equation [20]. The value of Cy can be made larger than any previously assigned value by increasing suitably the numerical values for P_1 and D. Obviously, the maximum value of Cy is infinite, and the values

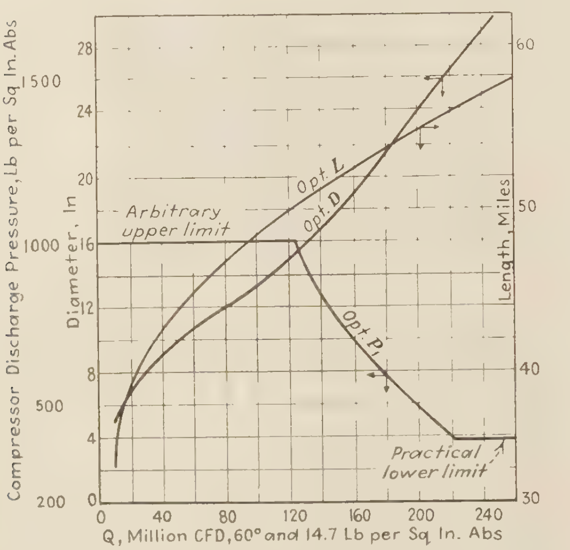

FIG. 1 OPTIMUM VALUES OF P, L, AND D FOR VALUES OF Q, BASED ON OPTIMUM COMPRESSION RATIO, Cr = 1.33 AND MINIMUM YEARLY COST

found from the partial derivatives must represent a minimum. The solution of Equation [26] for a range of values of P_1 and subsequent determination of Cy by means of Equation [22] as a function of P_1 is in many cases a more rapid and convenient method for determining the optimum operating conditions.

Using hypothetical values for material and labor costs which are approximately averaged over the country, but which do not represent any particular case, a set of optimum design conditions for the transportation of several different amounts of methane have been computed and plotted in Fig. 1. It may be seen that for quantities of methane up to approximately 100,000,000 cu ft per day, the optimum compressor-discharge pressure is over 1000 psi which was taken as an arbitrary limit. For greater gas quantities, the optimum compressor-discharge pressure decreases until, at approximately 220,000,000 cu ft per day, it is about 400 psi which was taken as an arbitrary lower limit. Both the optimum pipe diameter and compressor-station spacing increase with increasing gas quantities.

For methane at ordinary temperatures, the optimum compression ratio is not affected appreciably by deviation from the ideal gas laws, but for hydrocarbon gases which show greater deviations, the optimum compression ratio increases, the optimum operating pressures increase, and the optimum pipe diameter decreases.

Costs on a Minimum Initial Basis

If minimum initial investment is taken as the goal for gas-pipeline design, the total investment may be computed from the yearly cost Equation [17] by omitting the yearly charges and the estimated loss figure. Thus

$$C_I = \frac{XQZ_2a}{L10^6} + (Y + G)(28.2)D^2\left[\frac{P_1}{S - P_1} + \frac{P_1^2}{2(S - P_1)^2}\right] + ND + H \ldots\ldots [32]$$

On setting $\dfrac{\partial C_I}{\partial P_2}$ equal to zero, the optimum compression ratio is found to be 1.33, identical with the first case considered when methane or an ideal gas is considered.

On setting $\dfrac{\partial C_I}{\partial D}$ equal to zero, the form of this equation is similar to that found before

$$D^{6.85} + \frac{N}{(Y + G)(28.2)\left[\dfrac{P_1}{S - P_1} + \dfrac{P_1^2}{2(S - P_1)^2}\right]}D^{5.85} = \frac{4.85 \times Q^{2.85}Z_2aZ_{\text{avg}}}{10^6K_2^{1.85}(P_1^2 - P_2^2)(Y + G)\left[\dfrac{P_1}{S - P_1} + \dfrac{P_1^2}{2(S - P_1)^2}\right](28.2)} \ldots\ldots [33]$$

and may be solved in the same manner.

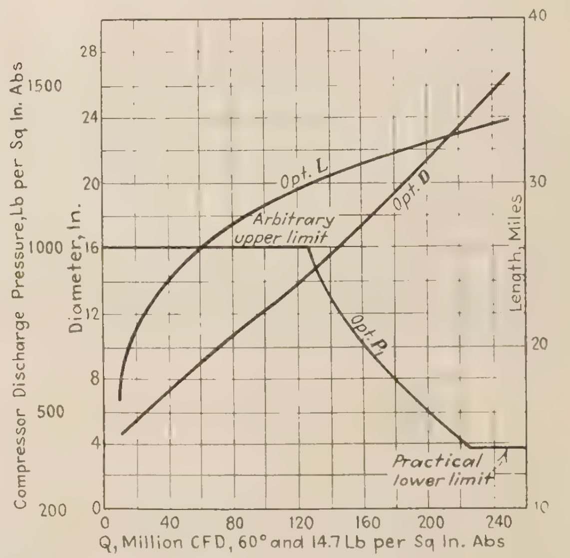

FIG. 2 OPTIMUM VALUES OF P, L, AND D FOR VALUES OF Q, BASED ON OPTIMUM COMPRESSION RATIO, Cr = 1.33, AND MINIMUM INITIAL COST

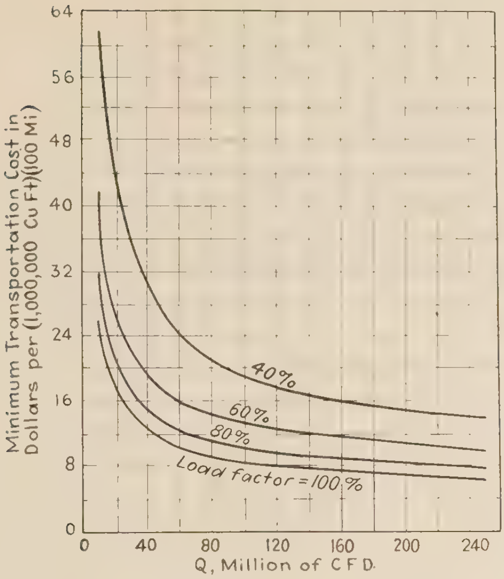

Fig. 3 Minimum Transportation Cost Versus Daily Quantity Transported, Based Upon Optimum Conditions for Minimum Yearly Cost

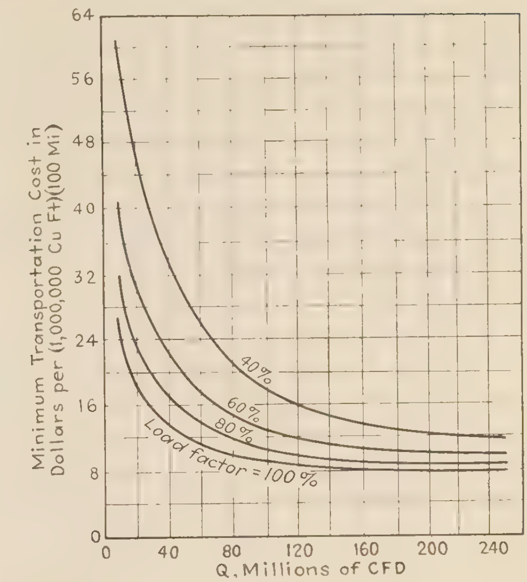

Fig. 4 Minimum Transportation Cost Versus Daily Quantity Transported, Based Upon Optimum Conditions for Minimum Initial Cost

The third defining equation for minimum initial investment, found by setting

$$\partial C_T/\partial P_1 = 0$$

becomes

$$P_1 = \frac{S}{1 + D^{2.283}\left(\frac{K_2^{1.85}S^2(Y+G)(28.2)}{XQ^{2.85}Z_2Z_{\text{avg}}}\right)^{1/3}(17.40)} \quad \ldots . [34]$$

Simultaneously, Equations [33] and [34] may now be solved graphically for any value of Q. In most cases of practical interest, however, it is more practical and instructive to solve Equation [33] for a range of pressures, and to plot C_I from Equation [32] as a function of P_1 for the gas flow under consideration. The results of a series of such solutions are shown in Fig. 2, wherein L is computed from the flow equation as before. It may readily be seen that, although the optimum operating pressures have not changed appreciably, the best pipe diameters and station spacings are smaller than for the case when minimum yearly cost was the goal. Actually, the percentage of total capital invested in compressor stations has increased, and the portion of total capital invested in pipe has decreased compared to the first case. Obviously, the cost of transporting a given quantity of gas per year is greater than that in the first case studied.

The effect of load factor on transportation cost may readily be computed from the yearly cost Equation [17] by segregating the fixed and variable operating expenses, and introducing the term F, load factor, expressed as a decimal. The new total operating cost is equal to F (operating expense) + (fixed charges), and the cost of transporting any given quantity of gas in dollars per cubic foot is given by the equation

$$\text{Cost} = \frac{F\text{ (Operating expense)} + \text{(fixed charges)}}{FQ \times 365} \times \text{miles} \ldots [35]$$

The effect of load factor on the minimum transportation cost is shown in Fig. 3, and on the transportation cost for minimum initial investment in Fig. 4 for the cases already discussed.

General Conclusions

While no specific conclusions may be reached without the substitution of actual cost figures in the equations advanced, some generalizations may be made. The optimum compression ratio is entirely dependent upon the nature of the gas handled. This dependence is threefold, based upon the power necessary to compress a given standard volume of the gas, the manner in which the power changes with compression ratio, and, to a lesser extent, on the change in compressibility of the gas with suction pressure. The best operating pressure for the system is increased as the quantity of gas to be transported decreases, as the allowable pipe stress increases, and as the compressibility of the gas increases. Pipe diameter and station spacing are rather complex functions of all of the design conditions, increasing with an increase in the quantity of gas to be handled.

It must be emphasized that the numerical data represented by the curves in this paper are derived from purely hypothetical cost figures and do not necessarily represent conditions for any actual gas pipe line.

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Limiting Isothermal Flow in Pipes

By R. C. Binder,[1] Lafayette, Ind.

An explicit formulation is given for the limiting isothermal flow of a gas in a horizontal pipe. For isothermal flow, the ratio of critical pressure to initial pressure equals the initial Mach number times the square root of the ratio of specific heats. The critical-pressure ratio for isothermal flow is higher than that for adiabatic flow. Some comparisons are drawn between isothermal, adiabatic, and incompressible flow.

In dealing with the general problem of the flow of gases through pipes, it is common to consider separately two cases, i.e., (1) adiabatic flow and (2) isothermal flow. Adiabatic flow has been detailed in the literature. Complete solutions, for example, can be found in the works of Stodola,[2] and Schüle.[3]

These solutions bring out some important differences between incompressible and compressible adiabatic flow. For one thing, expanding adiabatic flow is characterized by a limiting condition at which the fluid velocity equals the velocity of pressure propagation. The simple relations for incompressible flow do not indicate any limiting conditions. Limiting isothermal flow apparently has not received due and full recognition in the literature. The purpose of the present paper is to give a similar explicit formulation for isothermal flow, to detail limiting conditions, and to draw some comparisons between isothermal, adiabatic, and incompressible flow.

The following discussion will be confined to the flow in horizontal, constant-diameter pipes. It will be necessary to examine some basic relations first before investigating limiting conditions.

Basic Relations

The following nomenclature will be used:

- c = velocity of pressure propagation
- D = internal diameter of pipe
- f = pipe friction coefficient
- g = gravitational acceleration
- k = ratio of specific heat at constant pressure to specific heat at constant volume
- l = pipe length
- M = Mach's number, V/c
- p = absolute pressure at any point in pipe
- v = specific volume
- V = velocity of fluid

Let dl represent the length along a pipe of an infinitesimal element of fluid. The specific volume of the fluid in this element is v. The pressure drop across the element is dp and the velocity change is dV. The general energy equation, in differential form, can be written as

$$vdp + \frac{d}{2g}(V^2) + dh = 0$$

where dh is the lost head. The usual practice will be followed, in expressing the lost head in the form

$$dh = f\frac{V^2}{2gD}dl$$

The general energy equation can then be written as

$$\frac{2gv}{V^2}dp + \frac{2dV}{V} + \frac{fdl}{D} = 0 \qquad [1]$$

For isothermal flow, the Reynolds number is constant along the pipe, and hence the friction factor is also constant. The equation of state gives the relation

$$\frac{dp}{p} = -\frac{dv}{v} \qquad [2]$$

The equation of continuity gives the relation

$$\frac{dV}{V} = \frac{dv}{v} \qquad [3]$$

Limiting Conditions

The velocity of pressure propagation c is given by the relation

$$c = \sqrt{kgRT} = \sqrt{kgpv}$$

where R is the gas constant, and T is the absolute temperature. For isothermal flow c is constant along the pipe.

Let the subscript 1 refer to inlet conditions, and the subscript 0 refer to limiting or critical conditions. For example, p_0 is the limiting pressure. As gas flows along the pipe, the pressure decreases below the inlet pressure p_1, and the velocity increases above V_1. The pressure in the constant-diameter pipe cannot drop below p_0.

Equations [1], [2], and [3] can be combined to give the form

$$\frac{gpv}{V^2}\left(\frac{dp}{p}\right) - \frac{dp}{p} + \frac{fdl}{2D} = 0 \qquad [4]$$

Limiting conditions are reached at the end of the pipe when $fdl = 0$. For these conditions, Equation [4] shows that

$$\frac{gpv}{V_0^2} = 1$$

$$V_0 = \sqrt{gpv} = \sqrt{gRT} = \frac{c}{\sqrt{k}} \qquad [5]$$

The maximum velocity that can be developed in the pipe is V_0; this velocity is less than the acoustic velocity.

Mach's number is a convenient parameter for studying compressible flow. Mach's number M is defined as the dimensionless ratio V/c. The ratio of inertia to elastic force is directly proportional to the square of Mach's number. Let M_1 represent the inlet Mach number.

The critical pressure ratio p_0/p_1 can be determined by using the equation of state, equation of continuity, and Equation [5] to give

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[1] Associate Professor of Mechanical Engineering, School of Mechanical, Aeronautical Engineering, Purdue University. Mem. A.S.M.E.

[2] "Steam and Gas Turbines," by A. Stodola, McGraw-Hill Book Company, Inc., New York, N. Y., 1937, vol. 1, p. 62.

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Contributed by the Main Research Committee and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

$$\frac{p_0}{p_1} = \frac{v_1}{v_0} = \frac{V_1}{V_0} = \frac{V_1\sqrt{k}}{c} = \sqrt{k}\;M_1 \ldots\ldots\ldots [6]$$

For isothermal flow the ratio of critical pressure to initial pressure equals the initial Mach's number times the square root of k.

Various investigators have shown that the critical-pressure ratio for adiabatic flow can be expressed in the form

$$\frac{p_0}{p_1} = M_1^2 \left\{\left(\frac{k-1}{k+1}\right)\left[1 + \frac{2}{(k-1)M_1^2}\right]\right\}^{1/2} \ldots\ldots [7]$$

The solid line in Fig. 1 is for isothermal flow, whereas the dotted line is for adiabatic flow. Fig. 1 and subsequent plots in this paper were drawn for a value of $k = 1.40$; $k = 1.40$ for air, carbon monoxide, hydrogen, and oxygen. Fig. 1 shows that the critical-pressure ratio for isothermal flow is higher than that for adiabatic flow.

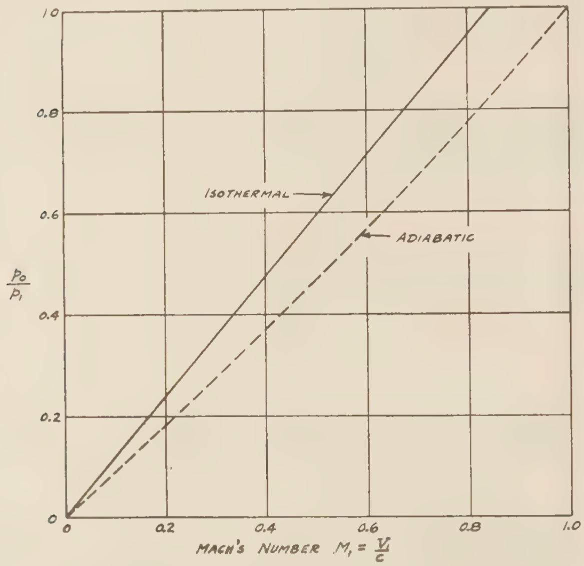

FIG. 1 LIMITING VALUES OF PRESSURE RATIO AS A FUNCTION OF MACH'S NUMBER, FOR GASES WITH $k = 1.40$

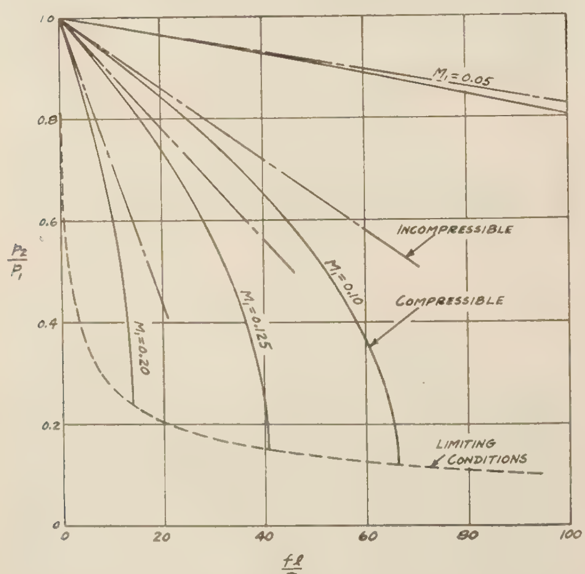

FIG. 2 VARIATION OF PRESSURE RATIO WITH fl/D AND MACH'S NUMBER FOR ISOTHERMAL FLOW, FOR GASES WITH $k = 1.40$

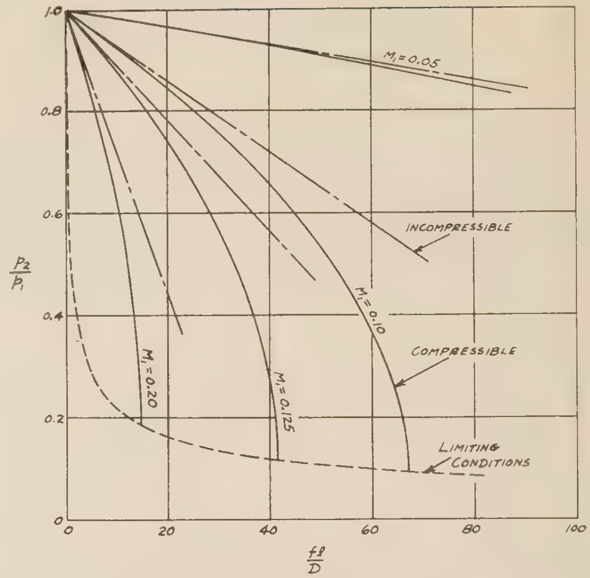

FIG. 3 VARIATION OF PRESSURE RATIO WITH fl/D AND MACH'S NUMBER FOR ADIABATIC FLOW, FOR GASES WITH $k = 1.40$

The continuity equation and the equation of state can be combined to put the first term of Equation [1] into a form convenient for integrating. The final result, after integrating, is

$$p_1^2 - p_2^2 = \frac{V_1^2 p_1}{g v_1}\left[2 \log_e \frac{V_2}{V_1} + \frac{fl}{D}\right] \ldots\ldots\ldots [8]$$

where the subscript 2 refers to a condition for any length.

It is convenient to rearrange Equation [8] such that the dimensionless ratio fl/D is a function of k, the initial Mach number, and the pressure ratio. A regrouping of terms gives the form

$$\frac{fl}{D} = \frac{1}{kM_1^2}\left[1 - \left(\frac{p_2}{p_1}\right)^2\right] - 2 \log_e \frac{p_1}{p_2} \ldots\ldots\ldots [9]$$

Fig. 2 shows a plot of p_2/p_1 versus fl/D at several initial Mach numbers for isothermal flow. The dotted line in Fig. 2 represents limiting conditions. Fig. 3 shows the same type of plot for adiabatic flow. Except for a region close to limiting lengths, the solid curves in Fig. 2 are somewhat similar to those in Fig. 3.

COMPARISON BETWEEN INCOMPRESSIBLE FLOW AND COMPRESSIBLE FLOW

The relation for the flow of an incompressible fluid in a pipe is simpler than the relations for the flow of a compressible fluid. The question is frequently raised as to the limits of application of the incompressible-flow relations, and the possibility of using the incompressible-flow relation for studies of compressible flow. The question of application frequently depends upon the accuracy desired.

The pressure drop for the flow of an incompressible fluid in a horizontal pipe can be written as

$$p_1 - p_2 = \frac{flV_1^2}{2gDv_1}$$

$$\frac{p_2}{p_1} = 1 - \frac{k}{2}\,(M_1^2)\,\frac{fl}{D} \ldots\ldots\ldots\ldots\ldots [10]$$

Equation [10] was employed to plot the straight dot-dash lines in Figs. 2 and 3. A comparison of a dot-dash line and a solid line, for the same M_1, brings out the differences between compressible and incompressible flow. Figs. 2 and 3 show that the difference between the incompressible and the compressible relations is not very great at low Mach numbers and short lengths. The difference becomes appreciable, however, at high Mach numbers and long lengths.

A more direct comparison between the two types of flow can be given by the type of plot shown in Fig. 4. Equations [9] and [10] were employed to plot the curves for the three initial Mach numbers. The difference between compressible and incompressible flow may be large. For example, the compressible-flow relation gives a pressure drop of $p_1 - p_2 = 0.6p_1$ for certain conditions with an initial Mach number of 0.30. The incompressible-flow relation for the same pipe length and same initial conditions gives a pressure drop of about $p_1 - p_2 = 0.305p_1$.

In some discussions of isothermal flow, the change in kinetic energy is considered negligible. The statement is sometimes made that, if the pipe is "long," the term $2 \log_e V_2/V_1$ is negligible in comparison with the term fl/D. If the change in kinetic energy is neglected, then Equation [8] takes the form

$$p_1^2 - p_2^2 = \frac{V_1^2 p_1}{g v_1} \frac{fl}{D} \qquad [11]$$

Equations [10] and [11] can be written as

$$\text{Compressible flow} \left\{ \frac{p_1 - p_2}{p_1} = 1 - \sqrt{1 - \frac{flV_1^2}{gDp_1v_1}} = 1 - \sqrt{1 - 2B} \qquad [12] \right.$$

$$\text{Incompressible flow} \left\{ \frac{p_1 - p_2}{p_1} = \frac{flV_1^2}{2gDp_1v_1} = B \qquad [13] \right.$$

where B represents the dimensionless ratio given in Equation [13]. Equations [12] and [13] can be compared for identical values of B. A graphical comparison is shown by the curve A in Fig. 4.

Sometimes the rule is given that the incompressible-flow relation can be used for isothermal compressible-flow problems if the pressure drop $p_1 - p_2$ is less than 10 per cent of the initial pressure p_1. Curve A illustrates the accuracy of this rule.

A comparison of curve A with the other solid curves in Fig. 4 shows the effect of the change in kinetic energy. There is a considerable difference between curve A and the curve for $M_1 = 0.20$, for example, at long lengths. For long lengths, it appears that the change in kinetic energy can be neglected only for low Mach numbers.

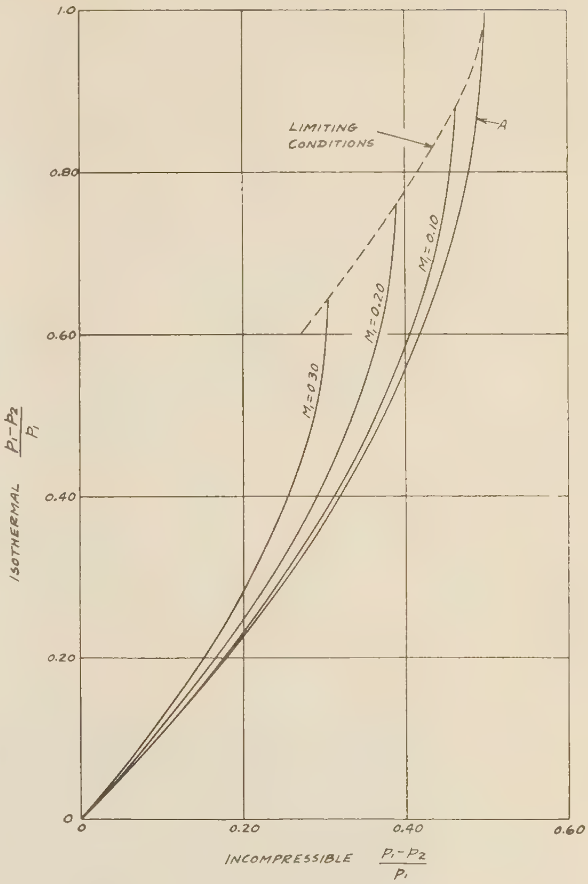

Fig. 4 Pressure-Drop Ratio for Isothermal Flow as a Function of Pressure-Drop Ratio for Incompressible Flow

Application and Design of Package Conveyers

By H. C. KELLER,[1] SYRACUSE, N. Y.

Conveyers used for material handling are divided into two groups, namely, for bulk materials like ore, gravel, coal, ash, sand, and so on, and for unit loads such as cartons, boxes, barrels, sacked goods, tote boxes, etc. This paper will discuss the latter group, dealing with application and design. In giving the various formulas on design, certain liberties with theory are taken and short cuts are used, but all formulas have been checked and proved by actual performance.

Roller Conveyers

ROLLER conveyers, Figs. 1 and 2, consist of side rails in which rollers are mounted. The load then travels by gravity on the rollers or is propelled along the line of conveyer.

Application. The load to be conveyed must have a rigid riding surface in order to span the space between the rollers. Fragile loads should not be permitted to travel by gravity. The best results are obtained when the loads are constructed of similar materials and are approximately uniform in weight. Heavy loads (in excess of 100 lb) should not be allowed to travel by gravity, without first making a careful investigation of all conditions to be certain that no damage will result. Loads up to 40,000 lb have been handled on roller conveyers, but such loads should not be allowed to operate by gravity.

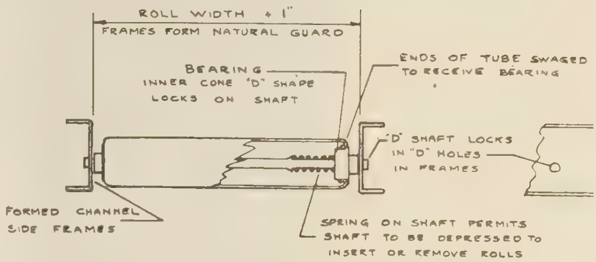

Fig. 1 Section Through Roller Conveyer

Design. The side rails are structural-steel members (usually angle or channel shapes), designed for maximum deflection of 0.33 in. in a 10-ft span.

The rollers are constructed of tubing with ball bearings adapted to the ends of the tubing. A shaft extends through the hubs of the bearings into the side rails. The tubing should be of sufficiently heavy gage to withstand damage due to impact of loads on the conveyer. The amount of impact is not always known,

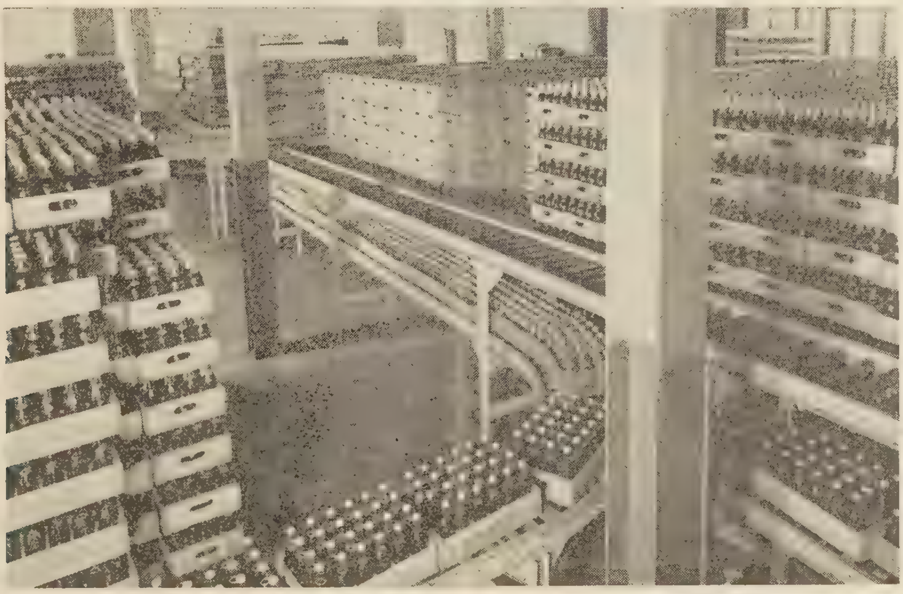

Fig. 2 Typical Roller Conveyer in Bottling House
(The Coca-Cola Company, Syracuse, N. Y.)

TABLE 1 DETAILS AND CAPACITIES OF ROLLER CONVEYERS

Roll diameter, in.	Load capacity of roll lb	Wall thickness
1 to 2	50 to 200	18 gage to 12 gage
$2^1/_8$ to $2^9/_{16}$	150 to 1000	16 gage to 7 gage
$2^3/_4$ to 4	500 to 5000	10 gage to $^1/_2$ in.
over 4	7500	$^3/_4$ in.

[1] Lamson Corporation.

Contributed by the Special Design Committee and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

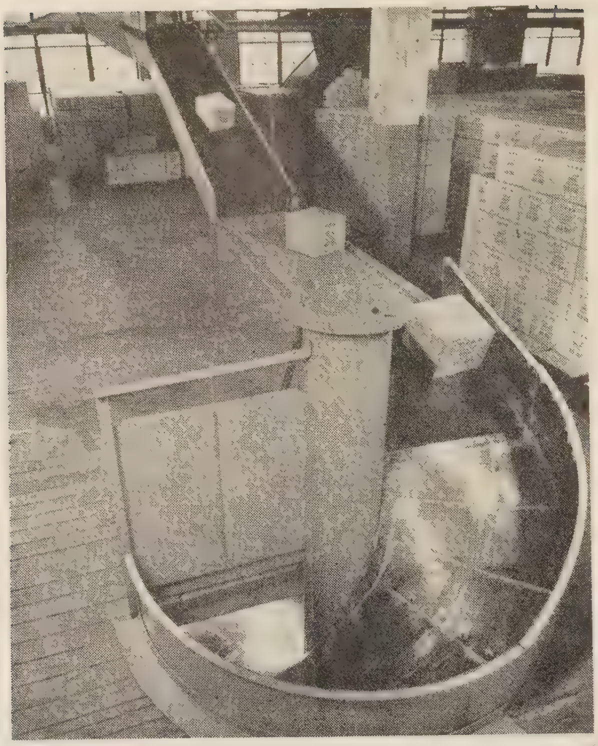

Fig. 3 Straight Chute Receiving Loads From Belt Conveyer and Leading to a Spiral Chute
(S. C. Johnson & Son, Racine, Wis.)

and we must depend on past experience. The wall thicknesses, indicated in Table 1, have proved satisfactory.

In almost every case, the wall thickness is greater than required to make the tubing structurally as strong as the bearings and shaft.

The shafts are designed so that they lock in the side rails, and the inner races of the bearings are in turn locked to the shaft. This prevents the bearing from turning on the shaft, and the shaft from turning in the side rails.

Generally the ball bearings are of the open free-running type, but frequently wet, dirty, or acid conditions require a grease-packed ball bearing. These latter bearings are sometimes equipped with grease fittings so that the grease can be replenished.

As the bearings are mounted in the ends of the rolls and the bearings are always adjacent to side rails, the capacity of the roll is not materially decreased as the length increases.

There should always be at least three rolls under the load, and the spacing of rolls in the frame is determined by the formula

$$\text{Maximum spacing} = \frac{\text{Minimum length of load}}{3}$$

The grade at which the conveyer is set depends upon the weight of the load and the type of riding surface. The type of bearing used in the roll also affects the grade. As each manufacturer has his own standard diameter of roll, thickness of tubing wall (affecting the dead weight), type of bearing, and so on, it is impossible to establish gradients for the entire line of roller conveyers. Generally, a grade of $^1/_2$ in. per ft will be satisfactory for cartons; $^3/_8$ in. per ft for steel tote boxes or hardwood boxes weighing between 15 and 75 lb. On hard-bottom loads, the grade is reduced as the weight increases, and this is also true for some cartons, but in the latter case only to the point where the weight does not cause the bottom of the carton to drape over the rolls.

Steel Chutes

Description. These chutes, Fig. 3, are made straight, curved, and helical (better known as "spiral chutes").

The straight chute consists of a bedplate and two side plates or guards.

The curved chute consists of a bedplate with outer and inner guard plates.

The spiral chute is usually built around a center core with a bedplate and outer guard plate.

Application. As all chutes are used only to guide loads from an upper level to a lower point, they should not be used when there is great variation in weight; when the weight is over 300 lb, or for fragile loads. The spiral chute offers some opportunity for control of speed by varying the shape of the bed and the pitch and is therefore preferred in most cases to straight chutes.

Atmospheric conditions also affect the operation of chutes, especially when cartons or sacked goods are being conveyed. This type of conveyer very seldom gives satisfactory service where there is likely to be great variation in the humidity of the surrounding air.

Design. Table 2 gives the gages of metal usually used in chutes. Straight chutes are frequently made by bending up side guards from the bedplate, in which case the gage specified for the bed is the one to use.

TABLE 2 METAL GAGES FOR CHUTES

Type of load	Straight chute Bed	Straight chute Guard	Curved chute Bed	Curved chute Guard	Spiral chute Bed	Spiral chute Guard
Cartons or sacks..	14	16	14	14	14	14
Wood boxes......	14	14	14 to 12	14 to 12	12	12
Wire-bound boxes or steel pans....	12	14	12	12	10	10

Under average conditions, straight chutes are declined at a 20-deg angle for cartons, 18 deg for wood boxes, and 15 to 17 deg

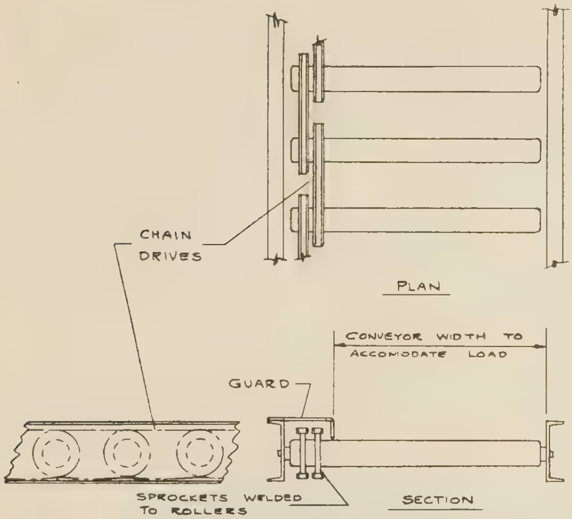

FIG. 4 MULTIPLE-STRAND, CHAIN-DRIVEN, LIVE-ROLL CONVEYER

for steel pans. On curved and spiral chutes the angle at the outer guard rail is the same as the angle of decline specified for straight chutes. The height of the side guards on straight chutes should be no less than one third the height of the highest load. On curved and spiral chutes, the outer guard rail should be somewhat higher than the greatest height of the load. The height of the guard is measured at right angles to the bed.

Live-Roll Conveyers

Description. This type of conveyer, Figs. 4 to 7, employs the use of roller-conveyer sections with various means used to power the rolls. Flat belts or round belts are snubbed against the underside of the rolls by means of snubbing rolls or sheaves, placed between the carrying rolls. The flat-belt type is used when the conveyer is straight and the round belt when there are curves in the conveyer. Another type of live-roll conveyer employs roller-chain drives over sprockets fitted to the roll shafts. In this case the roll and shaft are made integral and the shaft operates in bearings built into or mounted on the side rails. In other types of chain-driven conveyers, the sprockets are welded to the roll tubes, and the conventional roll with bearings adapted to the ends of tubing are used.

Application. As in the case of roller conveyers, the load must have a rigid riding surface but unlike the roller conveyer any variation in weight of the various loads is not important. Fragile loads should not be conveyed if the conveyer is to be used for storage, as blocking these loads will damage them.

Design. In addition to the features given for the roller conveyer, we must consider the matter of power required to operate the conveyer.

To determine the horsepower required to operate horizontal live-roll conveyers, the percentages of friction, given in Table 3, should be used.

TABLE 3 PERCENTAGE OF FRICTION IN LIVE-ROLL CONVEYERS

Type of conveyer	Weight of load, lb	Friction, per cent
Belt-driven..........	Up to 75	6
Belt-driven..........	Over 75	10
Chain-driven........	Unlimited	5

The percentages, given in Table 3, are based on antifriction bearings. Also, it may be advisable to explain why different frictions are given for belt-driven live-roll conveyers. The belting on this type of conveyer should never be snubbed more than just

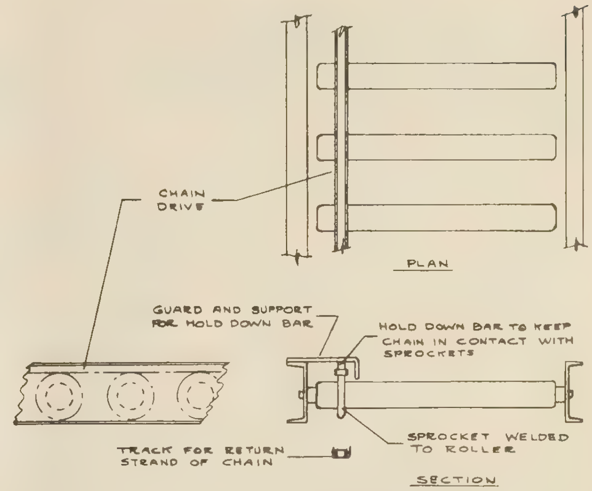

FIG. 5 SINGLE-STRAND, CHAIN-DRIVEN, LIVE-ROLL CONVEYER

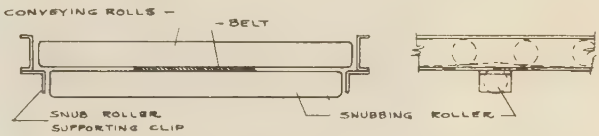

FIG. 6 FLAT-BELT-DRIVEN LIVE-ROLL CONVEYER

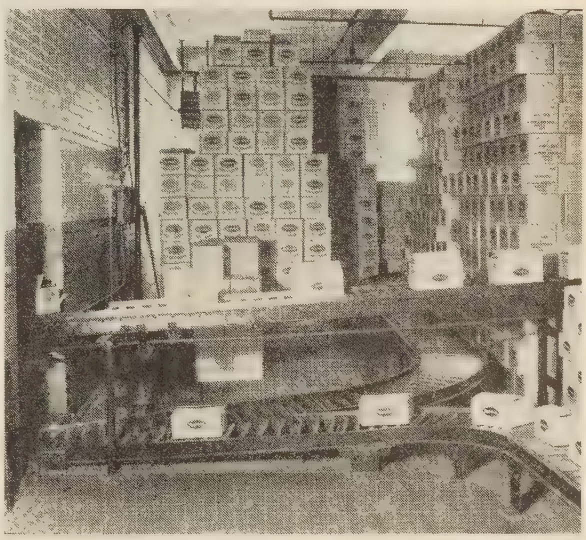

FIG. 7 LIVE-ROLL CONVEYER DRIVEN BY ROUND BELT SHOWN OVER ROLLER CONVEYERS
(Norwich Pharmacal Company, Norwich, N. Y.)

enough to propel the loads. It naturally follows that greater snubbing action is necessary as the weight of the load increases.

The horsepower required at the head shaft of belt-driven conveyers is

$$\text{Hp} = \frac{(W + w)\cdot f \cdot S}{33{,}000}$$

where W = weight of live load on conveyer, lb
w = weight of all rolls and belting, lb
f = friction given in Table 3
S = speed of conveyer, fpm

In the chain-driven type of conveyer, one design consists of a

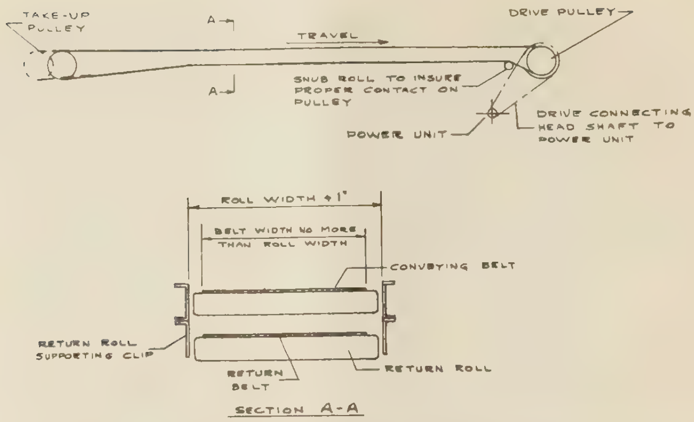

Fig. 8 Typical Belt Conveyer

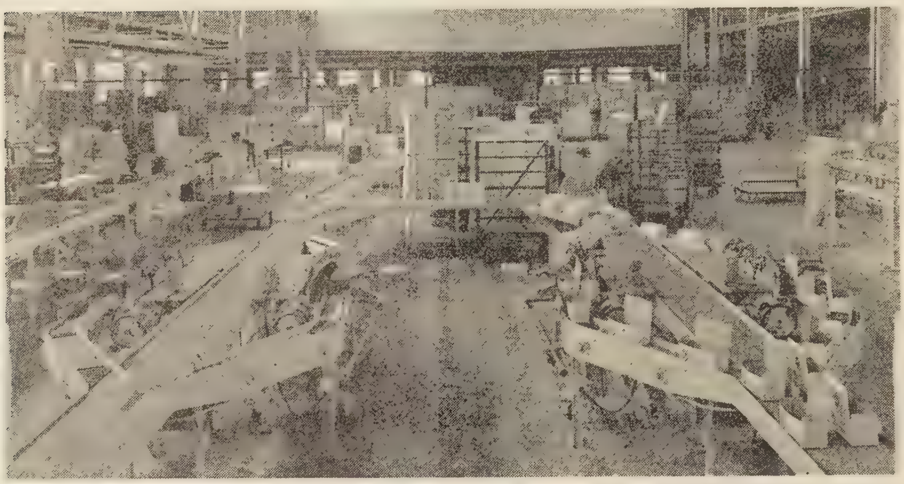

Fig. 9 Typical Belt-Conveyer Installation

(This shows two main-line belts bringing filled cartons from a packing table. Each belt carries a double line of cartons which are deflected to spur sidings where the cartons pass through closers and convey back into one line; the cartons now deflect to a table as shown for loading to trucks, or they may pass the deflector and deliver to the bundling conveyer.)

single strand of chain, while in another, multiple strands of chain are used. In the latter case two sprockets are employed on each roll and a complete drive is used from one roll to the next roll. The single strand of chain allows only a tangential contact with the sprocket and should only be used on light-duty conveyers. The multiple-strand design permits full 180-deg contact on each sprocket.

On single-straight-chain conveyers, the horsepower formula given for belt-driven conveyers can be used by changing the value of w. In this case, w = weight of rolls, sprockets, and chain.

On multiple-strand-chain-driven conveyers, the effective pull on each drive must be figured progressively, allowing 98 per cent efficiency for each chain drive.

Belt Conveyers

Description. Belt conveyers, Figs. 8 to 10, also employ roller-conveyer sections with belting supported by the rolls. In another type, the belting is supported by a steel or wood bed over which the belting slides.

Application. As the load is supported directly on the belting, almost any type of unit load can be conveyed on this conveyer. Unit loads weighing in excess of 50 lb per sq ft should not be con-

Fig. 10 Horizontal and Declined Belt Conveyers
(Sprague Warner)

veyed on the slider-bed type of construction, as excess weight causes the belting to seize to the bed of the conveyer.

Design. When the roller type of intermediate support is used, the spacing of the rolls on the load-carrying bed is determined by the length and weight of the load. When extremely light loads are to be conveyed rolls can be spaced 24 in. and in some cases 36 in. on centers. Heavier unit loads such as cartons, boxes, and so on, require roll spacing no greater than $^1/_2$ the length of the shortest load. The return belt on either type of conveyer is supported by rolls usually spaced no more than 10 ft on centers.

The horsepower required at the head shaft of horizontal belt conveyers is

$$\text{Hp} = \frac{(W + w) \cdot f \cdot S}{33{,}000}$$

where W = weight of live load on conveyer, lb
w = weight of rolls and belting, lb
f = friction given in Table 4
S = speed of conveyer, fpm

TABLE 4 PERCENTAGE OF FRICTION OF BELTING ON ROLLER AND SLIDER BEDS

Type of bed	Cotton or untreated-canvas belting, per cent	Rubber-impregnated canvas belting, per cent
Roller	3 to 5	3 to 5
Steel slider	20	21
Wood slider	22	30

There are numerous types of belting, each best suited for specific operating conditions. A complete study and analysis is impossible in this paper, but, generally, the cotton and untreated-canvas belt should be used where atmospheric conditions are fairly constant, as is the case in most indoor installations. The rubber-impregnated belt is used where there is great variation in humidity, and, in many cases where the belting is exposed directly to the elements, completely rubber-covered belting should be used. The latter belting is never used on slider-bed types of conveyers.

Belt conveyers can be inclined to elevate or to lower loads. Correction for the horsepower due to the incline must be made, and the conveying run of belting must be calculated independently of the return run. In elevating conveyers, the horsepower at the head shaft is

$$\text{Hp} = \frac{[0.05\, r + L \sin\theta + fL \cos\theta + 0.05\, l \cos\theta - l \sin\theta] \cdot S}{33{,}000}$$

where r = weight of all rollers, lb
L = weight of upper run of belting and live load, lb
l = weight of return run of belting, lb
θ = angle of incline, deg
f = friction given in Table 4
S = speed of conveyer, fpm

When the conveyer is used to lower loads the formula becomes

$$\text{Hp} = \frac{[0.05\, r + l \sin\theta + fL \cos\theta + 0.05\, l \cos\theta - L \sin\theta] \cdot S}{33{,}000}$$

It is frequently found that this formula gives a minus value, but it must be considered the same as a plus value in order to obtain the required strength in the drive gearing to hold back the load and prevent the conveyer from running away.

Another way of expressing the head-shaft horsepower is to substitute the term "effective pull" for all load factors in the preceding formulas

$$\text{Hp} = \frac{\text{Effective pull} \times \text{speed}}{33{,}000}$$

The effective pull permits us to determine other factors, such as total load on the head shaft, and the stress in the belting. The tension in the tight side of the belting at the head pulley is expressed as T_1 while the tension in the slack side is expressed as T_2. Table 5 gives the values of T_1 and T_2 for the most common conditions.

TABLE 5 TENSION IN BELTING AT HEAD PULLEY

Amount of wrap on pulley, deg	T_1	T_2
180	1.64	0.638
200	1.54	0.541
210	1.50	0.500
220	1.46	0.462
230	1.43	0.428

To obtain the maximum stress in the belting, multiply the effective pull by the proper factor for T_1. The maximum load on the head shaft is the sum of the effective pull multiplied by the factor T_1 plus the effective pull multiplied by factor T_2.

A take-up pulley is provided in the conveyer to compensate for stretch in the belting. The take-up is usually made by screw adjustment. The amount of adjustment varies with the type of belting. Canvas belting requires a take-up adjustment of 1 per cent of the length of the conveyer. Rubber belting requires approximately one half the adjustment provided for canvas belting.

Automatic weighted take-up pulleys are used on long conveyers and sometimes on shorter conveyers when there is a wide range in surrounding air temperature and humidity. The weight required to operate this type of take-up is twice the value of T_2 or

$$\text{Weight} = 2 \times \text{effective pull} \times \text{factor } T_2$$

If the conveyer is to be reversed to convey loads in opposite direction at different times, the weight should be twice T_1 or

$$\text{Weight} = 2 \times \text{effective pull} \times \text{factor } T_1$$

Slat Conveyers

Description. Slat conveyers, Figs. 11 to 13, are constructed with wood or steel slats (pallets) fastened to attachments on two strands of roller chain, which in turn travel in or on structural-steel tracks.

Application. This type of conveyer is used when the character of the load is such that it would damage the belting on belt conveyers or does not have a riding surface adaptable to travel on live-roll conveyers.

Design. The slats, Fig. 12, are designed as a simple beam supported at either end. The width of each slat is somewhat less than the pitch of the chain, i.e., about $3^1/_2$ in. wide for a 4-in-pitch chain, $5^1/_2$ in. wide for a 6-in-pitch chain, etc.

Wood slats are usually standard commercial sizes, as 2 in. × 4 in., 2 in. × 6 in., etc., dressed on all four sides.

Steel slats may be flat bar, with or without reinforcing members, formed or rolled channel shapes, etc.

When plain rollers are used in the chain, the tracks are structural angles or channels. On heavy-duty conveyers, a flanged roller is used in the chain, and standard rail sections are then used for the tracks.

The horsepower required at the head shaft of a horizontal slat conveyer is

$$\text{Hp} = \frac{(W + w) \cdot f \cdot S}{33{,}000}$$

where W = weight of live load on conveyer, lb
w = weight of chains and slats, lb
f = friction as given in Table 6
S = speed of conveyer, fpm

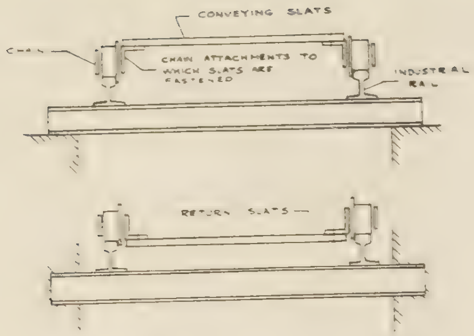

Fig. 11 Section Through Slat Conveyer
(Showing construction using flanged rollers in chain.)

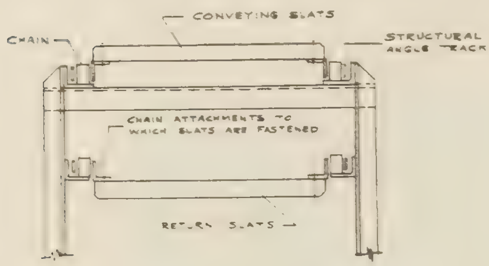

Fig. 12 Section Through Slat Conveyer
(Showing construction using plain or straight rollers in chain.)

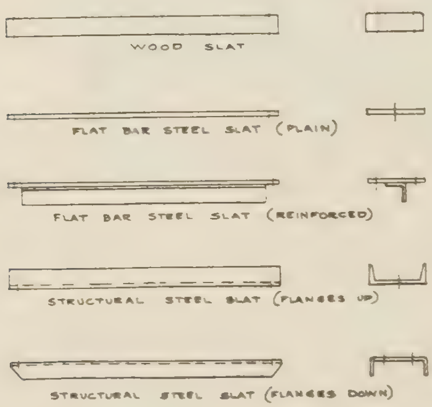

Fig 13 Some of the More Popular Forms of Slats

where L = weight of upper run of chains, slats and live load, lb
l = weight of return run of chains, and slats, lb
θ = angle of incline, deg
f = friction given in Table 6
S = speed of conveyer, fpm

and when loads are to be lowered

$$\text{Hp} = \frac{[l \sin \theta + f(L + l) \cos \theta - L \sin \theta] \cdot S}{33{,}000}$$

In the latter case, any minus value must be treated as a plus value as explained for belt conveyers.

Take-ups are provided to compensate for stretch in the chains. On this type of conveyer, the adjustment is of the screw type. The amount of adjustment is only enough to permit a chain link to be readily removed and has no relation to the length of the conveyer.

Apron Conveyers

Description. The apron conveyer, Fig. 14, is a modification of the slat conveyer in which the aprons or slats form a continuous bed. The aprons overlap one another and in some cases sides are provided so that the bed of the conveyer presents a continuous trough.

Application. This conveyer is seldom used in package conveying. It is employed only when small loads are to be conveyed and where the spaces between the slats on a slat conveyer would be objectionable.

Design. Refer to "Slat Conveyers," substituting the weight of aprons for slats in the horsepower formulas.

Drag-Chain Conveyers

Description. In this type of conveyer, Figs. 15 and 16, single or multiple strands of chain slide in steel tracks conveying the load directly on the chain.

Application. Single-strand chains are employed successfully when a uniform-size load is to be conveyed, and multiple strands when loads vary in size. This type of conveyer can negotiate 90-deg corners very readily with most loads. It is particularly adaptable to bottling houses, dairies, and breweries where the loads generally are uniform in size and weight.

Design. Refer to "Slat Conveyers," substituting the weight of the chain only for chains and slats in the horsepower formulas, and use a sliding friction of 30 per cent for the chain on the steel track.

When horizontal turns are employed, the effective pull of each section must be figured progressively.

The turns are usually designed so that the chain is guided in a steel track. Wheels or rollers may be used to reduce the friction loss at the turns, but it is very difficult to make an economical design because of the necessity of clearing the load resting directly on the chain.

Fig. 14 Detail of Apron Conveyer

TABLE 6 PERCENTAGE FRICTION OF CHAINS ON STEEL TRACKS

Diameter of roller in chain, in.	$1^1/_2$	2	$2^1/_2$	3
Friction, per cent	20	20	15	12

The horsepower for inclined conveyers elevating loads is

$$\text{Hp} = \frac{[L \sin \theta + f(L + l) \cos \theta - l \sin \theta]S}{33{,}000}$$

Overhead Chain Conveyers

Description. Overhead chain conveyers, Figs. 17 to 19, consist of a two-plane flexible chain supported from an overhead track by means of trolleys. Various types of attachments, from which the loads are suspended, are spaced on the chain. Frequently, cars or carriers are suspended from the attachments and the loads are then placed directly on the cars.

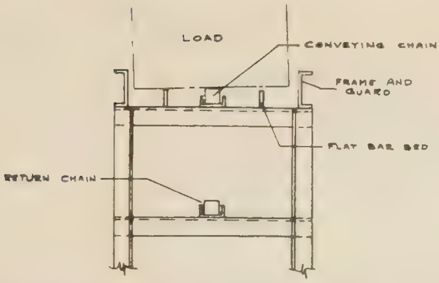

Fig. 15 Section Through Drag-Chain Conveyer

Fig. 16 Drag-Chain Conveyer With Pushers Automatically Converging Loads From Two Lines to a Single Line of Roller Conveyer
(Pepsi-Cola, Long Island City, N. Y.)

Application. Almost any type of load can be conveyed by this conveyer. The conveyer is extremely flexible; loads may be deposited on the cars or hooked to the chain attachments at any working height, the load then being elevated to the ceiling or floors above for transportation and again brought down to a working height for unloading at the next operation. The conveyer makes an ideal storage unit or traveling stock pile as any load not removed at a station will recirculate and keep returning to that station until the operator is ready to take it off and perform his work. The floor between operations is kept clear for other work and the otherwise wasted space at the ceiling is utilized to good purpose.

Design. The most common types of conveyer are the 3-in. and 4-in. sizes. These sizes indicate the size of I-beam track used. Two-wheel trolleys, operating in the I-beam, support the chain, and the load attachments are usually arranged directly below the trolley so that the pull is on the trolley, the chain then becoming the propelling means for advancing the trolleys and loads along the path of the track.

Horizontal turns are made in two types, the multiple-roller curve and the traction wheel. The roller type is more generally

Fig. 17 Overhead Chain Conveyer Towing Trucks

Fig. 18 Typical Loading-Station Dip on Overhead Chain Conveyer
(Glen L. Martin, Baltimore, Md.)

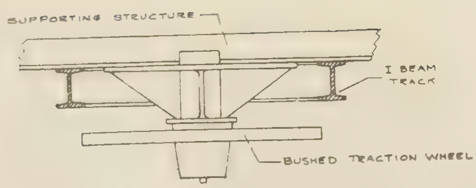

Fig. 19 Traction Wheel for Overhead Chain Conveyer

used because of its lower frictional resistance and its adaptability to large radii. When the turn is to operate in high temperatures (over 400 F), the traction wheel should be used, as it is provided with a special bushing and a large oil reservoir supplying ample lubrication.

The drives are made in two types, the corner-sprocket drive and the caterpillar drive. The corner-sprocket drive is used when conditions permit, and the caterpillar drive where the radius of the corner is too large for sprocket drives. The drive should be placed at the point of maximum pull and, preferably, near a down curve so that any slack in the chain will automatically be taken away from the drive. Drives should never be placed inside oven or spray-booth enclosures.

Take-ups are usually provided to permit removal of links when slack in chain becomes excessive. Take-ups may be omitted on long conveyers or those with several dips where links may be removed from the chain without the necessity of adjusting a sprocket or corner. Take-ups should always be provided on conveyers subjected to great temperature changes. In this latter case, the track should also be provided with expansion joints based upon a coefficient of 0.0000065 per unit length per deg F.

Radii of vertical bends or dips should be as large as practical for smooth operation.

The technical data, given in Table 7, will be helpful in designing the conveyer.

TABLE 7 DESIGN DATA FOR OVERHEAD CHAIN CONVEYERS

	Size of system	
	3 in.	4 in.
Chain:		
Pitch	3.031	4.031
Weight per foot, lb	2.25	3.125
Ultimate strength, lb	18,000	30,000
Allowable working load in lb (level)	1800	3000
Allowable working load in lb (dips)	1000	2000
Minimum amount of take-up, in	7	9
Trolleys:		
Weight, lb	3.5	7.5
Multiple of spacing, in	6	8
Maximum spacing, in	30	32
Maximum load, lb	80 to 150	200 to 400
Average friction, ball bearings, per cent	2	2
Average friction, bronze bearings, per cent	3	3
Track:		
Size of I-beam, in	3	4
Weight per foot, lb	5.7	7.7
Vertical Bends:		
Trolley spacing, minimum radius	30 in.–10 ft	32 in.–12 ft
Trolley spacing, minimum radius	24 in.– 8 ft	24 in.–10 ft
Trolley spacing, minimum radius	18 in.– 6 ft	16 in.– 8 ft
Trolley spacing, minimum radius	12 in.– 5 ft	...

To determine the horsepower at the head shaft, the formula is

$$\mathrm{Hp} = \frac{W \cdot f \cdot S}{33{,}000}$$

where W = weight of chain, trolleys, cars, and live load, lb
f = friction, depending upon type of bearing in trolley
S = speed of conveyer, fpm

When the conveyer has small dips or vertical bends, they usually balance, and there is no appreciable extra pull in the chain. If, however, the inclines are long, the extra pull must be added to the calculation, and the horsepower formula then becomes

$$\mathrm{Hp} = \frac{(W \cdot f + L \sin\theta) \cdot S}{33{,}000}$$

where L = live load on incline, lb
θ = angle of incline, deg

When the conveyer has more than four horizontal turns, the pull on the chain must be calculated for each section progressively, adding 5 per cent friction loss for each turn.

Vertical Conveyers

There are numerous types of vertical conveyers (Fig. 20 is an example). There are special applications of vertical belt conveyers and vertical slat conveyers. In both cases, cars or shelves are attached to the belt or slats and the loads then placed on these shelves.

Other continuous-operating vertical conveyers employ single or double strands of chain, attached to which are rigid or pendant cars. The loads sometimes are placed on and taken off the cars manually. The cars can be designed with fingers or arms that pass through corresponding fingers on stations so that loads are deposited automatically on and off the cars. The fingers of the stations may be constructed of rollers so that loads travel automatically into or out of the path of the cars. The station fingers may also be narrow belt conveyers, slat conveyers, drag chains; in fact any type of powered conveyer.

Vertical conveyers may also be of the reciprocating type where a single car delivers the load to another station and then returns to the sending station to receive another load.

There are unlimited combinations and designs of vertical conveyers, and it would be impossible to describe in detail even a few of the more popular types in this paper.

Special Conveyers

There are a great many special conveyers which are really modifications of the more standardized types of conveyers discussed in

Fig. 20 Reciprocating-Type Vertical Conveyers, Illustrating Typical Automatic Loading and Unloading Stations
(W. T. Gilmore Drug Company, Pittsburgh, Pa.)

this paper. There is one possible exception and that is the pusher-bar conveyers. This conveyer consists of two strands of chain with a bar extending between the chains. These bars push the load over a steel slider bed, a wood slider bed, or a roller bed. The pusher-bar conveyer at one time was very popular (it is still used extensively in bottling houses), but inclined belt conveyers are rapidly replacing the pusher-bar type of booster.

Automatic conveyers are frequently designed to pick up loads and discharge them at predetermined points. One design of fully automatic conveyer consists of a vertical conveyer with automatic loading and unloading stations. In this installation, an operator places a load on the loading-station platform and sets a dial indicating the station to which the load is dispatched. When an empty car approaches, it operates the loading station which places the load onto the car and sets a mechanism on the car. When that car reaches the designated unloading station (and that station is empty and ready to receive a load), the station picks the load off the car and deposits it on the unloading-station platform.

Another example of fully automatic conveyer consists of an overhead chain conveyer with reciprocating type of vertical conveyers, loading and unloading the cars on the overhead conveyer. In this installation, the loads are placed on the vertical-conveyer car near the floor level. When an empty car on the overhead conveyer approaches, the vertical conveyer elevates the load and deposits it on the car on the overhead conveyer. The overhead conveyer then delivers the load to another part of the building, or to another building, where a vertical conveyer removes the load and lowers it to a station near the floor level.

Pneumatic tubes are also used to convey unit loads, but the application and design features are too complex to be discussed in this paper.

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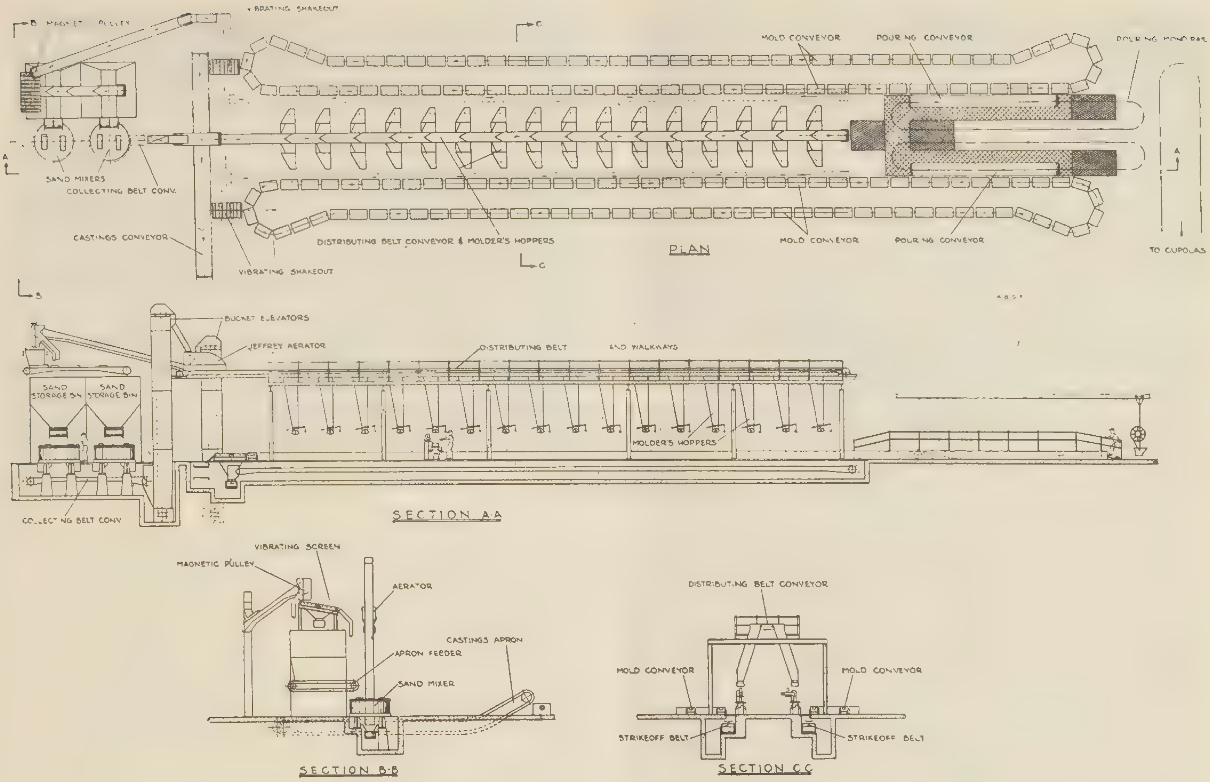

FIG. 1 TYPICAL LAYOUT OF A CONTINUOUS POURING SYSTEM, USING CONTINUOUS-POWER-DRIVEN MOLD CONVEYERS

Design Features of Conveying Equipment for the Foundry Industry

By F. B. HENRY and G. N. WILEMAN,[1] COLUMBUS, OHIO

The design of conveying equipment used in the foundry industry is covered in its general applications, with consideration given to the design of some of the more important details. Emphasis has been placed on equipment used in the handling of sand, molds, and castings, since they offer the major opportunities for economies and savings in manufacturing costs.

CONVEYING equipment for the foundry industry embodies conveyers and elevators handling practically all materials in the foundry, from raw materials to finished product. Raw materials, such as limestone, coke, sand, clay, pig iron, and so on, are handled mechanically from railroad cars to the point of usage.

Sand used in the molding process is handled mechanically from the shakeout through the reconditioning process, back to the molding stations.

Molds, after being made (except in the case of very large castings) are handled mechanically from the pouring and cooling stations to the shakeout. Flasks are commonly returned from the shakeout to molders by means of conveyers.

Cores used in making the molds are handled on conveyers from coremakers through the baking process, and frequently distributed to the molders.

Castings from the shakeout are often handled by conveyer through the cooling step to the cleaning room, and, in the case of production castings, are handled through the cleaning process on conveyers. Remelt at times is also returned to the melting operation by means of conveyers.

From the foregoing, it can be seen that conveying equipment plays a vital part in the operation of a modern foundry, and a conveying system often represents the largest equipment investment in the plant.

A few general rules should be kept in mind when designing equipment for any foundry. These are as follows:

1 Due to the extremely abrasive nature of foundry sand

[1] The Jeffrey Manufacturing Company.

Contributed by the Special Design Committee and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

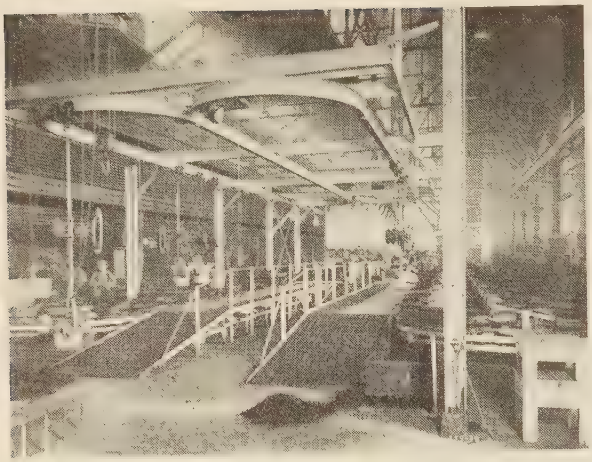

Fig. 2 Production Foundry Making Gray Iron

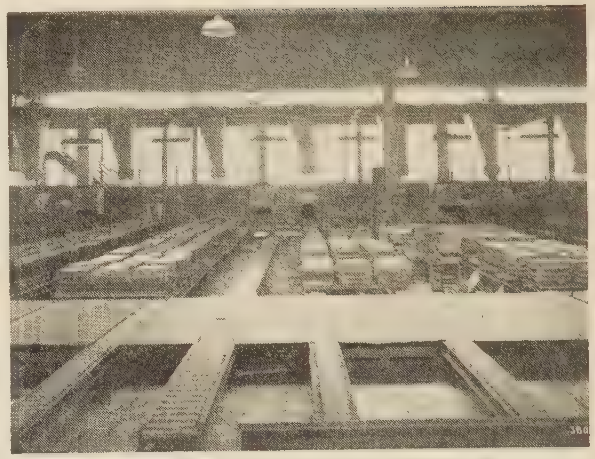

Fig. 4 Semicontinuous Foundry Making Small Steel Castings

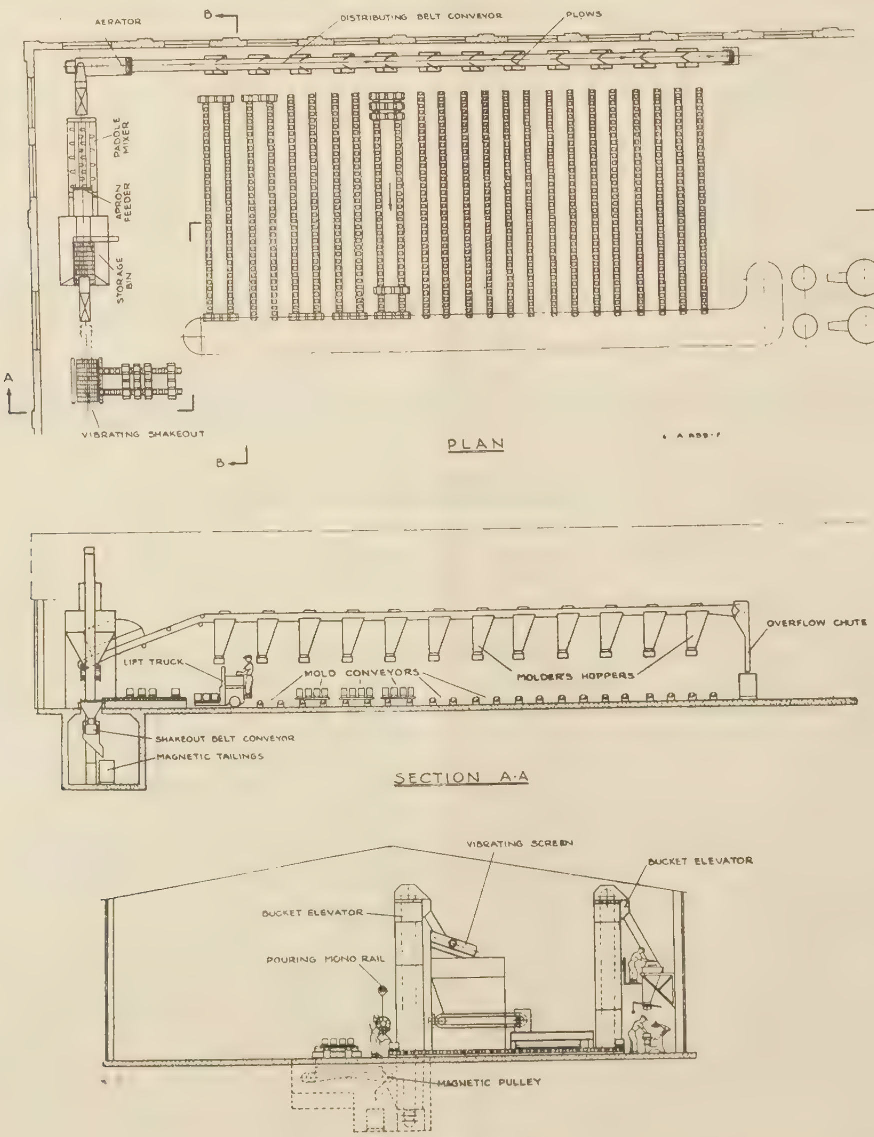

Fig. 3 Typical Layout of Foundry Using Roller-Type Conveyer, Mold Storage, and Central Shakeout

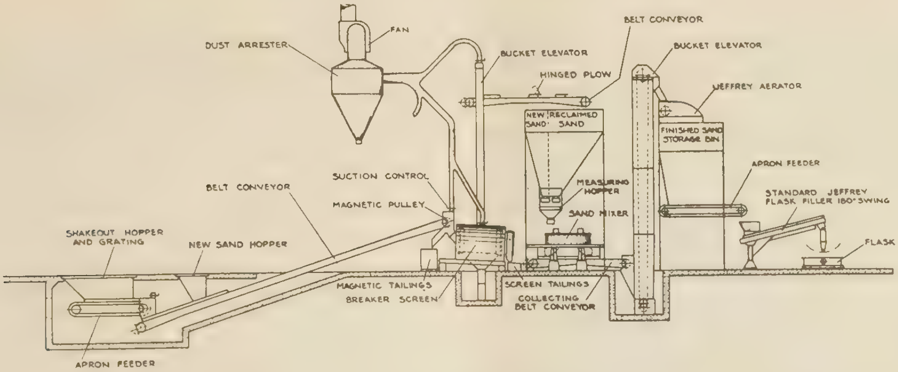

FIG. 5 TYPICAL LAYOUT OF STEEL FOUNDRY

and dust, friction parts should be entirely enclosed if possible, particularly those parts having high loads. This practically eliminates, or keeps to a minimum, all sliding types of conveyers.

2 Generous space should be provided in pits and walkways for proper maintenance and inspection.

3 Points in the system where dry sand is transferred from one conveyer to another, or from conveyer to bins, should have properly designed chutes and hoods for removal of dust, steam, smoke, and gases. Individual installations should be designed to meet state law requirements.

4 Chutes and hoppers handling prepared or damp sand should have angles of at least 60 deg, rounded corners, and generous cleanout doors.

TYPICAL ARRANGEMENTS FOR MECHANIZED FOUNDRIES

Before considering the individual types of conveyers used in the various applications, and their design features, it may be well to consider some typical layouts used in different types of mechanized foundries.

Layout of Typical High-Production, Continuous, Gray-Iron Foundry (Figs. 1 and 2). This arrangement lends itself to any foundry—steel, gray iron, malleable iron, aluminum, or magnesium—having high production of duplicate castings, with continuous hot metal available. Sand is prepared in the mixers, aerated, and distributed to the molders' hoppers at each molding station. After the mold is completed, it is placed on the mold conveyer for pouring, cooling, and conveying to the shakeout station. Sand passes through the shakeout machines, is screened, and conveyed to the storage bin for another cycle. The castings are collected from the ends of the shakeout machines and conveyed to the cleaning room.

Layout of Typical Gray-Iron or Brass Foundry; Noncontinuous (Figs. 3 and 4). This arrangement is applicable to any type of foundry having relatively small work and allows sufficient storage of molds to take care of intermittent metal supply. While this layout shows a paddle mixer in the sand plant, the muller type could be substituted if the sand required it. After mixing, the sand is distributed to the molders. Finished molds are stored on a roller conveyer for pouring and cooling. A car or conveyer will transfer the molds to the shakeout station, from which the sand is returned to the storage bin, while the castings may be put into boxes or conveyed directly to the cleaning room.

Layout of Typical Steel-Foundry System (Figs. 5 and 6). Steel foundries may use conveyers for handling molds from the molder to the shakeout, depending upon the class of work. Sand from the shakeout passes over a magnetic pulley, through a breaker screen to reduce lumps and remove refuse, then to the sand bins and mixers. Prepared sand is distributed by conveyer or bucket to the molding stations. In the case of large molds these are frequently hoppers fitted with feeders for filling the flasks.

FIG. 6 STEEL FOUNDRY WITH TRAMRAIL DISTRIBUTING SYSTEM

TYPES OF CONVEYERS USED

Following is a detailed discussion of various types of conveyers used in the typical systems described:

Shakeout (Figs. 7 and 8). The mechanical shakeout is now very widely used in all types of foundries and is available for molds from small snap molds, weighing possibly 100 lb, to very large molds weighing as much as 50 tons. They consist fundamentally of a vibrating perforated deck or grid on which the mold is placed, the sand falling through the grid to the conveying means underneath, with the flask and casting remaining on the grid or being discharged from one end. Large molds are frequently stripped before being shaken out. Supports for shakeout machines must be extremely heavy and rugged.

Feeder Under Shakeout. Means must be provided for conveying the sand from the hopper underneath the shakeout machine. This hopper is always designed to contain at least the total amount of sand in the largest mold being handled. This sand frequently is very hot and may contain considerable metallic refuse, such as gaggers and rods, as well as fins, sprues, and so on, which may be broken from the casting in the shakeout process. There are three types of conveyers used for feeding the sand from the shakeout:

(*a*) A belt feeder, Fig. 9, is sometimes used in cases where the castings are very light and the sand does not contain any ma-

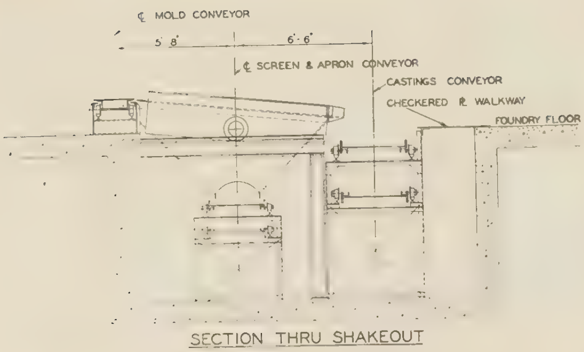

Fig. 7 Shakeout for Small Gray-Iron Castings

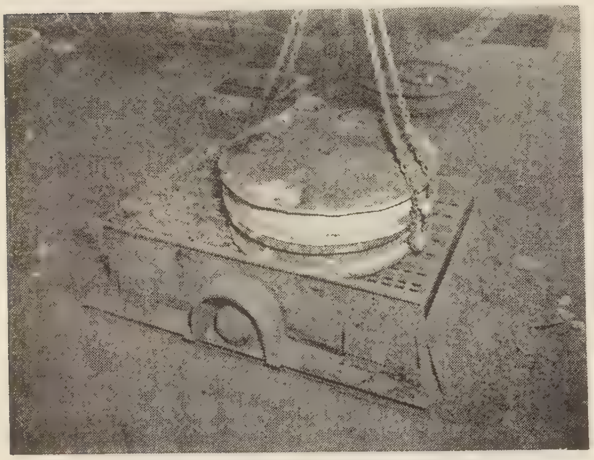

Fig. 8 Steel-Foundry Shakeout

Fig. 9 Belt Feeder Under Shakeout

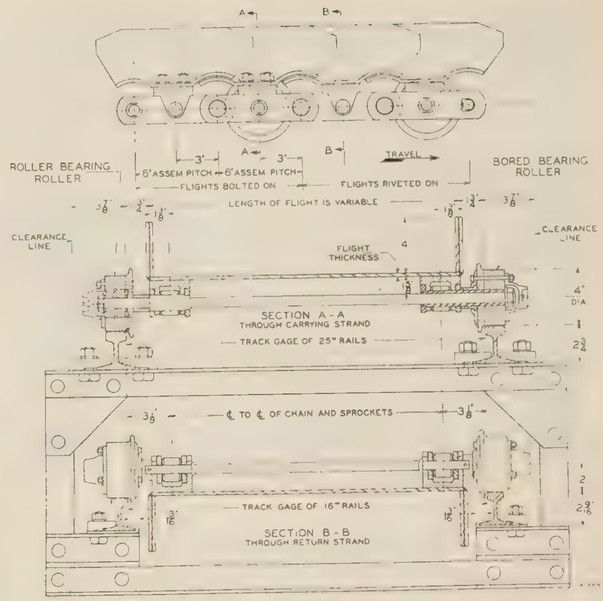

Fig. 10 Cross Section of Apron Feeder

Fig. 11 Apron Feeder Under Shakeout

terial amount of metallic refuse such as in a foundry making very light iron or brass castings. Closely spaced idlers or flat steel plate are required under the load.

(*b*) Apron feeders, Figs. 10 and 11, are commonly used for this service and where used should be of the leakproof type, with a tight-fitting enclosed outboard roller. Pans have welded ends to prevent leakage of dry sand and are removable from the conveyer without dismantling the chain. The drive should be protected by a shear pin or similar device. While the apron will stand heat in a satisfactory manner, damage frequently results to the pans from gaggers or rods which get through the shakeout grating.

(*c*) During the last few years, the electric vibrating feeder, Fig. 12, has come into wide usage in this application. It offers a perfectly smooth deck with no obstructions for rods or gaggers; it can stand the heat without injury, and is free from dribble since it has no return run.

Belt Conveyer Having Magnetic Pulley (Fig. 13). In the usual installation, sand is fed from the shakeout hopper to a belt conveyer equipped with a magnetic head pulley, for removal of magnetic refuse such as gaggers, rods, nails, sprues, shot, and so on. This is important for various reasons. It is essential that foreign material be removed to prevent injury to some of the other parts of the equipment. Gaggers and rods are salvaged for re-use. Sprues, shot, and other parts of the casting metal

are returned for remelt. Rubber belt is used when the sand is comparatively cool, with heat-resisting, impregnated-fabric belts for hot sands. Belt-conveyer idlers should have a high-pressure type of lubrication which forces out the old grease. Closely spaced idlers or a plate should be placed under the loading point of the conveyer to take the load.

A short belt with a magnetic pulley may be applied over the main belt for picking off large pieces of metallic material.

Screens (Figs. 14 to 16). Screens may be either of the rotary type or of the vibrating type and are used principally to get rid of nonmagnetic refuse such as wood wedges, tiles and rags, etc. Rotary-type screens are ordinarily used where heavy work is encountered, as they have a very definite lump-breaking and sand-cooling action. Vibrating screens are used where very fine sand is required, such as, in foundries making small castings. It is the usual practice to provide hoods and air suction on these screens for the removal of dust, fines, steam, etc.

Bucket Elevators. Bucket elevators used in the foundry industry are almost always of the centrifugal-discharge belt-and-bucket type, although chain elevators are used infrequently. Malleable buckets are universally used. Foot pulleys are of the slatted type with internal cones so that sand falling inside the belt will not build up on the foot pulley. Boots should have large and readily accessible cleanout doors.

Main Storage Bins (Figs. 17 and 18). Design of the storage bins depends upon the particular application and the method of feeding the sand from them to the mixers. The tendency in modern installations is to make them relatively large in capacity. In continuous systems, a minimum of 2 hr supply of sand is ordinarily used. Storage bins are often designed to include, in addition to the main compartment for old sand, one or more

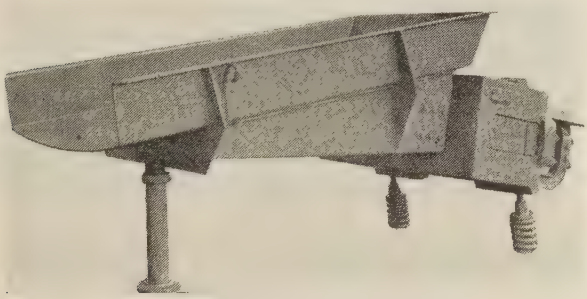

Fig. 12 Vibrating Feeder for Use Under Shakeout

Fig. 13 Shakeout Belt Conveyer Having Magnetic Head Pulley

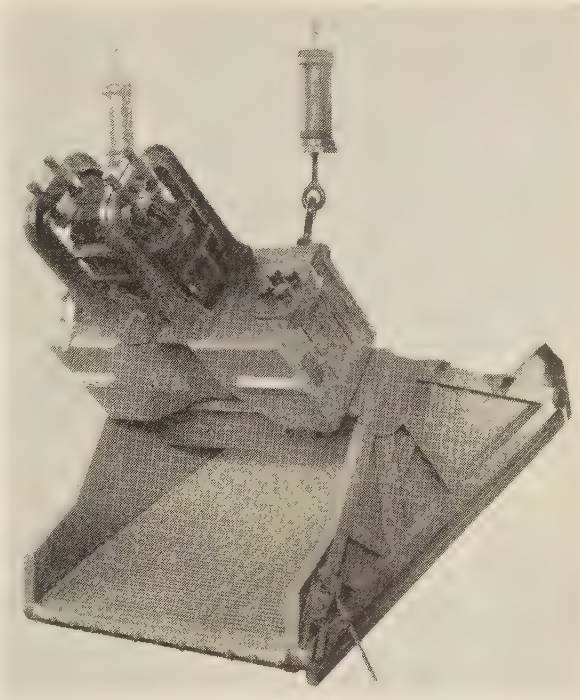

Fig. 14 Vibrating Screen

compartments for new sand or binders. Bins having apron feeders should be fitted with regulating baffles to insure a uniform sand withdrawal along the entire length of the bin.

Feeders From Storage Bins to Sand Mixers. There are two common methods of feeding the sand from the main storage bin to the sand mixers, either with gates and measuring hoppers, or apron feeders.

If muller-type mixers are used, the sand must be batched in suitable quantities. One method is the use of volumetric batch hoppers or of weigh hoppers, the batch hopper or weigh hopper containing the proper charge for the mixer. Weigh hoppers have the advantage of being able to properly proportion shakeout sand and new sand. The storage bins used with this type of batch method are gravity-type bins, provided with clamshell valves for feeding the batch hoppers which, in turn, are discharged by clamshell valves into the mills. With this type of bin it is best practice to divide the bin into two compartments, with two discharge valves, to prevent piping of the sand down through the center of the bin, Fig. 19.

The other method frequently employed uses apron-type feeders for charging the batch-type mixers. The feeders are arranged with either a limit switch, or timer so as to operate for a certain number of seconds to give the proper charge. This feeding arrangement will allow for more economical design of bin and one which is much lower in height, Fig. 20.

Paddle mixers use the continuous-moving-apron feeder almost exclusively. Vibrating feeders or small screw conveyers are used for feeding binders to the mixers, Fig. 21.

Sand Mixers. Sand mixers used are either of the paddle-mixer type or of the muller type. Paddle mixers are continuous mixers which give satisfactory results on natural sands. The muller-type mixers are batch mixers and are always used when synthetic sands are being made.

Aerators (Figs. 22 and 23). Aerators are used between the mixers and the distribution system on practically all installations.

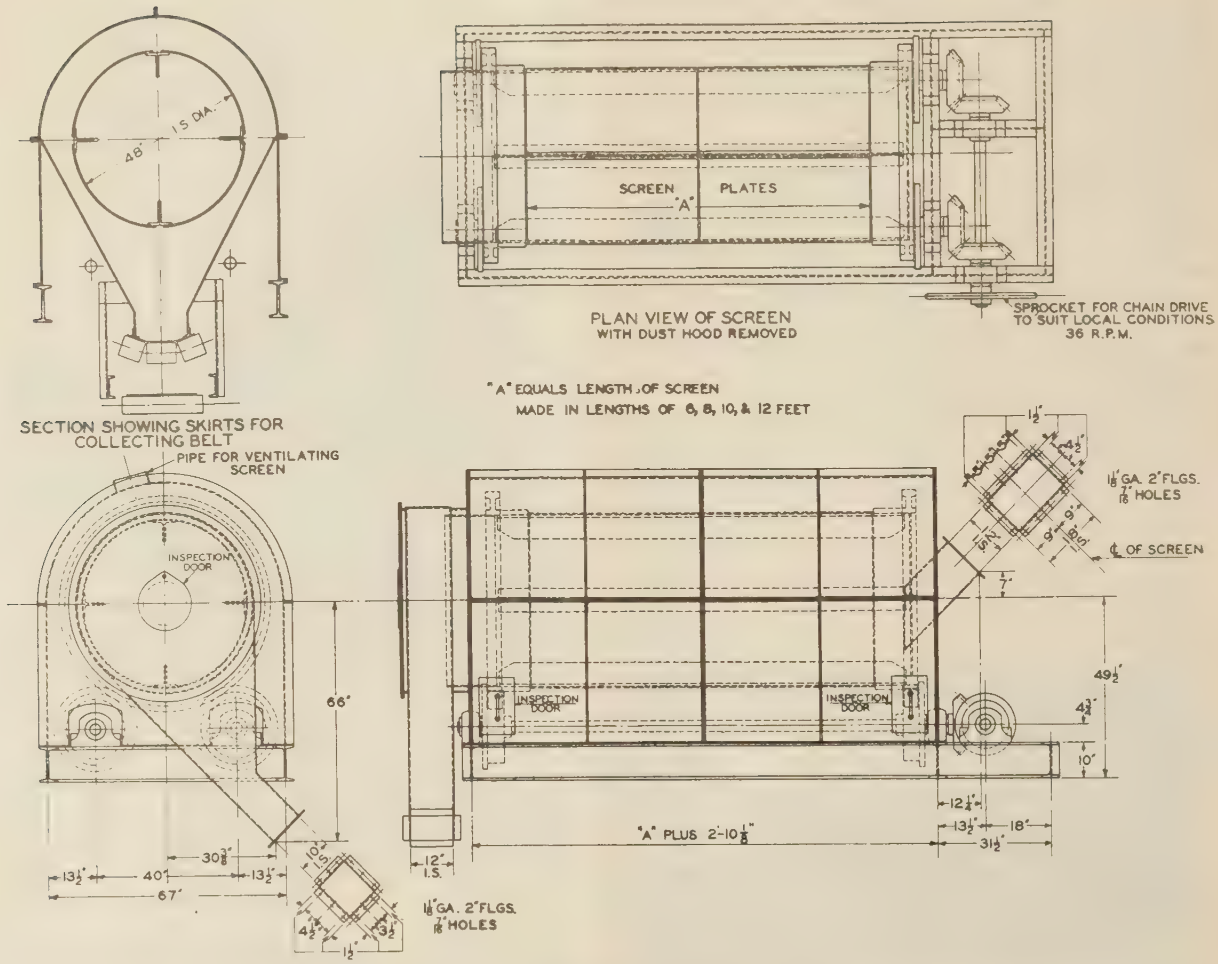

Fig. 16 Details of Rotary Breaker Screen

They are used for the final mixing and aeration of the prepared sand, before it is delivered to the molders.

Distributing System. Flat-belt conveyers with plows are most commonly used for distributing the prepared sand to the molders' hoppers. The belt is supported on flat idlers between the plows and on a flat plate directly under the plows. Plows are usually hinged, employing some simple mechanism for raising or lowering by hand. The part riding the belt should be edged with rubber to prevent injury. Belt cleaners should be installed at the head of this as well as any other belt handling prepared sand, Fig. 24.

In occasional installations, scraper-type conveyers are used. These scraper-type conveyers have the advantage that they will continuously and automatically fill the molders' hoppers. When one becomes full sand is scraped into next hopper, Fig. 25.

Hoppers for Molders. Hoppers and bins at the molding stations should be as small as practical. They usually contain from $1^1/_2$ to 2 tons of sand for smaller-size molds. They are designed with two vertical sides and with corners rounded to at least 3 in. radius. Clamshell valves are used for discharging the sand from the hopper to the mold. Quite frequently air cylinders are used for operating the valves, Fig. 26.

Where large molds are being made, it is necessary to provide a larger hopper, and in such cases the bottom of the hopper is fitted with an apron feeder for dragging the sand from the hopper into the flask, Fig. 27. Occasionally a swivel belt is employed between the end of the apron and the molding station. This allows the one hopper and feeder to service several molders, Fig. 28.

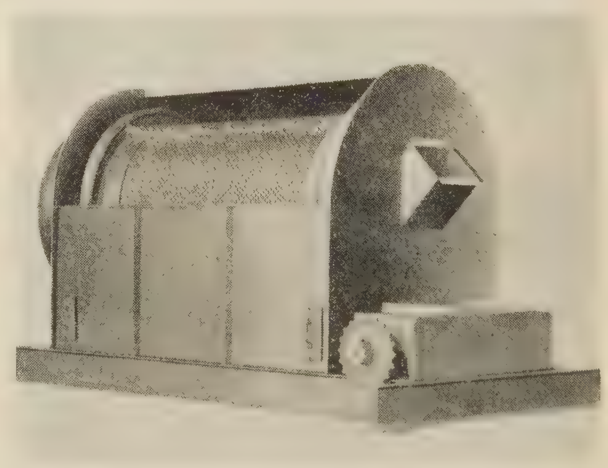

Fig. 15 View of Rotary Breaker Screen

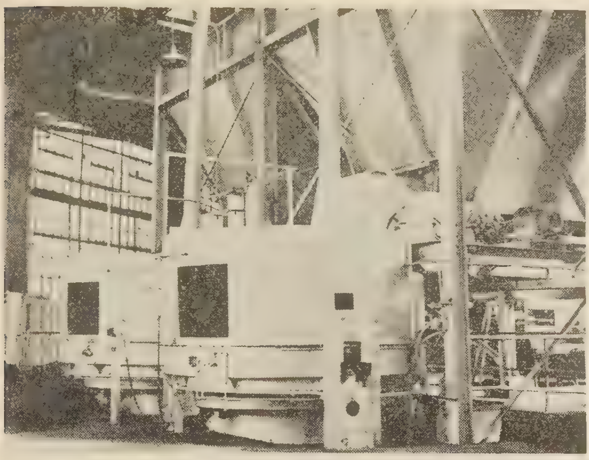

Fig. 17 Storage Bins With Apron Feeders and Batch Mixers

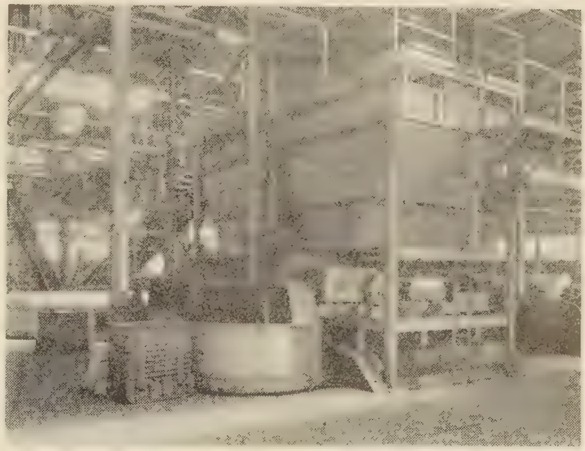

Fig. 20 Apron Feeders for Batch Mixing

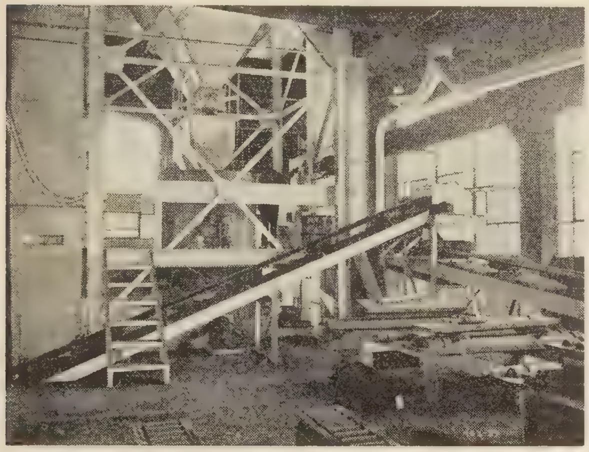

Fig. 18 Storage Bin With Measuring Hopper and Batch Mixer

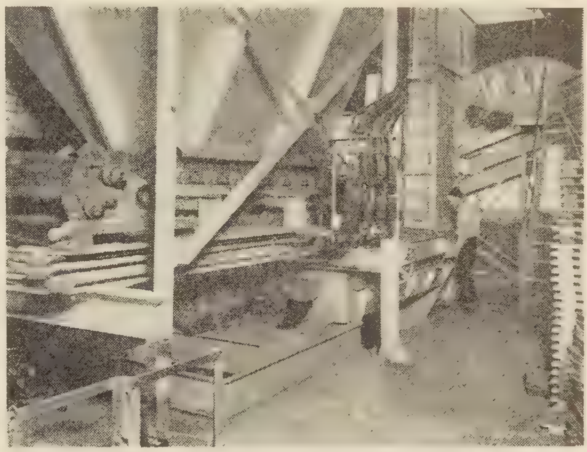

Fig. 21 Apron Feeder for Continuous-Type Paddle Mixer

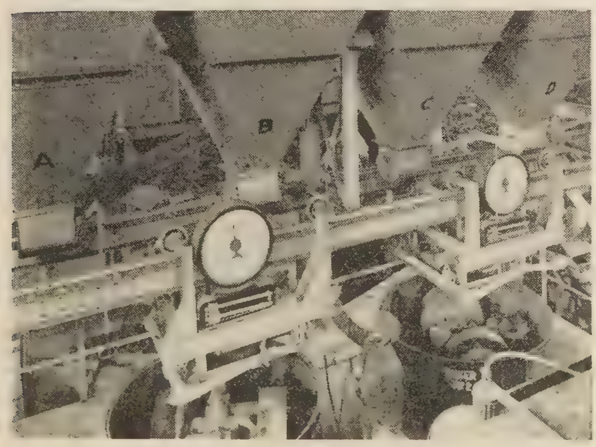

Fig. 19 Traveling Weigh Hoppers for Batching

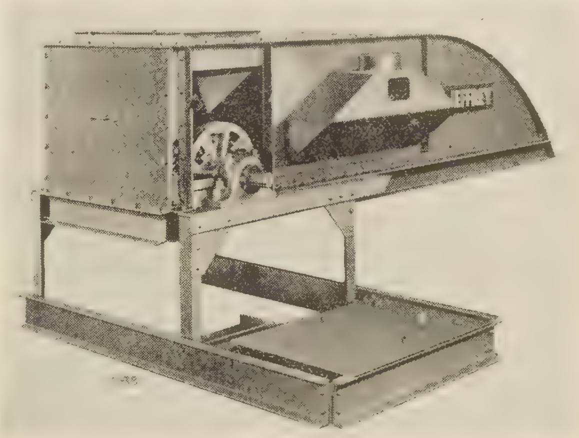

Fig. 22 Bar-Type Sand Aerator

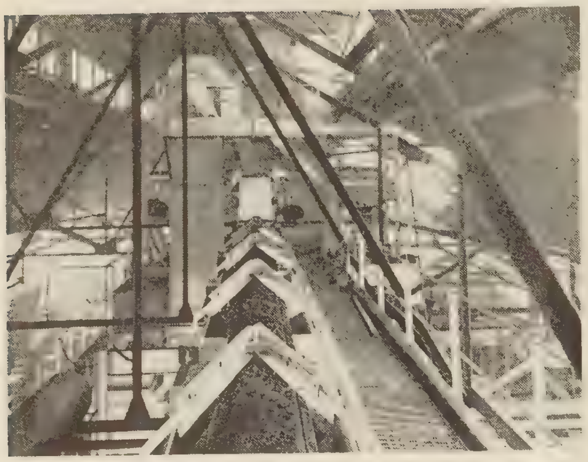

Fig. 23 Aerator Installed Between Prepared-Sand Elevator and Distributing Belt Conveyer

Fig. 26 Molders' Hoppers With Air-Operated Valves

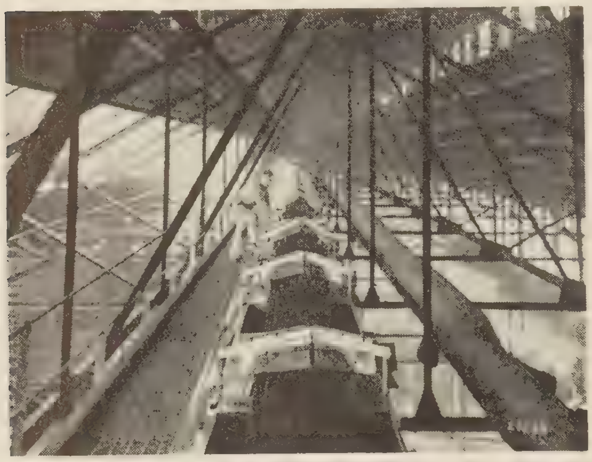

Fig. 24 Flat-Belt Distributing Conveyer

Fig. 27 Molders' Hopper With Apron Feeder and Chute to Flasks on Large Molding Machine

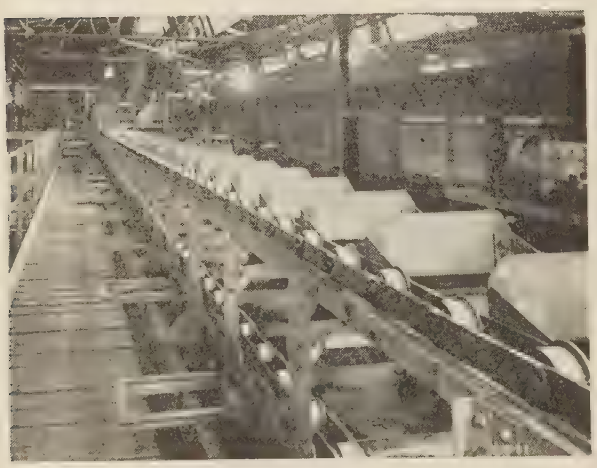

Fig. 25 Scraper-Type Distributing Conveyer

Fig. 28 Molders' Hopper With Apron Feeder and Swivel Belt for Floor Molding

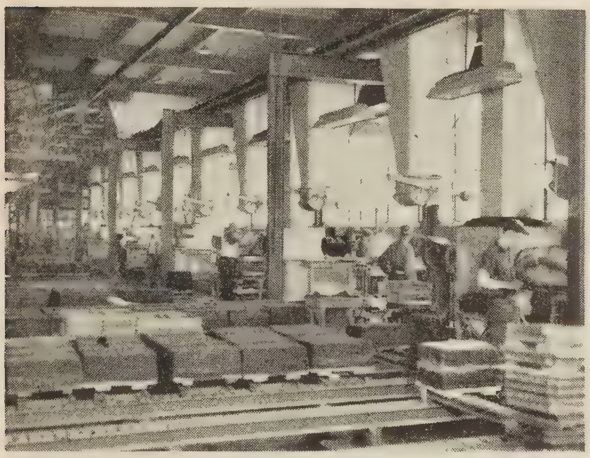

FIG. 29 SHOVEL-OUT HOPPERS FOR FACING SAND SERVING MOLDER ON EITHER SIDE

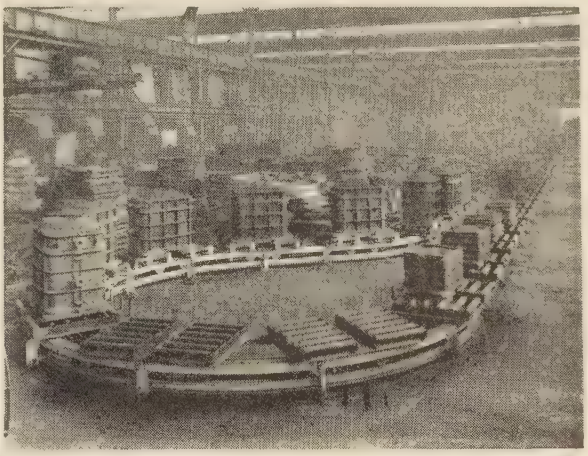

FIG. 31 CONTINUOUS-TYPE MOLD CONVEYER USING ROLLER CONVEYER TOPS

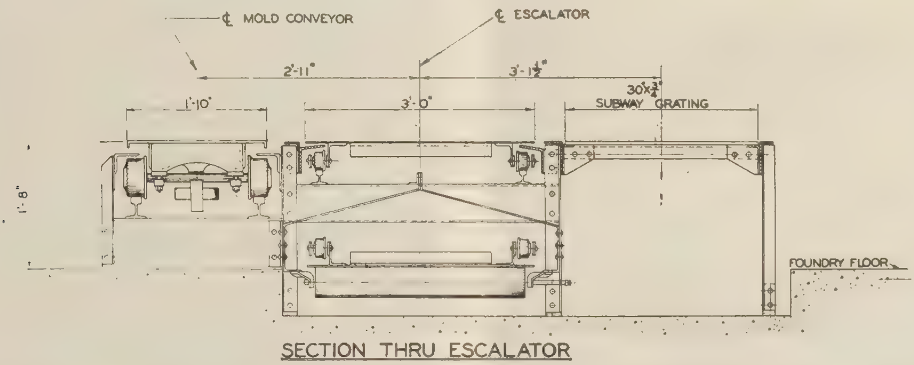

FIG. 30 CROSS SECTION THROUGH MOLD CONVEYER AND POURING PLATFORM

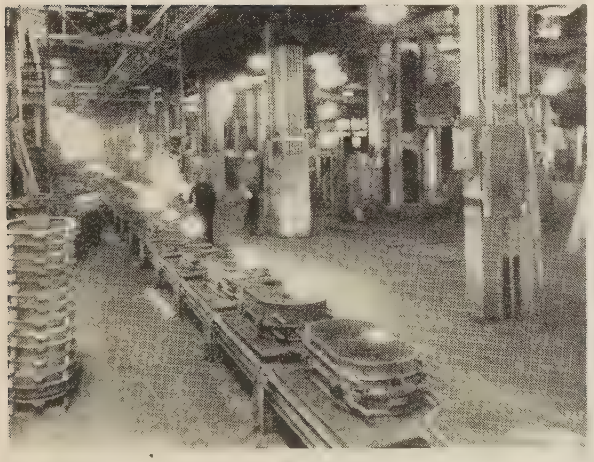

FIG. 32 CONTINUOUS-TYPE MOLD CONVEYER AT POURING STATION

FIG. 33 MOLD STORAGE USING ROLLER CONVEYERS

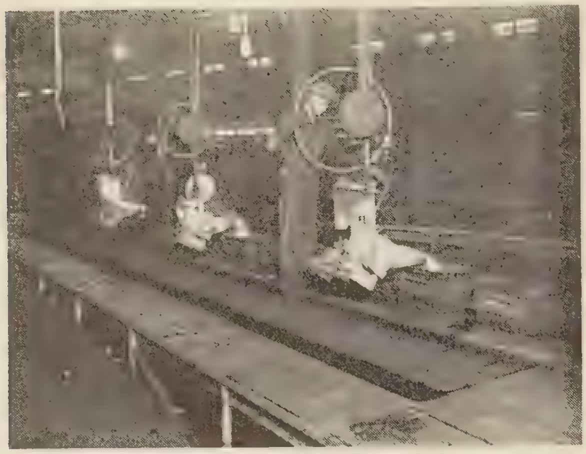

Fig. 34 Pouring Conveyer for Continuous-Type Mold Conveyer

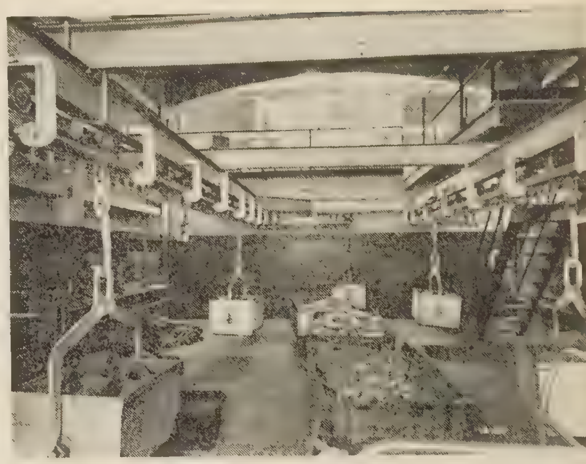

Fig. 36 Trolley-Type Castings Conveyer

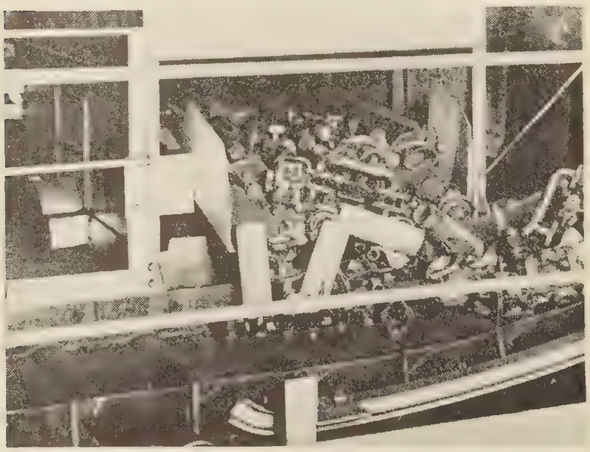

Fig. 35 Castings Discharging From Shakeout to Apron-Type Castings Conveyer

Fig. 37 Roller Conveyer Used for Flask Return

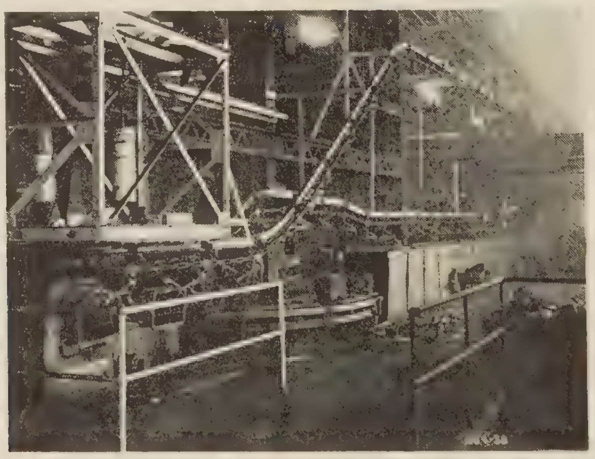

Fig. 38 Trolley-Type Flask-Return Conveyer

When facing sand is handled over the conveyer system, hoppers are provided on the floor, with shovelout openings. These hoppers are filled by separate plows and chutes from the distributing belt. In some few cases facing sand is stored in hoppers with feeders adjacent to the backing sand hopper, Fig. 29.

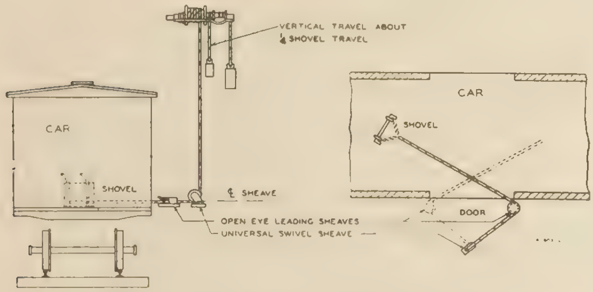

Fig. 39 Arrangement for Power Shovel

Fig. 40 Unloading Sand From Boxcar With Power Shovel

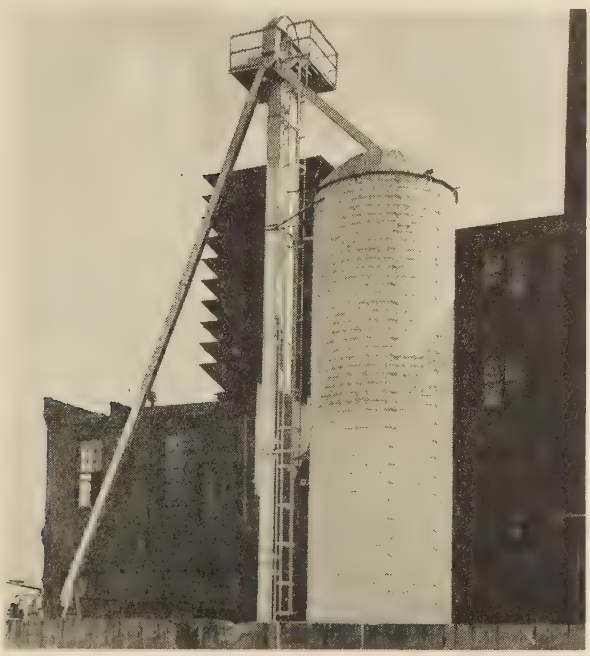

Fig. 41 Concrete Stave-Type Silo for Bulk Storage

Mold Conveyers. Layouts for continuous pouring will ordinarily utilize continuous-type mold conveyers for receiving the molds at the molding stations and conveying them through the pouring stations to the shakeout. They must be designed so that sufficient cooling time is available from the pouring station to the shakeout. This may vary from as little as 5 min. for very light castings to $^1/_2$ to 1 hr for larger molds. Air-cylinder pushers are frequently used to push the mold onto the shakeout. The use of a mold conveyer permits the molder to devote practically all of his time to molding, instead of using part of it for carrying out molds to the floor, Figs. 30, 31, and 32.

Where the pouring is not continuous, roller-type-conveyer layouts are used for handling the molds from the molders to the pouring station. This type of layout is also used where large numbers of alloys are encountered, so that the various types can be segregated for pouring, Fig. 33.

Pouring Conveyers. When continuous-mold conveyers are used and the speed exceeds about 15 fpm, moving pouring platforms are used. These operate in time with the mold conveyer so that the men pouring the metal can stand on this moving platform without any relative motion between the mold being poured and the ladle. These are apron-type conveyers with flat tops, Fig. 34.

Hot-Castings Conveyers. From the shakeout, it is necessary to take away the castings, and, in the case of smaller castings, this is frequently done with an apron-type hot-castings conveyer, Fig. 35. When used for this service, the apron should be of the leakproof type so that sand from the cores, etc., will not be spilled, but will discharge over the head with the castings, at one point. For heavier castings, such as cylinder blocks, the overhead-trolley-type conveyer is frequently used, Fig. 36, the castings being hooked on the trolley conveyer at the shakeout and frequently carried a considerable distance for cooling purposes.

Flask-Return Conveyers. Flasks must also be returned from the shakeout to the molders, and, in the case of smaller work, either the roller-type conveyer, Fig. 37, or the overhead-trolley conveyer, Fig. 38, may be used.

Sand Reclamation. Sand-reclamation systems have been introduced quite recently, and at the present time are occupying a prominent place in the plans for foundry improvements. As the name implies, they are systems for reclaiming sand which has outlived its usefulness and by special cleaning methods is returned to the foundry. The finished product compares very favorably with new sand, at considerably less cost. It also eliminates one of the present foundry problems, i.e., what to do with the spent sand.

Miscellaneous Conveyers. In modern installations, there is an increasing use of mechanical sand-unloading equipment. When dry sand is used, coming into the plant in boxcars, the power shovel has frequent application, Figs. 39 and 40. This power shovel may discharge into the bucket elevator, which will elevate the sand to the raw-storage bins. When the sand comes in hopper-bottom cars, it is common to use a track hopper with apron feeder to the elevating unit.

Concrete silos, either of the stave or monolithic type, are coming into wide usage for the storing of foundry sands, Fig. 41.

Several types of skip hoists have been used successfully for charging the cupolas in iron foundries.

Some of the larger steel foundries have used car hauls for handling the charging buggies to and from the open-hearth furnaces.

Coke and limestone have been handled from railroad car through track hopper, feeder, elevator, and belt conveyers to bulk storage.

The handling of bonding materials covers a wide range, going

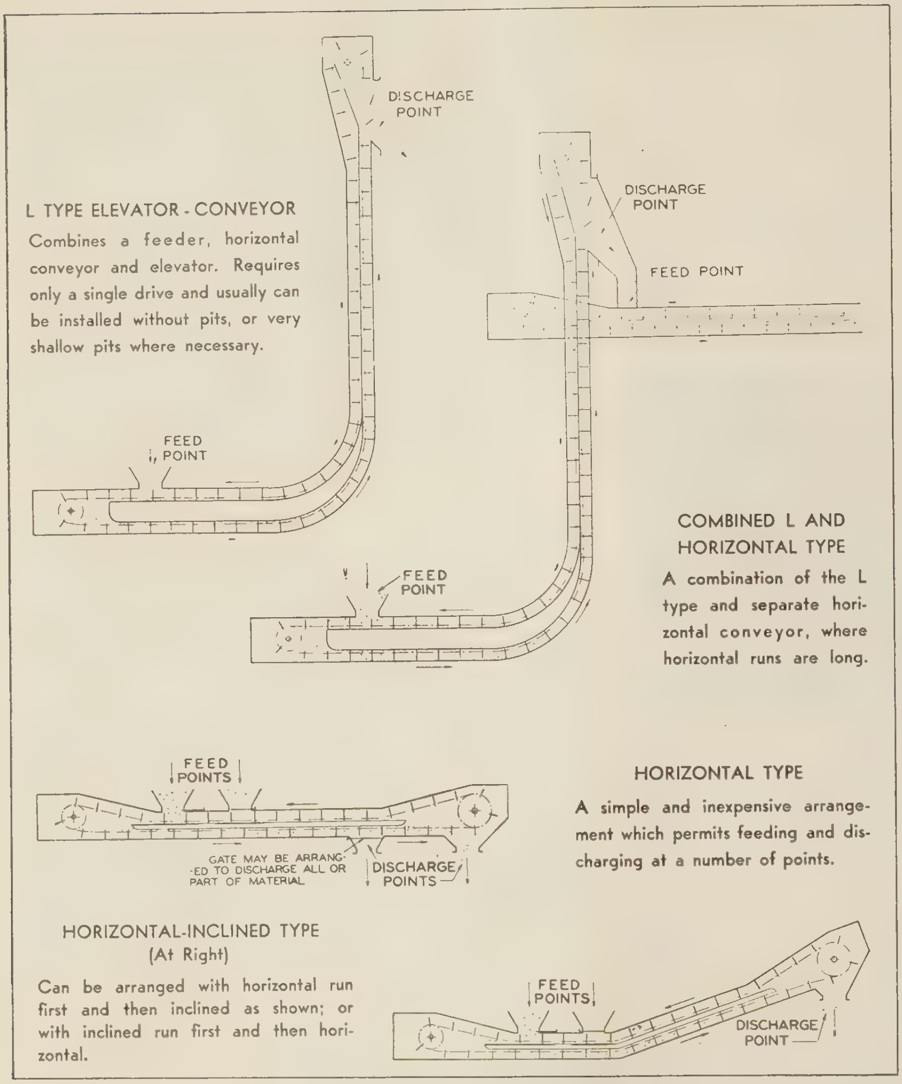

Fig. 42 Bulk-Handling Conveyers for Bonding Material

from entirely manual operations to complete mechanization. They can be successfully handled from railroad cars through small track hoppers and bulk-handling conveyers, Fig. 42, to storage silos. From the silos, they are handled through the same type conveyer or screw conveyers to storage hoppers at the mixing stations. Small screw or electric-vibrating feeders can be

(Continued at top of right-hand column)

timed to feed out prearranged proportions direct to the mixers on continuous types, or into weigh hoppers on batching types of mixers.

Several types of pig-mold conveyers have been developed for the nonferrous foundries for pigging the virgin metal and also the remelt, Fig. 43.

Conclusion

In conclusion, it is quite evident that conveying equipment for the foundry, regardless of the type, has been developed to a high degree, with individual units designed for particular requirements. The proper layout and selection of equipment results in material reduction in the pound price of castings produced. This is accomplished in several ways, i.e., by increasing the output of the molders, reduction of labor, greatly increasing production with no increase in building space, reduction of scrap losses through better control of sand characteristics, and providing better working conditions for employees. Conditions caused by the war have influenced the installations in recent years. A manpower shortage has required that machinery be employed to the greatest possible degree. This has resulted in better working conditions, which have reacted favorably to the mechanized foundry. Foundries so equipped have a material advantage in the competitive market which will probably develop after the war.

Fig. 43 Pig-Mold Conveyer for Magnesium, Using Apron Conveyer to Elevate Pigs to Boxes

Discussion

W. S. Giele.[2] Materials handling in foundry operations is characterized, first, by its relative magnitude. Conditions of foundry operation vary so widely that exact data on any particular operation would have little application in other operations.

It may be said, however, that the total weight of material passing over the shakeout may easily be 20 times the net weight of castings finally shipped. In other words, the shipment of each 100 lb of castings may involve the handling of a ton of material at the shakeout, and every pound of that ton will have been handled several times before it gets to the shakeout.

Materials handling in foundry operations is further characterized by the abrasive nature of the materials handled and by the high temperatures frequently encountered.

The authors have referred to the application of high-pressure-type lubrication to belt-conveyer idlers in order that new lubricant may force out the old and the abrasive grit with it. Where feasible, it is believed to be desirable to go a step further by installing a centralized lubrication system, piped to the various bearings, in order to eliminate the introduction of the abrasive dirt that accumulates on the high-pressure fittings.

It is, of course, advantageous to keep speeds low and to operate with a minimum of lubrication. Ball and roller bearings in conveyer rolls often last longer when run dry than when lubricated, in foundry applications. Impregnated bushings of wood or of metal often give good service in the foundry. Powdered graphite carried into a bearing by some volatile liquid like kerosene gives good service in the wheel bearings of mold-oven cars and in car-hearth annealing furnaces. Where plain bearings are used it is often advantageous to use a wear sleeve on the shaft.

Materials handling in foundry operations is characterized also by wide load surges and by the frequent introduction of foreign material; and equipment must be designed with ample overload protection and surge capacity at every point.

The authors refer specifically to the use of shear-pin hubs on apron-feeder drives. The use of this device or its equivalent is recommended on all drives where an overload might cause serious damage to equipment. It is desirable, also, so to connect the drive that the motor stops automatically when the pin shears.

In any system including a number of units, each feeding the next in sequence, it is important to interlock the electrical controls so that the units will start or stop only in the proper sequence. Pilot lights, to indicate whether or not units not visible from the operating station are running, are desirable.

It is well, also, in a system of this sort to make each unit of a capacity slightly larger than the capacity of the unit which feeds it to make sure that there will be no flooding.

Bucket elevators should have provision to prevent back-tracking in the event of power failure. Self-energizing friction brakes are often used. Positive ratchets are better. They may be of the silent type. Bucket elevators are not well adapted to handling facing sands. An inclined belt conveyer or a skip hoist is preferable for this service.

The authors make a good point in recommending storage bins of large capacity. While the 2-hr supply mentioned in the paper is adequate for continuous systems, as there stated, a much larger capacity is required to absorb the wide surges common in intermittent operation. A 10-hr supply is none too much for a foundry which pours and shakes out all its molds on a Saturday night and resumes operation on the following Monday morning.

[2] Walter Giele Company, Lebanon, Pa.

The authors develop some very good points on bin design. The writer would add only that, if molding sand did not have the property of bridging over an opening, it would never be possible to lift a cope. Vibrators will often relieve a bridging bin. Volumetric or gravimetric hoppers will deliver more nearly uniform batches if provided with vibrators to insure complete discharge at each cycle.

Vibrators are frequently of advantage on molders' hoppers, particularly on facing sand in steel foundries. Where facing hoppers are supplied with shovelout openings, it is possible to make them larger at the bottom than at the top to minimize the tendency to bridge.

Bins with downwardly converging sides opposite each other, particularly when in the shape of inverted cones or pyramids, have a marked tendency to bridge when used for prepared or other damp sand. A vertical partition will often reduce this tendency.

Roller conveyers work to good advantage wherever it is desirable to transfer loads from one conveyer system to another by means of cranes or tramrails. Roller-conveyer systems also work well with high-lift fork trucks. Roller-conveyer lines may be double-checked to segregate finished work from unfinished or to permit return of empty containers, as between a molding station and a pouring station.

Trolley conveyers are adaptable to intermittent operation on miscellaneous jobbing work as well as to continuous mass production.

Among their peculiar advantages are the following:

1 They can be made to travel in any path and will follow vertical as well as horizontal curves. They can be made to go around or over or under obstructions and to serve one or more floor levels.

2 They do not obstruct the floor.

3 The wheel bearings and track surfaces are above the dirt level.

4 They consume very little power.

5 They are endless and will return empty containers.

6 They will assemble components from a series of stations to a single assembly point or distribute to a series of stations. For illustration, they will collect cores from a series of core-making stations, bring them all to one spray station, and then distribute them to racks according to size.

7 They will carry work through process as through a spray booth, a dipping tank, or a drying oven.

8 They will accommodate any time interval between operations by changing length while keeping speed constant, as is done in castings-cooling conveyers.

Authors' Closure

Mr. Giele has brought out several interesting points in his discussion.

Regarding the use of centralized lubrication systems in connection with sand-handling machinery, while this may be desirable, it is hardly practical in the case of belt conveyers, elevators, etc., as this would involve piping the grease over very great distances. In many cases belt conveyers may be several hundred feet long.

We are very much interested in his comments on the running of ball and roller bearings dry. In foundry mold conveyers, which run at a very slow rate of speed, we have found that by running the roller bearings in the wheels dry their life is considerably lengthened. Only a small amount of light oil is added, to prevent the bearings from corroding.

On the subject of bucket elevators, back-stops are certainly desirable. We use the self-energized band brake type. Inclined belt conveyers are certainly preferable in handling facing sands, but in many instances bucket elevators are used successfully when

space limitations are such that inclined belts cannot be used.

Regarding the use of vibrators on storage bins, these are sometimes necessary to relieve bridging. However, if the opening or gate at the discharge point of the bin is not open, the vibrator may tend to pack the sand. The vibrators should be operated only when the discharge of the bin is open.

Air jets have been used in large storage bins for the purpose of relieving sticky and bridging action.

The Theory and Design of Electronic-Control Apparatus

By W. D. COCKRELL,[1] SCHENECTADY, N. Y.

Since the mechanical engineer is vitally concerned with control mechanisms involving electronics in some form, the author believes the explanation of theory involved and design of electronic apparatus given in this paper should be helpful. The almost infinitely fast and inertialess electronic tube can supplement mechanical control to produce many results hitherto considered impossible. After outlining the fundamental conceptions of the electronic tube, describing the types of tubes used, and explaining something of the operation of individual control circuits, the author indicates practical applications of electronics to standard control circuits. Examples of such controls are the photoelectric relay, electronic timer, motor control, resistance welding, photoelectric register control, and others.

Introduction

THE mechanical engineer has a larger stake in electronic control, particularly in photoelectric control, than many realize. In this new tool he has a means of supplementing his own mechanical operations in many ways. The almost infinitely fast and inertialess electronic tubes can supplement his mechanical control to produce many results hitherto considered impossible. The beam of light, or the small voltage, or change in magnetic or electrostatic field, produced by mechanical or chemical means, may act as a lever, pushrod, cam follower, or trip latch. On the other hand, since most electronic control must eventually end in mechanical motion, the electronic engineer must depend upon the mechanical engineer for assistance in solving problems of vibration, high-speed operation, and hunting or stabilization problems. No complete apparatus can be considered satisfactory unless it is correct both electronically and mechanically.

Electronics

Electronics has been defined as the art of working with an electric charge or current in the form of a stream of electrons or ions, free from the restraint of conventional conductors. As such, electronics includes the high-frequency phenomena of radio and television. However, in this paper we will simplify considerably the theoretical background to be covered. The theory required for industrial electronics, for instance, compares with that required for short-wave radio as the gravitational theory of Isaac Newton compares with that of Dr. Einstein.

Speed of Electrons and Electronic Radiation. It is sometimes difficult for one accustomed to dealing with mechanical things to realize the terrific speed at which electronic equipment can function. Airplane designers worry when the speed of their plane approaches that of sound, about 1100 fps. The propagation of electronic radiation is that of light, 186,000 mps. In a vacuum, an electron is accelerated by a difference of potential of 1 volt to a speed of 386 mps. Compare this with the pull of gravity which produces a speed of only 32 fps in the first second.

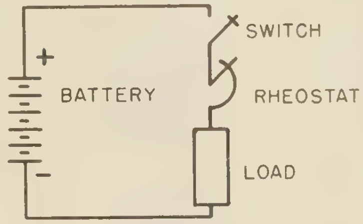

Fig. 1 A Direct-Current Electric Circuit With Rheostat Control
(Conventional current flow is clockwise; electron flow is counterclockwise.)

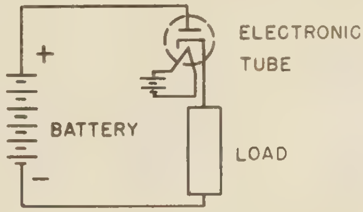

Fig. 2 A Direct-Current Circuit With Electron-Tube Control
(Electron flow is counterclockwise.)

Nature of the Electric Current. Modern electrical theory considers the current as a flow of electrons through the circuit. These electrons, the ultimate particles of negative electricity, are drawn to a potential more positive, just as a ball rolls down hill under the pull of gravity. To control a current is to control this flow of electrons. Those materials which permit a free flow of electrons are called "conductors," and those which resist the electron movement are called "insulators" or "dielectrics."

Control of Electric Current. The usual method of controlling an electric current is to change the resistance of a part of the circuit, or to replace part of the conducting path by an insulator, such as by opening a switch, Fig. 1. Another way is to control the electron flow by means requiring no mechanical movement; this is the electronic way, Fig. 2.

The Electronic Tube. The electronic switch consists of a device which may be inserted in the conductor path, and through which we may control the flow of electrons at will. Fundamentally it must consist of two parts, i.e., a source of electrons or emitter, called the "cathode," and an element to receive the electrons, called an "anode." To permit the best movement of the electrons, these elements are enclosed in an evacuated envelope.

Types of Emission. Electrons may be made available at the surface of the cathode in one of three ways. The electrons may be drawn out of the cold cathode by a positive charge placed very close to the surface. This method of obtaining electrons is used in cold-cathode voltage-regulator tubes and in ignitrons. A second method is the application of heat to special cathode

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[1] Electronics Application Engineer, General Electric Company.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

materials so that the electrons are literally boiled free from the cathode surface. This is the customary method. Finally, some cathode surfaces are sensitive to the action of light and liberate electrons when struck by light of various colors. This includes both visible light, that beyond the red (the infrared), and the waves too short to be seen by the human eye (the ultraviolet). By these three methods electrons are liberated, which may be drawn by a positive potential in a steady stream of negative electric current to the anode, Fig. 3.

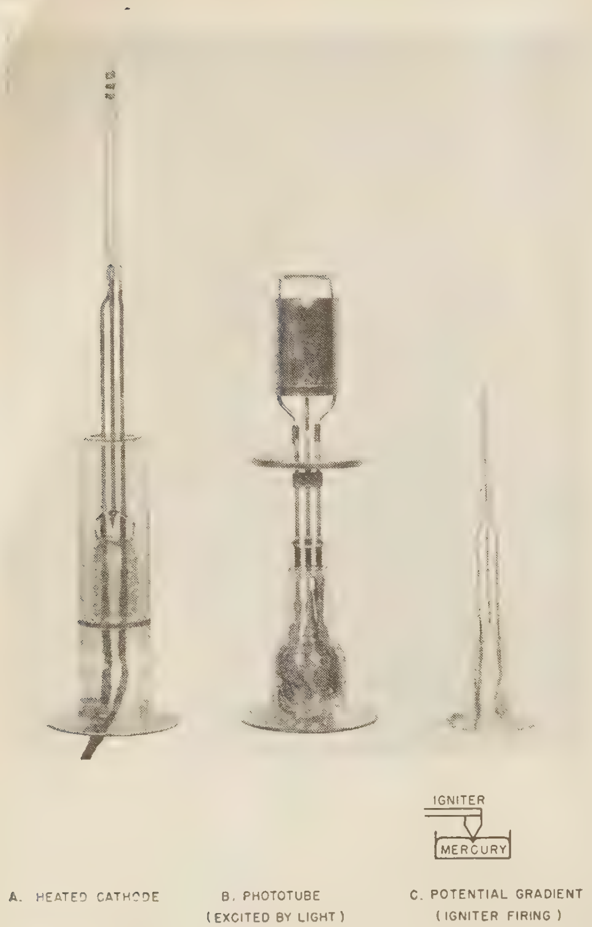
Control of Electron Stream. The control of the electron stream may be accomplished in three different ways, or by a combination of them. The most obvious way is to change the amount of energy for liberating the electrons. Thus if we remove the light from a photoelectric surface, or if we decrease or stop the heat from the thermionic cathode, we will reduce or stop the number of electrons available for the stream. These methods are shown in Fig. 4.

Since the stream of electrons represents an electric current, it may be acted upon by a magnetic field just as the current in a motor armature reacts to the magnetic field of the motor. This magnetic control of the stream has been used to a certain extent for television picture tubes, for the generation of high-frequency oscillations, and for a few other purposes, but has not been utilized to the extent to which the third form of control has been exploited.

The third, and most important, form of control is that of electrostatic deflection. This means simply that, if we insert a third element in the tube between the anode and cathode, and if this element is made negative, it will repel or deflect the electron stream away from it, and if positive, will assist in accelerating the electrons. This control element may consist simply of a plate, which will bend the stream of electrons toward or away from itself and thus change the point at which they strike the anode. This has a limited use in the cathode-ray oscillograph and in indicator tubes. A much more important effect, however, is obained by making this element a grid, or series of parallel wires, through which the electron stream must pass on its way to the anode. If this grid is made sufficiently negative, the electron stream is throttled down or even stopped entirely, and no current can flow between the cathode and anode, Fig. 5. A very important item to be noted in this form of control is that the grid, being negative, attracts no electrons, and hence there is no current flowing to the grid. Electric power consists of the product of current and voltage. Since there is no current there is, theoretically, no work done here so that the electron stream can be controlled with the expenditure of a minimum of power.

Sometimes more than one grid is used in a tube, Fig. 6. If positive with respect to the cathode, these other grids will assist in speeding up the electrons past the first grid. If negative, they will reduce or stop the flow of electrons in the same manner as a series of valves in a pipe.

We have now done two things. We have reduced the electric current to a stream of pure electrons, and we have found means



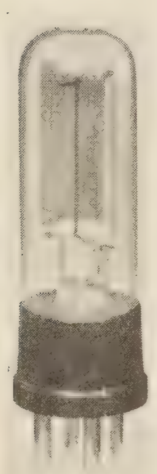


Fig. 3 Methods of Obtaining Electron Emission From a Cathode

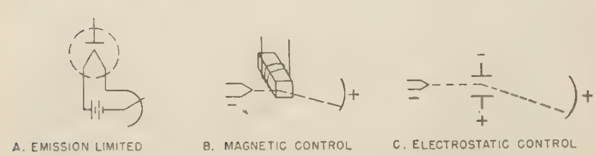


Fig. 4 Methods of Control of Electron Stream

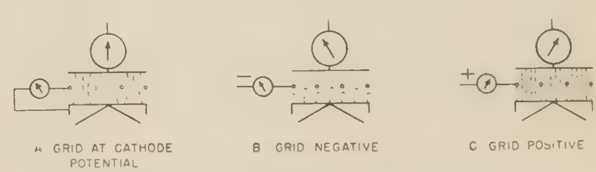


Fig. 5 Grid Control of Electron Stream

(The grid is placed in the electron stream between the cathode and anode. When the grid is at the same potential as the cathode, it affects the flow of electrons very little. When the grid is made more negative than the cathode, it repulses the electrons, and thus decreases the flow from cathode to anode. When the grid is positive with respect to the cathode, it increases the flow of electrons, but also attracts a flow of electrons to it.)

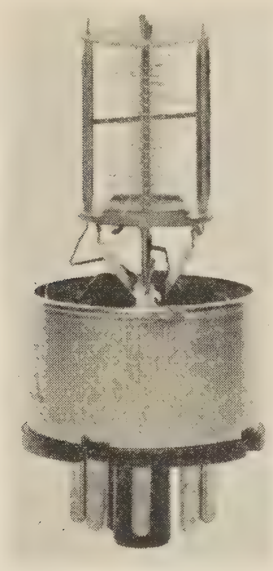


Fig. 6 A Pentode, 3-Grid, Amplifier Vacuum Tube With Glass Envelope Removed

to control this stream with a minimum amount of power. Finally, since the speed of the electrons between cathode and anode is so great, we can, by changing potential of the grid fast enough, start, stop, and control the flow of electrons at a much more rapid rate than would ever be required for use with mechanical devices.

Uses of High-Vacuum Electron Tubes

We have described the electron tube as a valve, but since we have made only one of the elements capable of emitting the electrons, current can flow in but one direction, so it is a one-way valve. Some functions make use of the one-way effect, others use the throttling effect, while a few use both effects.

Rectifiers. The one-way action may be used like a mechanical pawl-and-ratchet action or, if preferable, like a crank-and-piston action, to change the alternating current normally supplied by the power companies to direct current which is more useful for such things as operating variable-speed motors, charging batteries, and for many chemical processes. The usual alternating voltage supplied to a transformer reverses its potential 120 times per sec. If we connect our two-element electron tube in such a circuit, it will permit the electrons to flow only when the anode potential is positive. This will permit the current to flow through the load in only one direction; but, since the power supply is reversed half the time, this load current will be intermittent. This is similar in action to a one-cylinder gasoline engine and, like the gasoline engine, a large electrical flywheel is necessary to provide continuous power to the load, Fig. 7. To secure a more even flow of power, we may use a reverse transformer winding to supply a positive potential to a second rectifier during the half-cycle that the first is inoperative, Fig. 8. This, of course, provides better direct-current output with a smaller electrical flywheel. Finally, by using a number of tubes, 3, 6, or 12, and a multiphase alternating-current supply, we may produce a smooth direct-current power, just as a multicylinder gasoline engine produces a smooth flow of power with a minimum-sized flywheel, Fig. 9.

The Electronic Amplifier. A second most important use of electronic tubes is to amplify, or step up, a weak input signal to a strong output signal. This input signal must be applied to the

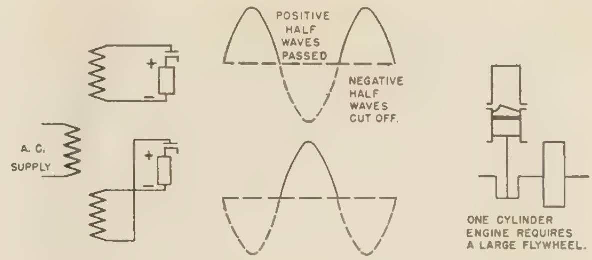


Fig. 7 Half-Wave Rectifiers Compared With a One-Cylinder Gasoline Engine

(The cathode heater circuit has been omitted for simplicity. Curves are graphs of power output against time.)

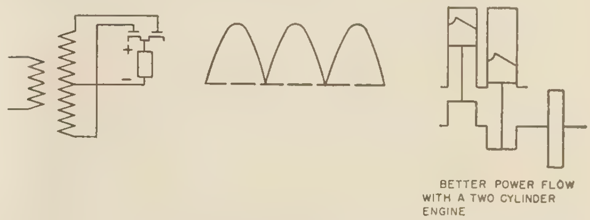


Fig. 8 A Full-Wave or Biphase, Half-Wave Rectifier

(It will be noted that this is a combination of the two half-wave rectifiers of Fig. 7. Curves are graphs of power output against time.)

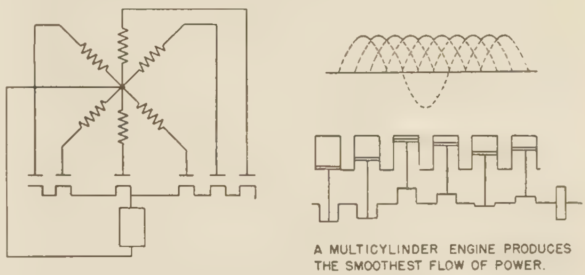


Fig. 9 A Multiphase Rectifier Produces a Smooth Direct-Current Output

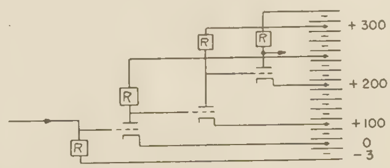


Fig. 10 A 3-Stage Vacuum-Tube Amplifier Circuit

(The current through each tube flows through the resistor R to produce a potential, which is applied to the grid of the next tube. The power is taken from a battery of 303 v, off which are tapped the required potentials for the individual tubes. Such an amplifier can step up the incoming signal many thousand times.)

grid of the amplifier tube as a voltage or potential change. If the weak signal is in the form of a small current, it must be transformed to a voltage by forcing the current through a high resistance to obtain a large potential difference across it. It is similar to obtaining a high-pressure reading from a small flow of water by forcing the water through a small pipe. This resistance may be many millions of ohms, so that even should the input signal consist of currents expressed in millionths, or even billionths, of an ampere, the voltage produced will be usable. It will be remembered that no power was required on the grid of the tube, but some power is required in this input resistor.

The change in potential on the tube grid will cause a change

in the current or electron flow between cathode and anode. If, we place in series with the anode another resistor, or impedance, through which the tube current must be forced, the potential change in this impedance will be much greater than that of the grid potential change. If this, in turn, is transferred to the grid of a second tube, and its anode current is again passed through an impedance to control the grid of a third tube, it is evident that immense amplification or magnification of the first potential change will be available at the anode of this final tube, Fig. 10. Most amplifier circuits transfer their signal between an anode and following grid through a capacitor or transformer, but the principle remains the same.

Let us pause to note that no power has been produced by the tubes. They have simply acted as valves to release a larger amount of power from some external power source, such as a battery which forces current through the tubes and their connected impedance.

Theoretically we can amplify signals thousands of times in a single tube, and, in standard industrial applications, a voltage amplification of hundreds of times, and power amplification between input and output of millions of times, is quite common.

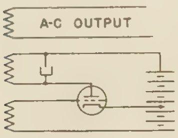


Fig. 11 A Vacuum-Tube Oscillator to Convert Direct-Current Power to Alternating Current

(The transformer at the left feeds the alternating-current power to the load and supplies a small amount of alternating-current power to control the electron-tube action. This is similar to the action of the valves and mechanism of a steam engine, which takes the constant pressure from the boiler and from it produces the necessary reciprocating action.)

Oscillators. Since the power required at the grid of the tube is only a very small part of the output-controlled power, we may take a very small amount of this output power and feed it back to control the grid at the proper frequency thus obtaining a controlled alternating-current output from the direct current supplied to the tube, Fig. 11.

Rotating equipment can generate frequencies up to 5000 or 10,000 cycles per sec. Because of the immense speed of the electrons, electronic equipment can generate frequencies up to many millions of cycles per second.

Oscillator tubes provide the type of power necessary for radio broadcasting, dielectric heating of plastics, and induction heating of small pieces of metal. However, it must be remembered that to control, by the use of high-vacuum tubes, any large amount of power, such as needed for industrial purposes, is extremely expensive, so that most electronic control of industrial power is ultimately obtained from the gas-filled tube which will now be discussed.

Gas-Filled Tubes

When in a simple two-element vacuum tube, the electrons are being drawn from cathode to anode, an interesting thing happens. Electrons move across the space because they are negative charges, being attracted by the positive anode. However, since they themselves are negative, they will tend to shield all electrons between themselves and the cathode from the action of the anode in the same manner that the negative grid does. This effect is so great that, in order to obtain the 1 amp current necessary to light a 100-w lamp, it might be necessary to apply 1000 v between anode and cathode. This means that high-vacuum tubes are most useful at low currents or with high-voltage loads.

However, if we introduce into our vacuum-tube envelope a small amount of gas, about 0.0001 of atmospheric pressure, a remarkable change in this space charge takes place. The gas may be one of the inert gases, such as Argon or Xenon, or the mercury vapor above a small drop of mercury at normal room temperature.

Ionization. If the positive potential on the anode is less than 8 v, the electrons leaving the cathode bump into the many gas molecules on its path to the anode so that their progress is slowed down very much and the current flowing is very small. However, somewhere between 8 and 12 v, depending upon the type of gas and the pressure, some electrons attain sufficient velocity so that, in striking these gas atoms, they knock an electron or two from them. Then, for each original electron, we have two free electrons moving toward the anode, and the atom, losing an electron, becomes a positive ion which drifts toward the cathode. The glancing electron may hit other atoms and knock out other electrons on its erratic course to the anode, until finally almost the whole area between cathode and anode, no matter what its shape or length, is filled with flying electrons and positive ions. Of course many atoms recombine with electrons giving up their energy in the form of the familiar glow seen in gas-filled tubes. Yet, on the whole, this so-called "plasma" is composed of approximately equal numbers of positive ions and negative electrons. Electrons are always arriving at this plasma from the cathode and are leaving at the anodes, but the plasma maintains always a potential of about 10 or 15 v positive with respect to the cathode for, if more electrons should leave than arrive, the average potential of the plasma will rise, accelerating the electrons to a higher speed and thus knocking more electrons from the neutral atoms within the plasma. If more electrons arrive than leave, the potential of the plasma will be lowered and the average potential of the arriving electrons from the cathode will be less. Thus, no matter what the current from the cathode, even though it be thousands of amperes, the potential of the plasma will change very little. Fig. 12 gives an idea of the potential gradient or voltage distribution in a vacuum tube and in a gas-filled tube.

Cathode Abuse. It must be noted that this low voltage potential of the anode for high current depends upon having a sufficient number of electrons available at the cathode for the ionization. If for any reason there are not sufficient electrons available, the plasma potential may rise above 22 v. At about this potential, the heavy positive ions are drawn toward the cathode with sufficient velocity to damage the cathode surface, and may even chip off flakes of the active material, ruining the surface within a very short time. Thus it will be understood why it is so necessary that hot-cathode tubes be permitted to warm up to obtain an efficient emitting temperature before the anode circuit is closed.

Mercury-Pool Rectifier and the Ignitron. In the mercury-pool or tank rectifier, and in the ignitron, the cathode consists of a pool of mercury. Once ionization has taken place in these tubes, the positive ion, drifting near the cathode, draws out the electron by the potential gradient effect. In this manner, a plentiful supply of electrons is assured at a potential drop between 15 and 20 v. These, then, are the tubes which are used for the high currents required for resistance welding, railroad electrification,

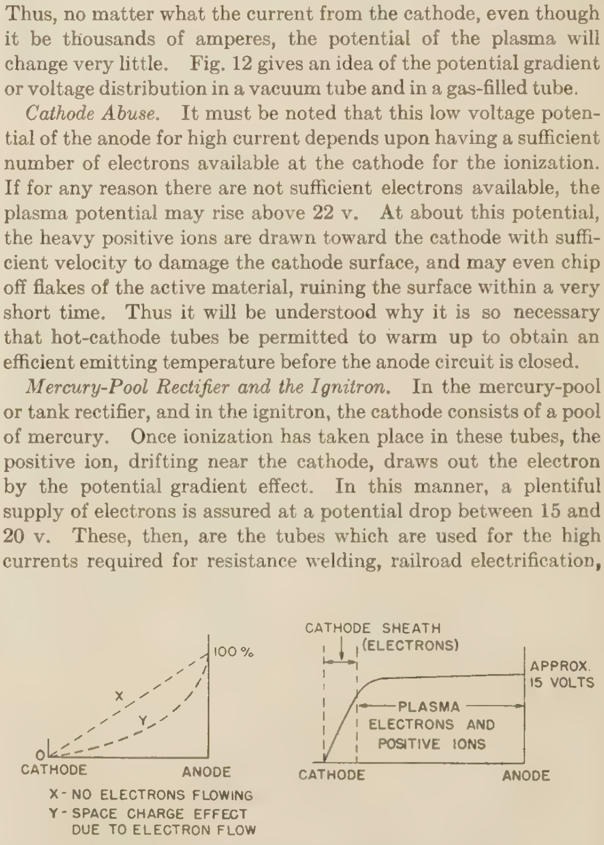


Fig. 12 Electrical Potential Gradient Between the Cathode and Anode of a Vacuum Tube and a Gas-Filled Tube

(In the vacuum tube, the anode potential may be many hundred volts. In the gas-filled tube it should never be more than 20 v.)

and electrochemical processes. Fig. 13 shows the construction of an ignitron.

Before the ionization action can take place, however, the plasma must be established. In the tank rectifier, this is done

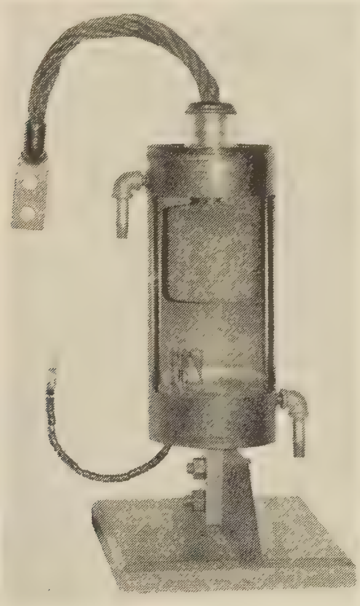


FIG. 13 CUTAWAY VIEW OF AN IGNITRON

(Normally there is a pool of mercury in the bottom of the tube, into which the igniter, shown at lower left, projects. The anode in the top of the tube is a large block of graphite, which, with its heavy stranded lead projecting from the top of the tube, is designed for the heavy current conducted by this tube. The cathode connection is the copper lug by which the tube is mounted. Inlet and outlet pipes are shown for the water, which circulates within the hollow shell of the tube for cooling.)

by mechanical means, and in the ignitron by the action of a small piece of crystal dipping into the mercury, which is called an igniter. For most precise applications the igniter is controlled by a hot-cathode thyratron.

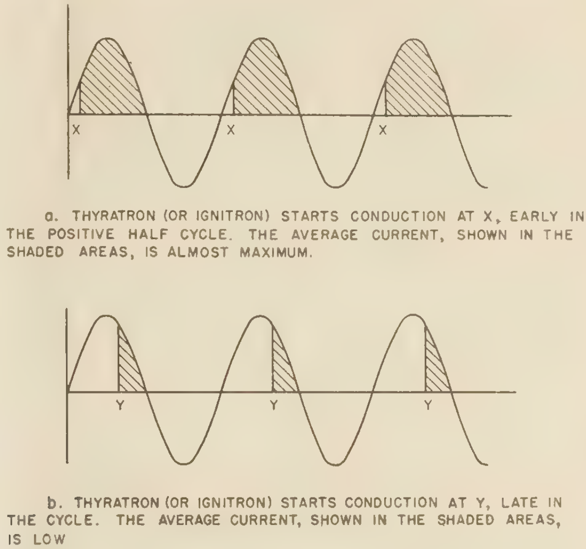


FIG. 13(*a*) OPERATION OF A THYRATRON

(The average current of a thyratron or ignitron is changed by retarding the time of beginning conduction in each cycle. These tubes may be connected in multiphase circuits also to permit a smoother flow of output power.)

Gas-Filled Tubes With Grids. Since the gas filling in the tube makes possible its use with low losses when conducting high currents, it is the one which we must use for the usual applications requiring power, such as driving motors or operating a solenoid. We wish to control these gas-filled tubes also by some form of grid, as in the case of the high-vacuum tubes. Unfortunately, however, the control cannot be attained in the same manner so that a slightly different method of control must be obtained from the grids in the gas-filled tubes.

The Hot-Cathode Thyratron. It should be fairly evident that the conventional grid, inserted in the gas-filled tube in the plasma between cathode and anode, would have little effect on the erratic and random motion of the electrons and ions filling the space. It is true that, when the grid is made quite negative before a positive potential is applied to the anode, it may keep

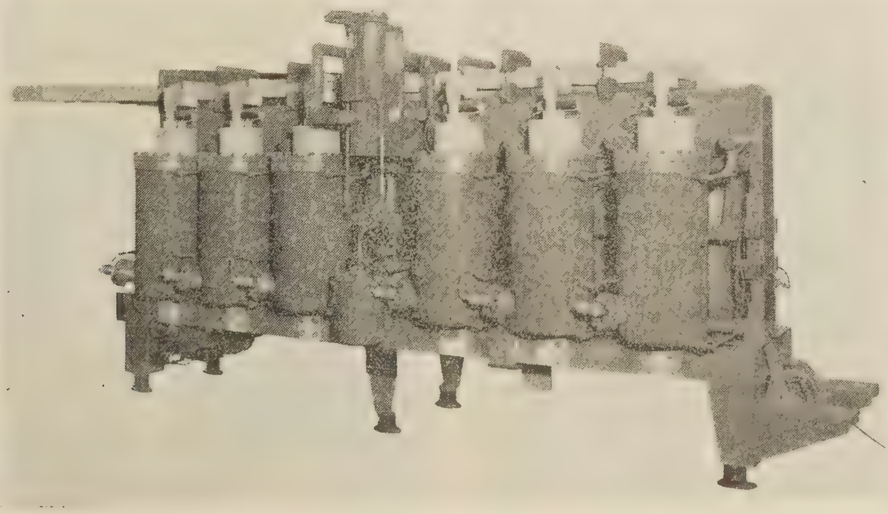


FIG. 14 A LARGE MULTIPHASE IGNITRON RECTIFIER

(Six large ignitrons are shown. They are connected in a circuit similar to that of Fig. 9, to convert alternating current into the direct current required in a large aluminum plant.)

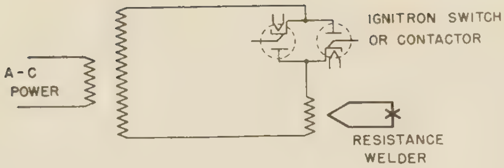


Fig. 15 Inverse-Parallel Ignitron Connection, Acting as an Alternating-Current Switch to Control a Load Such as a Resistance Welder

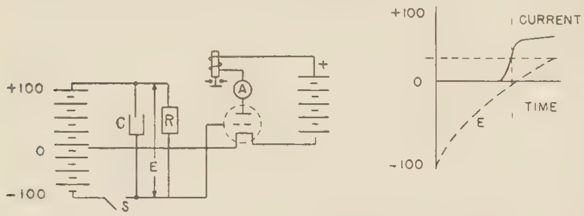


Fig. 16 Elementary Electronic-Timer Circuit

(The battery at the left charges the capicator C to a potential of 200 v. The grid of the electron tube is connected to the negative end of the capacitor, and hence is 100 v negative with respect to its cathode. Therefore no electron flow takes place in the tube. When the starting switch S is opened, the capacitor charge leaks off through the resistor R, and the grid potential gradually rises until the grid approaches the cathode potential, and electrons begin to flow in the tube, sufficient to pick up the relay and close the control circuit.)

all electrons from flowing from the cathode and thus prevent the formation of the plasma. But once the plasma has formed, the grid is powerless to stop the electron flow.

Since the grids can only prevent the conduction through the thyratron, but cannot stop it once it has begun, a question may arise as to the usefulness of this type of tube. The answer lies in its use on alternating-current power. It will be remembered from the discussion on rectifiers that, when alternating-current power is applied to the tubes, the anodes become positive one half of the time and negative for the other half. During the time that the anode is negative, no electrons will flow, and the positive ions may recombine with the electrons, or deionize, thus permitting the grid to regain control. If the grid is negative when the next half cycle begins, the electron flow cannot begin until the grid becomes positive, and thus it can determine the point in the positive half cycle where conduction starts. By retarding the grid firing angle to the desired point within this positive half cycle, one may control the *average* current passed by the thyratron and, thus, the output to the load. The same reasoning may be used concerning the action of the igniter in the ignitron, to decrease the maximum current which may be passed by that tube.

Thyratron and Ignitron Circuit. The thyratron and ignitron may be used in the rectifier circuits discussed previously, but with much higher efficiency for larger loads. In fact, multiphase ignitron rectifiers have become the basic means of obtaining direct current from alternating current in many industries. Thyratrons, so connected, supply variable power for the control of direct-current motors and similar uses, Fig. 14.

Although electronic tubes are inherently rectifiers, two of them may be connected in the "inverse-parallel" circuit, to permit passing both halves of the alternating-current wave and thus become an effective alternating-current switch which combines both high-speed switching action and throttling action. It is this type of circuit that is used to supply the precisely measured amount of power for resistance welding, Fig. 15.

Miscellaneous Control Circuits. Both thyratrons and vacuum tubes may be used in connection with the more conventional electrical components, such as capacitors and inductances, to produce a variety of functions.

If the energy is stored in a capacitor and then is permitted to be dissipated through a resistor, the capacitor potential will be reduced at a well-known and easily calculable rate. If this potential is applied to the grid of an electron tube, the anode current will be varied, or the thyratron grid will reach the firing potential in a definite time. In this way, an accurate and unvarying timing relay is developed, Fig. 16.

Combinations of resistors, capacitors, and inductances are used to obtain the grid and anode phase relations required for the proper functioning of the thyratron control circuit. Electronic tubes are used for the excitation of the fields of electric generators in many jobs in both the industrial and military fields. Thyratrons supply the direct-current power for the large saturable reactors such as used for the Radio City Music Hall lighting and for critical electric-furnace applications.

Needless to say these are only a few of the functions of the tubes and the methods by which they are controlled. There are many others, and the number will continue to grow, as long as man exercises ingenuity and the types of tubes are improved or modified to suit the new uses and demands.

Standard Electronic-Control Circuits

Now that we have the tubes and the individual control circuits available, let us begin to put them together to obtain some practical applications.

A Photoelectric Relay. Suppose we start with a simple photoelectric relay. To make this, we will take first a simple rectifier to obtain direct current from the alternating-current supply (which is usually the only kind available). At the output end of the rectifier, we will place a capacitor to store up energy between the positive pulses of the alternating-current circuit, and across this output we will also place a series of resistors which will act as a voltage divider to permit obtaining both anode and grid potential from the one direct-current source.

Next, we will add the necessary phototube to convert the change in light into a change in electric current. The drop across the phototube resistor is then applied to the grid of an amplifier tube, in order to step up the signal sufficiently to operate a small contactor placed in the amplifier-anode circuit.

As a finishing touch, we may add a rheostat so that we may vary the amount of direct current which is normally applied to the grid, in order to select the amount of light which must fall on the phototube before the grid is made sufficiently less negative to operate the contactor. As further refinements, we may add a very small capacitor between grid and plate of the amplifier tube

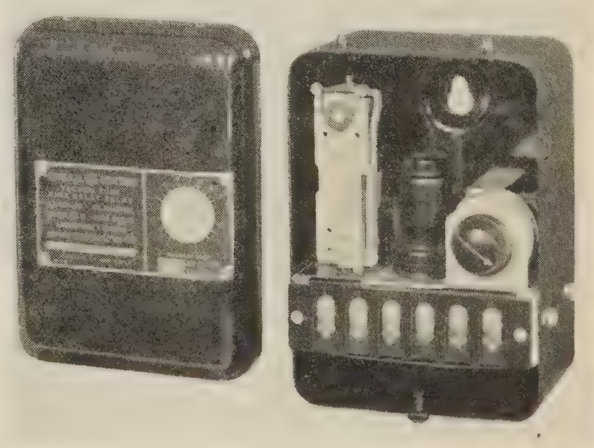


Fig. 17 Commercial Form of Electronic Timer

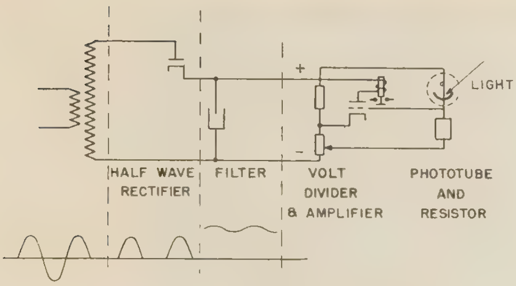


Fig. 18 Photoelectric-Relay Circuit

(This combines the half-wave rectifier of Fig. 7 and a capacitor filter which acts as a flywheel to produce a steady direct current. Next, a tapped resistor voltage divider provides the necessary potential for the electronic amplifier tube and the phototube. When the phototube is dark, no electrons flow through it or its resistor, and the electron-tube grid is far negative, so that no electrons flow in the amplifier tube. When light strikes the phototube, the phototube electrons, forced through the phototube resistor, raise the grid of the amplifier tube sufficiently to permit it to conduct and pick up its relay.)

Fig. 19 Commercial Photoelectric Relay Using Circuit of Fig. 18; Cover Thrown Up

to prevent operation on slight light flickers and another capacitor between the direct-current potential and ground to prevent action due to slight changes in the alternating-current power supply.

The circuit is shown in Fig. 18, and the relay in Fig. 19. Hundreds of these relays have been sold and are now doing all sorts of useful jobs, such as counting people or autos, packages and cabbages, and cartons freshly painted.

An Electronic Timer. In a similar manner, we may build up an electronic timer by combining the capacitor-resistor fundamental timing circuit with an amplifier and a contactor operated by the amplifier, Fig. 17.

Tube Control of Motors. A motor control, Fig. 20, which will regulate the speed of the motor quite accurately may be made by a pair of rectifiers, one or both of which may be controllable, to supply the motor armature and field of a direct-current motor. Then we may obtain an indication of the motor speed, either by connecting a tachometer generator to it which will produce a voltage proportional to speed, or take the back-induced voltage from the motor. This is compared to a reference voltage, and the difference is amplified and applied to the control of the armature or field to hold the motor speed constant. If we wish, we can add other circuits which operate from the motor current to limit this current to a safe value for motor and tube in spite of the demand for speed. A control can even be added to compensate for the resistance of the motor armature to obtain a flat compounding which is superior to that which can be obtained from a direct-current compound motor.

Resistance Welding. The electronic control of resistance welding has played a major part in the fabrication of equipment used to fight this war. The principle of resistance welding is old, that of squeezing two pieces of metal together and then passing current through the resistance of their junction sufficient to melt the material and thus cause it to fuse together. This method of fabrication is extremely fast and, in some cases, the cost is only one tenth that of riveting. However, satisfactory welding of such materials as stainless steel, which must be melted at the junction point, but must heat and cool through the critical temperature quickly to prevent loss of its stainless properties, and aluminum, which is a very good conductor of both heat and electricity and is so soft that a little too much heat will quickly ruin the weld, is made possible only by the precise timing and heat control of the electronic device. To obtain the most effective control, we start with the electronic timer, Fig. 21 (*a*), or if this is to be a seam welder which produces a series of pulses rather than a single hot spot, with two timers, one to control the heating period and the other the cooling period between the heat pulses. We may then add a heat control which determines the phase angle at which the control thryratron and ignitron will fire to produce the desired amount of heat at each pulse, Fig. 21 (*b*). Finally, we have the large ignitrons themselves which pass

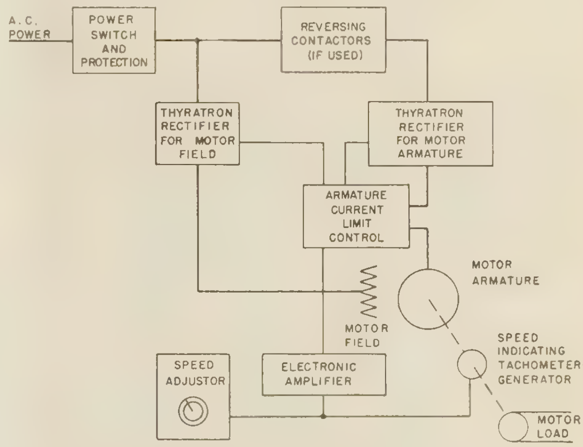


Fig. 20 Electronic Motor Control

(The indication of motor speed, taken from the tachometer-generator, is compared with the speed-adjustment setting, and the resulting signal is amplified to control the power applied to the armature and field of the motor. The current-limit feature prevents overloading motor or tubes. Speeds constant to 1 per cent from no load to full load over a speed range of better than 20 to 1 are possible.)

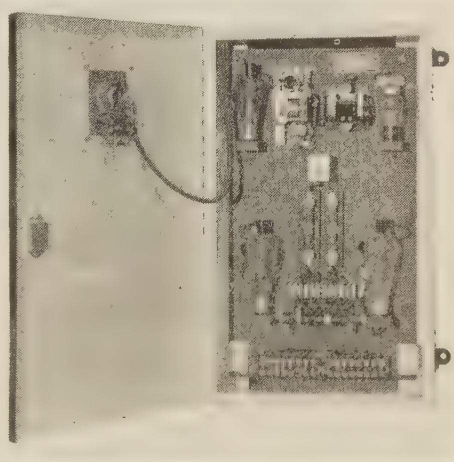


Fig. 21(*a*) Electronic Timer Controls Duration of Heat Pulse, for Resistance Welder

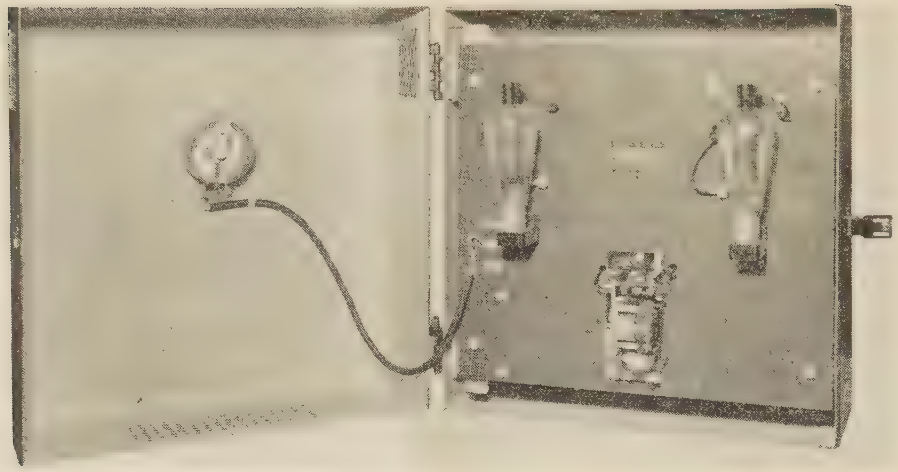


Fig. 21(*b*) Thyratron Grid-Phase Control Panel to Control Rate of Heating (Watts) During Heat Cycle of Resistance Welder

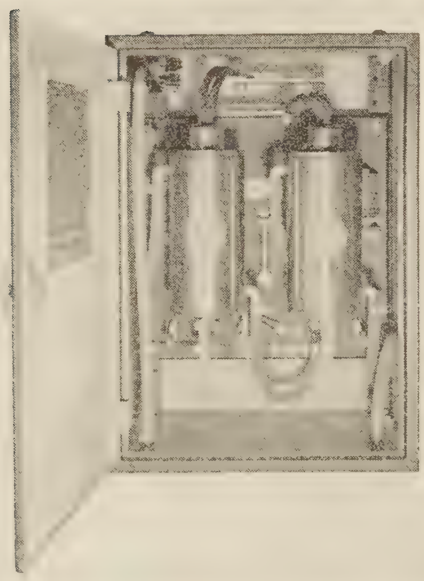


Fig. 21(*c*) Ignitron Contactor Switches on and off Resistance Welder Power

(This is the type of panel used in the circuit of Fig. 15.)

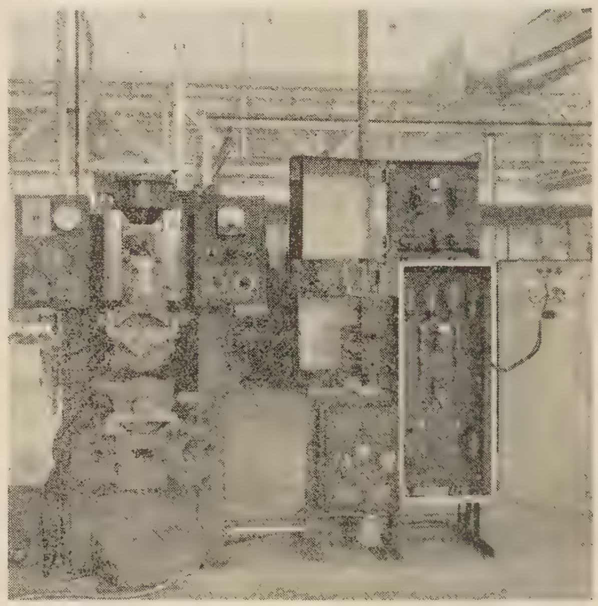


Fig. 21(*d*) This Equipment, Shown in an Actual Installation, Is the More Usual Form of Panel Combining Functions of the Three Panels Shown in Figs. 21(*a*, *b*, *c*)

the actual power current required by the welder, Fig. 21 (*c*). These large tubes are capable of handling tens of thousands of amperes for the short time required to make the weld. They are cooled by a flow of water which circulates in their double shell.

Other types of welders use rectifiers to store up the energy in a large bank of capacitors, or in an inductance, and then discharge this energy suddenly into the weld. They are thus able to supply a large amount of energy in a very short time by storing up this energy from the power line over a period of time, rather than demanding it all at once from the electrical supply.

Photoelectric Register Control. The circuits described have been more or less self-contained electronic equipments operating a particular device. In the case of the photoelectric register control, that is, the registering of printing with some mechanical operation such as cutting or folding the printed paper, there is allowed much more freedom of engineering in the over-all application. For instance, for the simplest type, say that of a candy-wrapping machine where the paper moves very slowly and may be stopped each time that a bar is to be wrapped, an index mark may be printed along the edge of the sheet so that, when it intercepts the light to the phototube, either by transmitted or reflected light, the paper will be stopped in order that the required operation may take place before the paper is permitted to move forward again. Fig. 22 shows a typical wrapping machine.

On the other hand, when the paper moves somewhat more rapidly, it may be drawn forward slightly too much, each time, and then, when the register mark reaches the limit of the permitted tolerance (as determined by the position of a

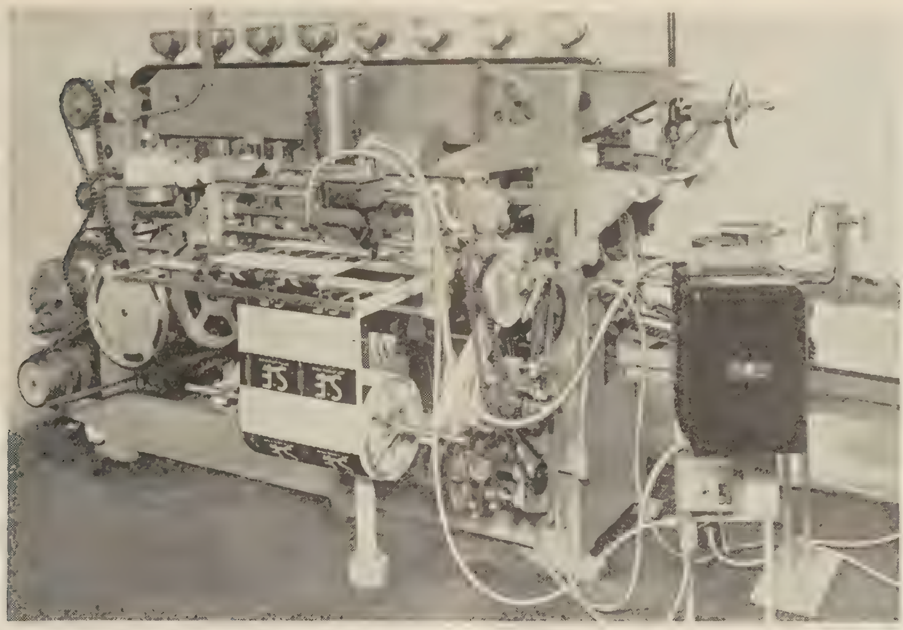


Fig. 22 Photoelectric Register Control of Label Position on Wrapping Machine

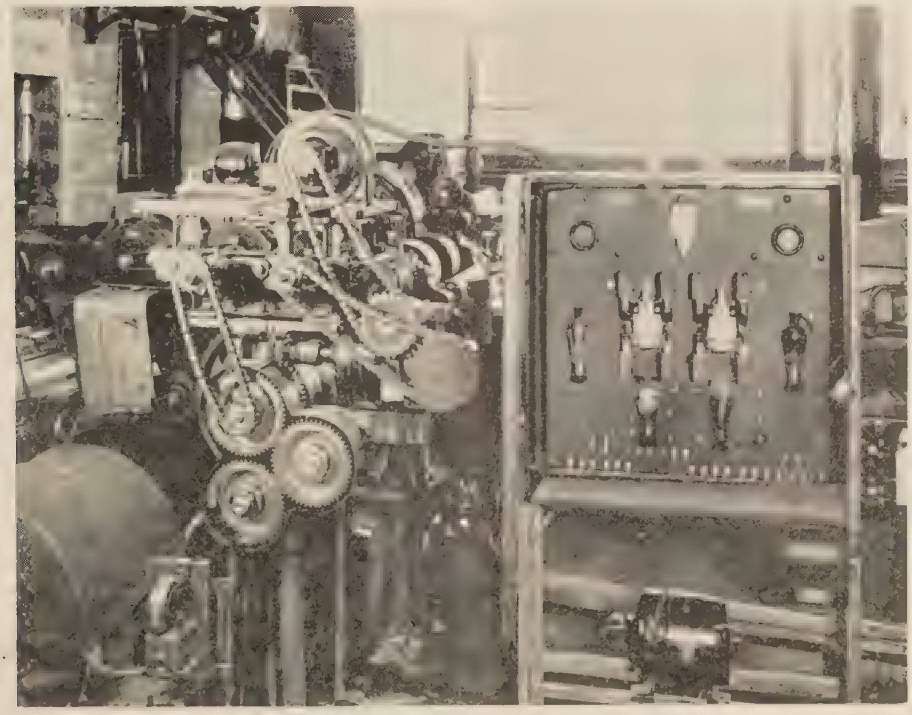


Fig. 23 One of the First Photoelectric Register-Control Equipments, 1932
(Note mass of chain and gearing.)

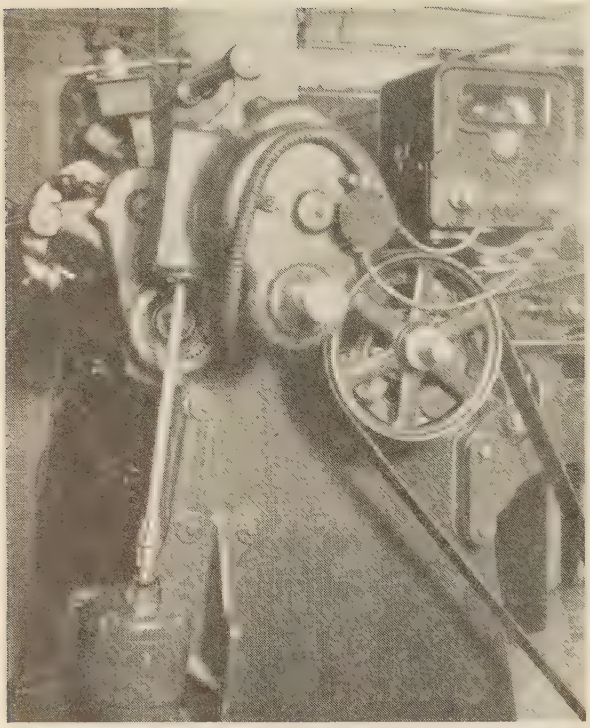


Fig. 24 (*a*) Modern Paper Cutter or "Sheeter" With Photoelectric Register Control
(One correcting motor, electronically controlled, is sufficient.)

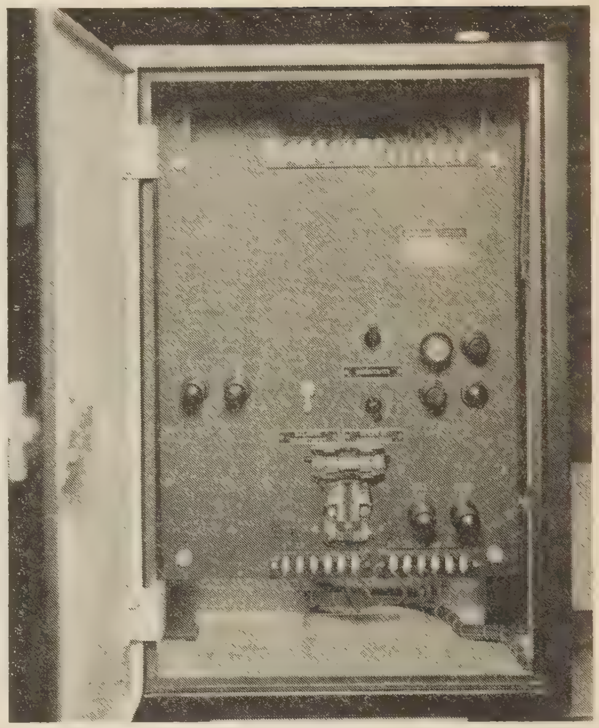


Fig. 24 (*b*) Electronic Panel for Control of Correcting Motor on Machine Shown in Fig. 24 (*a*)

contact geared to the knife), the paper may be moved back by a notching action and then permitted to creep forward again.

As speed is increased, it becomes necessary to control the speed of the web accurately so that it will always be cut near the proper point. This can be done completely by electronic means, all the way from the phototube to the thyratrons supplying power to the correcting motor driving a differential to synchronize exactly the speed of the web and cutter. For most precise action at high speeds, a combination of phototubes and a slotted disk may be used to indicate the position of the knife.

However, at any point in this complete system from the first amplifier tube and its small contactor to the end, mechanical devices may be used to replace electrical devices. It is here that the ingenuity of the mechanical engineer is necessary to determine the best combination of electrical and mechanical equipment for that particular application.

It is interesting to note that through the years sound judgment, as shown by results, has dictated the ever-increasing part to be taken by the electronic components. Fig. 23 shows one of the first installations (1932) and is typical of the large part that mechanical operation played in the older equipment. Fig. 24 (*a*) and (*b*) shows a modern equipment in which tubes do many things formerly done by mechanical means.

Application of Electronic Control

By E. H. VEDDER,[1] EAST PITTSBURGH, PA.

This paper describes typical applications of electronic control. Motor control including an adjustable-speed drive for direct-current motors and a speed-regulator system are shown. The importance of resistance welding in production of sheet-metal assemblies and the use of ignitron equipment for control of time and current are discussed. Many uses for control using light, including loop, tin-flow, and paper-cutting regulation are described, as well as pinhole detection in steel strip, and general-purpose photoelectric relays. The importance of close co-ordination of design of the electrical and mechanical elements of a complete system is emphasized.

ELECTRONIC control is now used in a variety of ways as a component part of systems which are usually partly mechanical and partly electrical. Inasmuch as many functions may be done either electrically or mechanically, careful consideration must be given to co-ordination of the two elements to produce the best system.

The first principle we need to establish is that electronics will live up to its publicity only if it is really put to work as an accepted workaday tool. If electronics is allowed to become a gadget business, we have failed to utilize the tool properly. The gadget angle can be avoided only by critical survey of every application. Electronic control has been developed which will do many things better, less expensively, faster, or more precisely, as well as many things previously impossible. But electronics is not a cure-all for every trouble. It is merely a new tool which must be properly used within its limitations. To illustrate the type of things for which electronics is suited, a few applications will be described.

Motor Control

Many motor-drive problems, particularly those connected with machine tools, are best solved by the characteristics inherent in a shunt-wound direct-current motor. The almost universal availability of alternating-current power and relative unavailability of direct-current power has made it much more difficult to use the desirable direct-current motor for these drive applications. There has also been a growing desire for a greater nicety of control than can be readily obtained even if a direct-current system is available. An electronic control has now been developed which operates a direct-current motor from an alternating-current line and has additional desirable features not previously available. The electronic adjustable-speed drive shown in Fig. 1 consists of a direct-current shunt-wound motor whose field and armature are energized by separate thyratron rectifier circuits. These rectifiers are so controlled as to vary the input to the motor, regulate the motor speed to a preset value, and limit the maximum current furnished the motor during acceleration and reversal.

The control permits smooth stepless adjustment of motor speed over a range of at least 20:1 below the base speed of the motor, and over an additional field-control range of as much as 4:1 above the base speed of the motor. This speed control is readily obtained over the full range below and above basic speed by means of a small potentiometer located in the push-button station convenient to the operator.

Combined with the wide range of speed control are the very flat speed-torque characteristics over the whole range. This is obtained by speed-regulating circuits which insure that the motor will be held to the preset speed chosen by the operator. In a properly adjusted system, the speed over a 10:1 range below basic speed will not vary more than 4 per cent from a preset value, with torque varying from no load to full load, nor will it vary more than 8 per cent for any speed within the speed range of 20:1. Normal variations in alternating-current voltage have only a small effect on the speed regulation.

It is necessary that capacitors made of foil and paper be wound with uniform tension to insure good insulation. The electronic motor-speed control is used for this purpose. The operator controls speed in this application with a foot-operated speed adjuster consisting of a variable reactor with foot treadle. Continuous speed adjustment is available from zero to maximum winding speed. Greatly increased production resulted because of the smoother speed control by the operator.

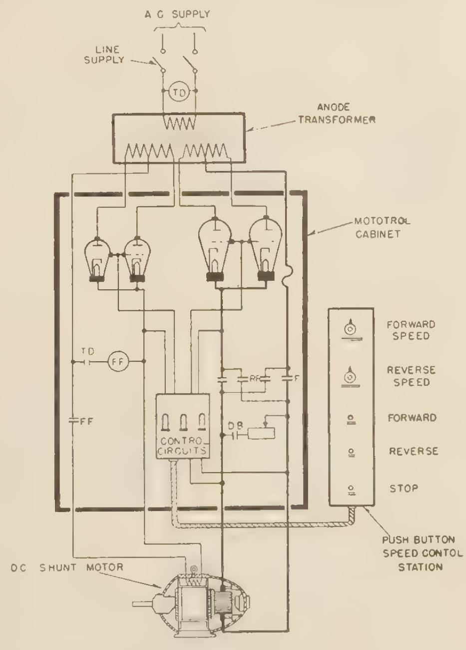


Fig. 1 Schematic Arrangement of Electronic Adjustable-Speed Motor Drive

⫲	Relay contact which is open when relay is de-energized
—O—	Relay magnet coil
—□—	Dynamic braking resistor
TD	Time delay protective relay to prevent operation until the cathodes are heated
FF	Field failure relay to prevent operation of motor if field is not energized
R, F	Reverse and forward contacts on contactor for reversing motor rotation
DB	Dynamic braking relay contact. Used to brake motor when stopping or reversing

[1] Manager, Resistance-Welding Engineering, Westinghouse Electric & Manufacturing Company.

Contributed by the Special Design Committee and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

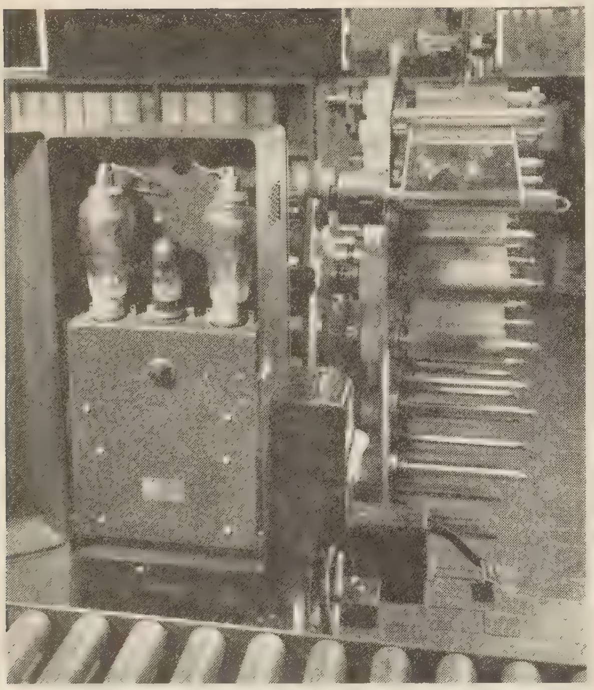


Fig. 2 Electronic Adjustable-Speed Motor Drive on Capacitor Winding Machine

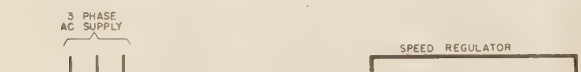


Fig. 3 Schematic Diagram of a Variable-Voltage Speed Control With Electronic Speed Regulator

There are many motor-control systems where speed regulation is desired, but there is no advantage in full electronic control. For this purpose, a speed-regulator system has been developed as shown in Fig. 3. In this particular system, the direct-current motor is furnished with power by a motor generator set. The motor-generator-set field is energized from the electronic regulator. The motor speed is indicated by the speed-indicating generator whose output voltage is proportional to speed. The indicating voltage is applied through an amplifier and regulating circuit to the grids of the rectifying tubes which furnish the generator field. This device has found ready acceptance in paper mills for regulating the speed of a single-motor paper-machine drive. It maintains the speed to an accuracy of approximately 0.5 per cent and may be adjusted over a speed range of 10:1. A similar regulator is used to hold constant speed on the Wright Field 40,000-hp wind-tunnel drive.

Resistance-Welding Control

Resistance welding is being extended rapidly to speed production of fabricated assemblies. It is particularly adaptable to manufacturing processes involving large quantities of repetitive assemblies. It is most commonly used on sheet metal, although heavier parts including heavy plate and premachined pieces are also being welded.

Resistance welding is accomplished by passing a heavy current through two or more pieces of metal. The resistance of the metal to current flow develops heat, which fuses the pieces together. The material is clamped between copper or copper-alloy electrodes which apply pressure and conduct the heavy current to the junction point. The welding machine itself consists essentially of a heavy framework supporting a special transformer, electrodes, and pressure head for the electrodes.

For many years, the use of resistance welding was confined almost entirely to the coarser sheet-steel structures which could tolerate a considerable amount of electrode marking, warping, and discoloration. The process has become in recent years more of a precision process and has been extended to a variety of alloy

Fig. 4 Seam-Welding Machine

metals such as stainless steel, brasses, and bronzes, and to nonferrous metals of sharp fusion points such as aluminum. In wartime industries, welding has rapidly found its place in the manufacture of airplanes, trucks, shells, gasoline drums, airplane drop tanks, and other similar assemblies.

Resistance welding is divided into several different general types, the more distinct classes being spot, projection, butt, and seam welding. Spot, projection, and butt welding differ only in the type of electrodes and method of holding the materials during welding and use the same control unit. The control measures out one or more pulses of current to make an individual weld. Seam welding uses roller electrodes, and the control is quite different from that for spot welding. The control causes the current to flow in individual pulses of a few cycles' duration each and with a few cycles time between each pulse for as long as welding is proceeding.

Fig. 5 is a typical resistance-welding system, showing in this particular case spot-welding electrodes, although the diagram applies equally well to seam welding when

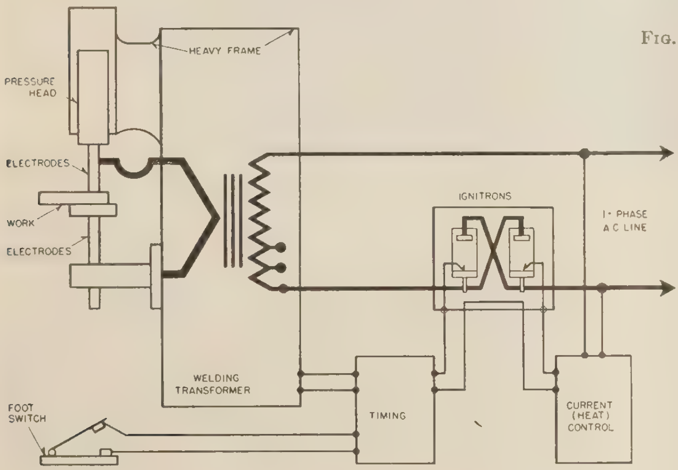


Fig. 5 Schematic Arrangement of a Spot Welder and Its Control

(The timing and current controls shown as blocks operate in the control circuit of the ignitron power tubes to time the duration of the flow of current and determine its magnitude. The proper time and current values are selected by experience and preset manually for the material being welded.)

roller electrodes are substituted for the pointed tips. The resistance-welding system consists of the welding machine with the welding transformer mounted in it, together with control equipment including an ignitron power assembly for controlling the energy fed to the welding transformer, a timing control, and the current or heat control. When the operator's foot switch is depressed, the welding electrodes are lowered to the work and when pressure has been fully applied, the timing unit causes the ignitron unit to conduct current to the welding machine. The timing control determines the length of time that welding current will flow and, after the predetermined length of time, causes the current to stop. The timing is usually between 1 and 30 cycles. For various work, it is necessary to have different values of current, and therefore the current input to the welder must be adjusted when setting up. The heat-control unit serves this purpose. For some special applications the current is modified automatically during the progress of the weld. Electronic control has proved valuable for resistance welding because of reduced maintenance at high operating speeds and also because of the precision obtainable.

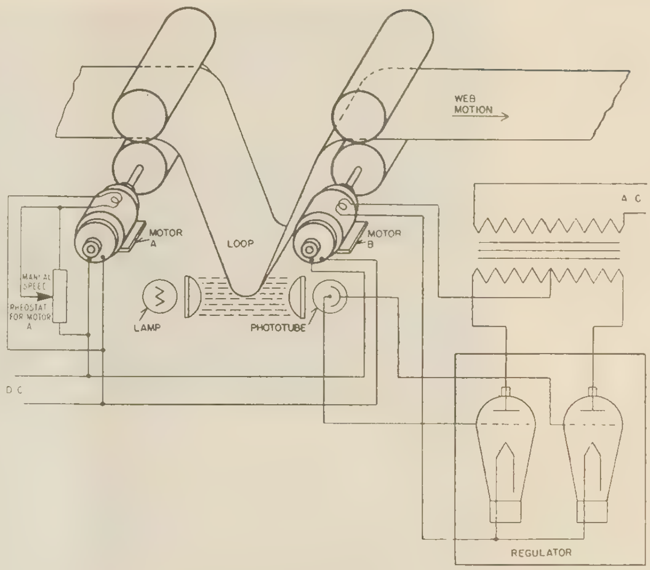


Fig. 6 Schematic Arrangement of a Loop Regulator

The ratings of welding control are based upon the current-carrying capacity of the control and the duty cycle primarily, the ignitron tubes being rated at currents up to 10,000 amp during the weld at low-duty cycle. Inasmuch as the welding transformer steps current up several times, this value of primary current would correspond to perhaps 80,000 or 100,000 amp in the weld itself. Most equipment operates from industrial voltages of 220 or 440 v, although control is now available for 2300-v operation. The high voltage operation is particularly desirable for large welding machines to minimize the effect of the high current demand on the line voltage and in many cases to save on the amount of bus required for feeding the power to the machine.

Control Using Light

Control utilizing light is called "photoelectric" because it is actuated by the response of a light-sensitive element called a "phototube." Photoelectric control is useful for actuating a system in response to a change of color, shade, density, shape, or position without mechanically touching the controlling object.

Many of the more important photoelectric applications may be classed as process regulators. Thus a loop regulator on a paper-coating machine or on a steel strip mill watches the loop position and calls for slowing down or speeding up one of the sections of the mill to maintain a constant loop. The system in Fig. 6 is a simplified arrangement used where the speed range is not more than about 4 to 1. The web is driven by motor A into the loop and taken out of it by motor B. Motor A and motor B can, for instance, operate two different sections of a conveyer or two separate stands of a mill. The speed of motor A is adjusted as desired, and the regulator then keeps the speed of motor B at the proper value to maintain a constant loop between the two sections. An increase of loop length reduces the amount of light reaching the phototube from the light source and modifies the field excitation of motor B, adjusts its speed, and brings the loop back to the proper position.

Fig. 7 Bread-Wrapping Machine With Photoelectric Control of Cutting to Place Trade-Mark at Same Spot on Each Loaf

A regulator used frequently in the paper, cellophane, and glassine processing industry is that for maintaining the position of cut when taking from a roll of preprinted stock for making bags or packages. Although the machine feeds paper at a rate approximately correct to match the speed of the rotating knife, the cut will never remain registered perfectly without either manual or automatic supervision. A register regulator corrects for changes of paper slippage, stretch, and speed relative to the cutter speed. The position of the printed material is indicated by a rectangular register mark in the design or margin which is scanned by a phototube. The position of the mark with respect to the cutter is checked and the speed of paper feed modified if necessary. A typical installation of this type is shown in Fig. 7, for packaging bread.

When slitting strip material, it is often necessary to regulate the sideways position of the web to insure cutting at the proper point relative to a trade-mark or printed design. The slitter regulator in Fig. 8 scans the edge of the paper web or a line in the printed design. The proper web position is obtained by moving the roll of material to the position called for by the phototube in the scanning device. The same device is used to scan the edge of a web and correct the roll position to insure smooth roll windup when rewinding mill rolls. An installation of this device on a printing press is shown in Fig. 9.

The wartime shortage of tin brought about the necessity for thinner coating of tin on strip steel for cans. This has been accomplished by the electrolytic tinning process. The electroplating process leaves the tin surface with a silvery appearance and somewhat porous structure. It is necessary to heat the strip enough to melt and flow the tin and thus obtain a more perfect coating. Not only does an electronic high-frequency oscillator supply power for flowing the tin after electroplating, but it is also necessary to regulate the flowing of the tin so that exactly the right amount of high-frequency power is furnished, thereby avoiding spoilage. If too much power is furnished to the high-frequency induction heating coil, the flow line will occur much earlier in the heating coil than it would at lower power levels. It is therefore desired to maintain the power input to the heating coil at a level which will cause the tin to flow at a certain predetermined position in the coil. The flow line is readily distinguished because the tin coming directly from the plating tank has a very diffusing surface whereas that tin which has been flowed has a mirror surface. The regulating system is shown in Fig. 10 with a light source and phototube scanning the plated strip. The scanner detects the exact line at which this change of condition occurs. If the speed of the tinning line changes, it is necessary to modify the power input accordingly. This is accomplished through the scanner which observes that the position of flow line has changed, and which energizes through the thyratrons either relay F or R to start the induction-regulator motor. This adjusts the regulator so as to increase or decrease the power furnished to the oscillator and to bring the flow line back to the proper position.

Detecting Holes in Strip Steel

Not all photoelectric applications are for regulation purposes. An important steel-mill application of another type is that for detecting holes in strip steel. This application is particularly desirable on steel which will be used for the canning industry. Prior to the use of pinhole detectors, it was difficult to inspect visually the strip effectively and food spoilage resulted. The pinhole detector, Fig. 11, will inspect steel strip at speeds up to 1500 fpm and detect $^1/_{64}$-in. or smaller holes. The detector operates a marker which places a visible mark on the strip at the location of the hole so that it can be readily located later on. In many installations, the detector also operates a classifier to reject defective sheets automatically, after they have gone through the shear which usually immediately follows the pinhole detector.

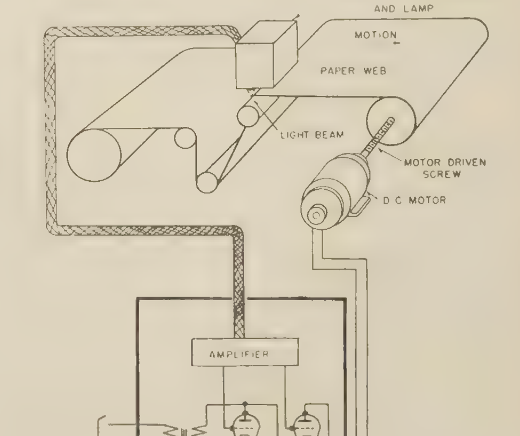


Fig. 8 Schematic Arrangement of Regulator for Controlling Slitting or Rewinding of Paper, Glassine, or Cellophane

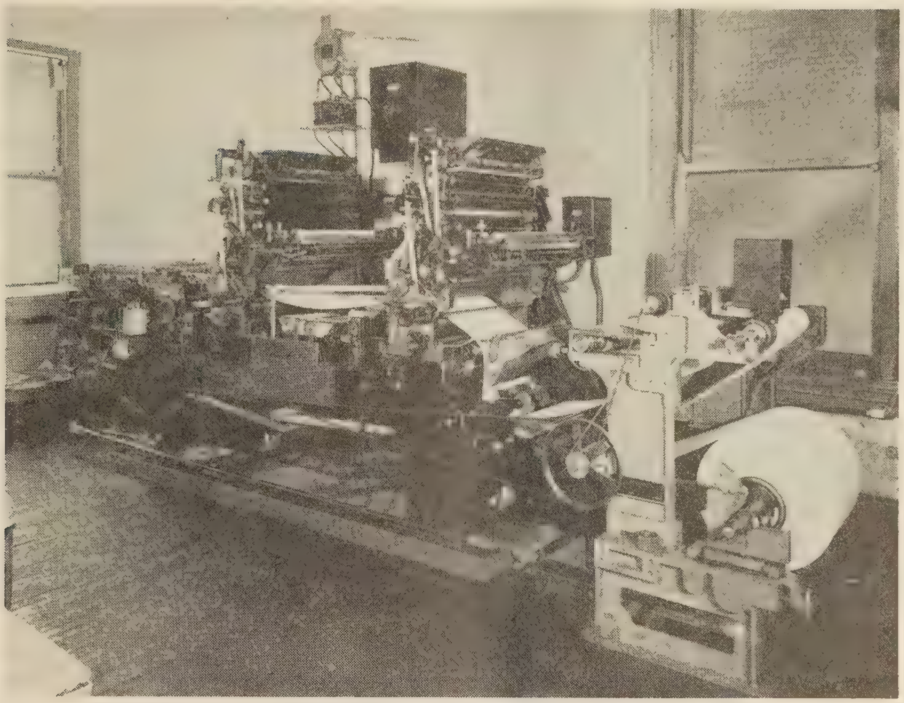


Fig. 9 Slitter Regulator Used on a Printing Press to Maintain Position of Paper Web

The pinhole detector is probably one of the more prominent commercial applications of sorting.

Much loose publicity has been given to the use of phototubes for sorting applications. It should be realized that the phototube responds only to a difference of light energy reaching it and also in some degree to changes of the color of the light. The phototube does not have any intelligence and therefore cannot duplicate the performance of the human eye for the great majority of sorting problems. The only way in which a phototube can be made to appear to have judgment or intelligence is to design the optical system so that the phototube will receive a varying amount of light depending upon a particular characteristic of the material which it is desired to detect. The application of phototubes to sorting problems is usually very expensive because of the considerable development involved and many times cannot be justified for that reason alone. This development expense can be absorbed if the commercial possibilities for the finished design are sufficiently enticing, but if they are not, the individual user is not ordinarily willing to pay the development charges.

Many applications for photoelectric control can be taken care of by general-purpose devices called photoelectric relays which may be considered as light-sensitive limit switches. These are available in varying forms and at varying prices, depending upon the desired sensitivity and functions to be performed, and can be adapted by the purchaser to his own application. These include counting, drinking-fountain control, door control, and conveyer control as shown in Fig. 12, and numerous others known only to the many users.

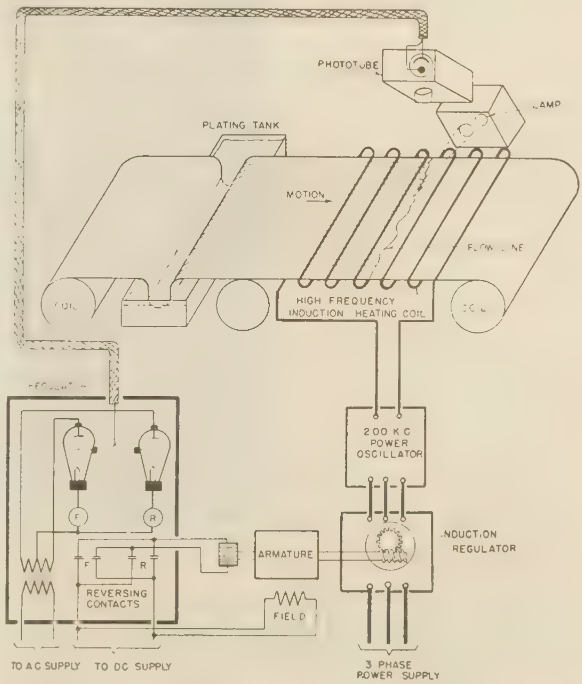


Fig. 10 System for Regulating Power Input in Tin-Flowing Process

—O—	Relay coil	F	Forward relay
⊣⊢	Relay contact	R	Reverse relay

Conclusion

The applications described show a typical variety for electronic control. These have come about through close co-ordination of the electrical and mechanical design. Only by this co-operation has it been possible to insure that the electrical and mechanical equipment will match each other in precision and reliability. If they do not match, a regulator with $^1/_{64}$-in. accuracy may be attempting vainly to control a machine with $^1/_8$-in. backlash, or a

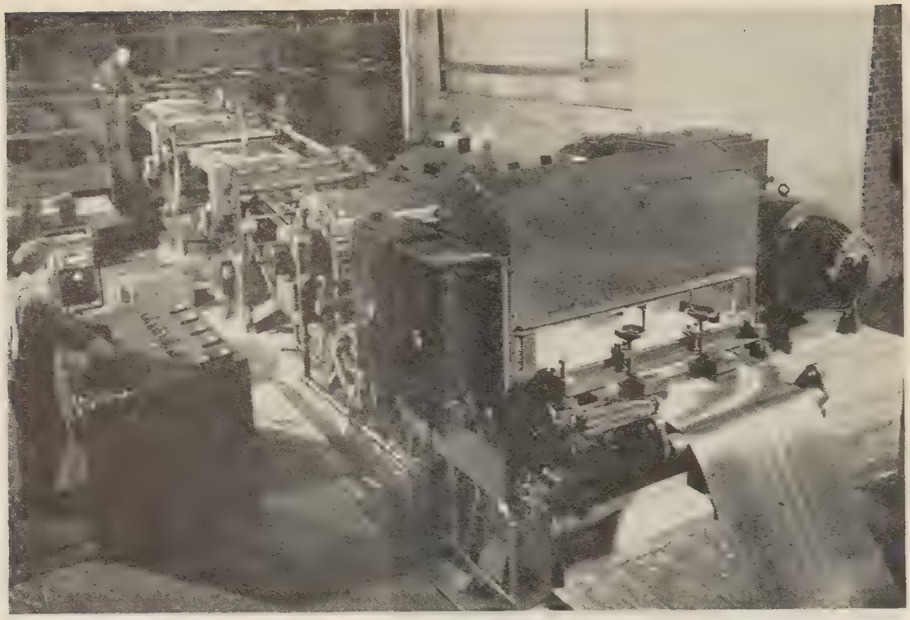


Fig. 11 Pinhole Detector in a Steel-Strip Shearing Line

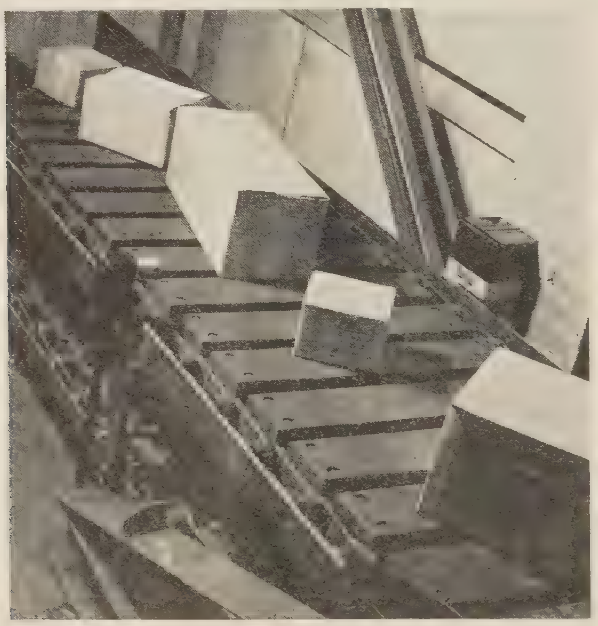


Fig. 12 Counting or Conveyer Control

precision machine may lose its value because of coarse control.

Electronic control, properly selected for applications where it can perform better or less expensively, will do what seems to be miracles. This is so because new tools with incredible speed and versatility have been added to the designer's kit. His vision is thus broadened beyond the pure mechanical or the electromechanical device to the electronic tube which is purely electrical and has incredible speed and versatility. Through proper combination of all three of these tools designers are unfolding new engineering accomplishments with amazing possibilities.

Radiation Pyrometry in Turbosupercharger Testing

By V. P. Head,[1] West Lynn, Mass.

With increase in operating temperatures and speeds of turbines such as are used in turbosuperchargers, the demands for turbine-wheel and bucket temperature measurement have become increasingly exacting. Requests appear for such measurements under operating conditions with measurement tolerances varying from ±50 F in flight test to ±5 F in certain ground tests. This paper outlines the procedures used for measurements ranging from as low as 300 F up.

The following alternate procedures have found limited application in measuring turbosupercharger turbine-wheel and bucket temperatures:

1 *Optical Pyrometry.* Bucket hot-spot temperatures have been determined with a monochromatic optical pyrometer of the disappearing-filament type. Above about 1400 F such readings have been accepted with an allowance of about ±30 F.

2 *Fusible Alloys.* Temperature versus radius curves on the wheel have been obtained with fusible-alloy plugs with accuracy limited by the interval between consecutive melting points of the alloys available. Maximum temperatures during the tests, rather than instantaneous values, were obtained and have been valuable in determining wheel operating temperatures with various cooling methods.

3 *Fusible Compounds.* Fusible compounds commercially available as pellets, with melting points at 50 F intervals, have been used to give wheel temperature versus radius curves with a probable error of 1 or 2 per cent of the absolute temperature. Again indications represent maximum values, but the pellets are made to indicate approximate instantaneous values by approaching desired test temperatures gradually with no overshoot.

4 *Contact Thermocouples.* Base-metal thermocouples, peened into the wheel at various radii, have been used successfully with slip rings to give instantaneous readings in comparing cooling methods, but speed limitations and complexity of rotating parts have prevented the more general application of this procedure.

General Methods and Applications of Thermoelectric Radiation Pyrometry

Thermoelectric radiation pyrometers commercially available have been applied to give the local temperatures at wheel-rim and bucket surfaces within about ±50 F at 1000 F, and with increase in accuracy at higher temperatures, by mounting and cooling appropriately. The area of the spot observed has been varied arbitrarily by the proximity of the pyrometer to the "target" (wheel or bucket surface), by the size of aperture, or by the use of lenses. A knowledge of the pyrometer-jacket temperature as well as the pyrometer output has been required, and the use of a cooled shutter for determining the magnitude of the "zero drift" (described in the section entitled "Theory") has helped to facilitate the readings. In flight tests, conducted by the General Electric Company and Boeing Aircraft Company in 1941, adequate results were obtained by scooping atmospheric air to cool the pyrometer mounting and prevent soot accumulation, where a spread of 50 F in the observed temperatures was permissible. The pyrometers used were of the glass-bulb vacuum thermocouple type.[2]

Such thermoelectric radiation pyrometers have been adapted to meet recent exacting demands for greater accuracy at lower temperatures. Calibration repeatability within ±5 deg over a range of temperatures from 600 F up has been obtained. The inherent error of base-metal thermocouples used to measure the calibrating-target temperatures has limited the absolute accuracy obtained.

The experimental miniature pyrometer described herein has provided instantaneous readings of wheel and bucket temperature during test and is now being employed as a guide for the control of test conditions when arbitrary temperature versus radius curves are desired.

The arrangement may be applied to the temperature measurement of gases and vapors by sighting into the end of a long "honeycomb"[3] in a well-lagged duct through which the gas may be passed.

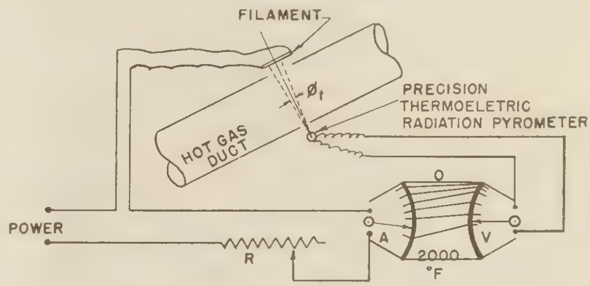


Fig. 1 Proposed Adaptation of Kurlbaum's Flame Method to Static Temperature of High-Velocity High-Temperature Gases Not Completely Transparent

(A: Ammeter calibrated for filament temperature. V: Galvanometer indicating filament temperature by thermoelectric radiation-pyrometer output, calibrated with clean, dry cold air in duct. R: Manual filament-temperature-control resistor. ϕ = pyrometer sight angle, completely subtended by filament surface.)

The static temperature of hot gases at high velocity may be measured by replacing the optical pyrometer of Kurlbaum's flame method[4] with a precision thermoelectric radiation pyrometer. One possible arrangement is suggested in Fig. 1, in which the manual adjustment of a single resistor permits variation of a background temperature until the actual and indicated background temperatures are equal, when the indicated tempera-

[1] Supercharger Engineering Division, General Electric Company.

Contributed by the Aviation Division and Industrial Instruments and Regulators Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

[2] "Vacuum Thermocouples of the Radiation Type," by S. S. Stack, *General Electric Review*, vol. 42, 1939, pp. 365–366.

[3] "Measurement of High Temperatures in High-Velocity Gas Streams," by W. J. King, Trans. A.S.M.E., vol. 65, 1943, pp. 421–431.

[4] "The Measurement of Flame Temperature," by G. Ribaud, Y. Lauri, and H. Gaudry, *Journal of the Institute of Fuel*, vol. 12, 1939, pp. S-18–S-30.

ture will be actual gas-static temperature, regardless of gas emissivity.

It is proposed that the arrangement may be used as a secondary standard in the calibration of base-metal thermocouples, thermometers, and the like, for temperatures above 1000 F.

It is conceivable that rigorous application of thermoelectric radiation-pyrometry theory may permit sharper definition in the "International Centigrade Temperature" scale at elevated temperatures, reducing the spread entailed by the brightness-comparison methods of optical pyrometry, since the "readability" of the output increases with approximately the third power of the measured temperature.

An interesting application is the measurement of pressure at high vacuum, using a target of constant temperature such as a lamp with controlled voltage, and connecting the pyrometer chamber to the vacuum to be measured, as described further under "Tests."

Arrangements. Fig. 2 shows the arrangement of a mounting used in a high-velocity hot-gas stream. The commercial pyrometer, an evacuated-glass-bulb type, was wrapped in aluminum

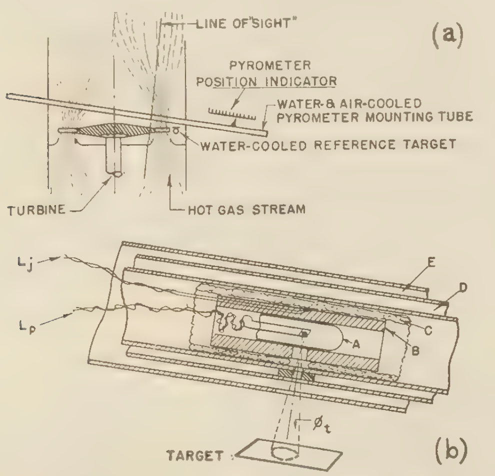


Fig. 2 (*a*) Typical Installation of Pyrometer Mounting for Turbine-Wheel Traverse. (*b*) Mounting Arrangement for a Commercial Vacuum-Type Pyrometer

A — Pyrometer
B — "Heat mass" (extra heavy pipe)
C — Thermal insulation
D — Cooling-water passage
E — Air Passage
L_p — Pyrometer leads
L_j — Pyrometer-jacket thermocouple leads
ϕ_t = Solid angle of sight

foil and encased in a heavy-walled tube of high thermal conductivity and high specific heat, which acted as a heat "flywheel" to prevent sudden changes in pyrometer temperature. A thin layer of thermal insulating material supported the massive tube within a cylindrical water-cooled outer jacket. An additional jacket carrying air prevented soot accumulation at the sighting aperture. A water-cooled reference target permitted zero-drift observation, and even with the best combination of heat capacity and thermal insulation, corrections amounting to several times the full-scale reading of the galvanometer were frequently required. For this purpose a zero-drift compensator was inserted in the pyrometer circuit, Fig. 3, so that by introducing the proper electromotive force, the galvanometer could be rapidly set to zero. The reflection of extraneous radiation from the surface of the reference target left this adjustment open to some question, and the necessity for the elimination of zero drift, together with the need for reduced bulk in the pyrometer mounting led to the development of the miniature radiation pyrometer, Fig. 4.

Fig. 4 shows the essential features of a thermoelectric radiation pyrometer embodying the theoretical requirements for precision. Tests conducted on a model of this type show that its repeatability is well within the limits of error of the base-metal thermocouples employed in calibrating. It has the further advantage of small size and self-contained "heat capacity," so that the jacket required for mounting in a high-velocity hot-gas stream may be kept to a minimum bulk. The necessity for a cold reference target and zero-drift compensator has been eliminated.

Attempts to seal this model to hold a vacuum permanently have been unsuccessful to date and the device is employed at atmospheric pressure. The sacrifice of high output has been largely offset by the relative freedom from changes in calibration

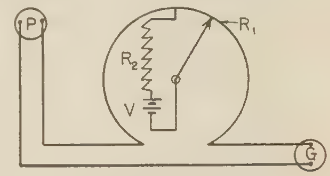


Fig. 3 Rapid Zero-Drift Compensator for Precision Thermoelectric Radiation Pyrometer Circuit

P — Pyrometer
G — Galvanometer or potentiometer
R_1 — Low-resistance slide-wire
R_2 — High resistance
V — Dry cell

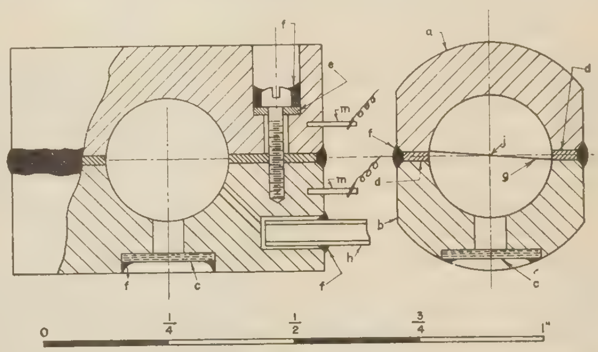


Fig. 4 Miniature Radiation Pyrometer Embodying Precision Requirements

a, b: Upper and lower half-jackets, serving also as thermal capacitance and electric conductors; commercially pure copper
c: Window: glass, rock salt, etc.
d, e: Electric insulation
f: Sealing compound
g: Thermoelectric element with receiver and hot junction at center (*j*), and cold junctions at pyrometer walls
h: Glass evacuating and sealing tube
m: Pyrometer leads
Not shown: Thermal insulation, outer cooled mounting, and sight-limiting aperture

with small changes in jacket temperature, a characteristic which is highly desirable. The output is read on a light-beam-type galvanometer of portable construction having a full-scale deflection (100 mm) at about 50 μv. The characteristics of the pyrometer are such that a relatively small target temperature range is conveniently spread over the upper 90 per cent of the scale range, Fig. 7(*b*). The use of a direct-reading galvanometer requires that there be no change in lead resistance between calibration and application, but no attempts to use a potentiometer have been successful. The time required to balance a potentiometer permitted inestimable zero drift with the commercial high-output-type pyrometer, and the readability of the potentiometers available has been inadequate for the miniature type.

Nomenclature

The following nomenclature is used in discussing the theory of thermoelectric radiation pyrometry:

T_t = target absolute temperature
T_j = absolute temperature of pyrometer jacket
T_r = absolute temperature of pyrometer receiver
K_0 = constant governing radiation between target and receiver
K_1, U_1 = constants governing radiation between receiver and jacket
K_2, U_2 = constants governing convection between receiver and jacket
U_3 = constant governing conduction between receiver and jacket
ϕ_t = solid angle of radiant influx from target to pyrometer receiver
y = exponent of temperature difference applicable to convection between receiver and jacket

Theory of Thermoelectric Radiation Pyrometry

Fig. 5 shows an idealized radiation pyrometer: a small receiver of low heat capacity, a spherical jacket of high heat capacity and temperature T_j, and a transparent window through which a target of temperature T_t may be "seen" by the receiver through a solid angle of ϕ_t steradians. We may write the steady-state energy-flow equation governing the receiver temperature T_r

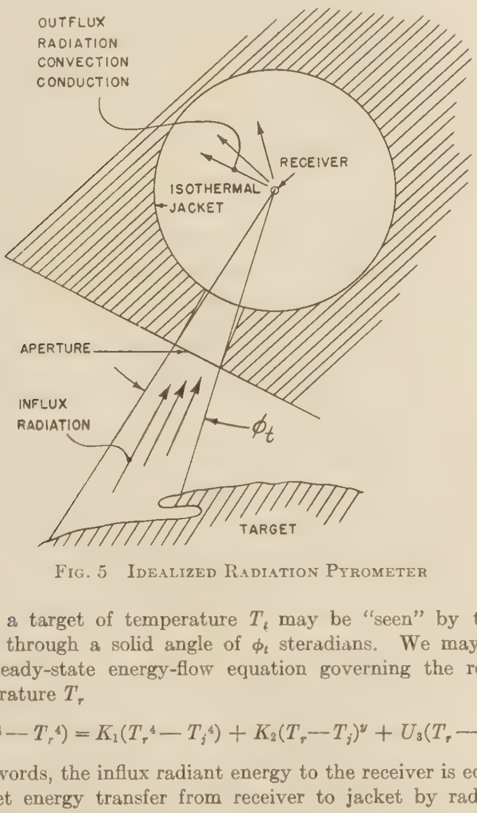


Fig. 5 Idealized Radiation Pyrometer

$$K_0(T_t^4 - T_r^4) = K_1(T_r^4 - T_j^4) + K_2(T_r - T_j)^y + U_3(T_r - T_j) \quad [1]$$

or in words, the influx radiant energy to the receiver is equal to the net energy transfer from receiver to jacket by radiation, convection, and conduction.

We may introduce a gross approximation for the convection term without seriously jeopardizing our qualitative conclusions

$$K_2(T_r - T_j)^y = U_2(T_r - T_j) \quad [2]$$

If the difference between T_r and T_j is less than, say, 1 per cent of either, the receiver-to-jacket radiation term may be written with fair precision

$$K_1(T_r^4 - T_j^4) = U_1 T_j^3(T_r - T_j) \quad [3]$$

On the left side of Equation [1], T_r may be replaced by T_j, since compared to T_t their difference is in general negligible.

We may now rewrite Equation [1] to give a function which is more readily scanned by eye, if slightly less precise

$$K_0T_t^4 = (U_1T_j^3 + U_2 + U_3)(T_r - T_j) + K_0T_j^4 \quad [4]$$

Then in order to measure T_t with precision, we must measure precisely a quantity proportional to $(T_r - T_j)$ and we must know T_j itself at least approximately.

Many forms of radiation pyrometers have been built in which radiation from a hot target is intercepted by a receiver to produce a temperature difference between the receiver's "hot" junctions and "cold" junctions located vaguely somewhere else. It must here be recognized that the criterion temperature difference is that between the receiver and the jacket walls. Failure to locate the thermoelectric cold junctions at the walls leads to a time lag between the walls and the cold junctions when the slightest drift in jacket temperature occurs. This will produce the "zero drift," previously referred to, which in low-temperature applications may frequently amount to *several hundred per cent* of the criterion temperature difference $T_r - T_j$, and render the output of the pyrometer meaningless. This random location of cold junctions "somewhere else" may be largely responsible for the prevalent notion that radiation pyrometry is impractical at temperatures appreciably below 1400 F.

Let us therefore locate in our ideal pyrometer a thermoelectric element with hot junction at the receiver and with cold junctions at the walls, and since we will require a knowledge of $T_r - T_j$ to within a few thousandths of a degree, let us use a massive high-conductivity jacket to assure that its temperature will be *uniform* throughout.

The term $K_0T_j^4$, Equation [4], will be found negligible for 0.1 per cent accuracy in target temperature measurement, when T_t is more than 4 times T_j, while a 4 per cent variation of T_j from that at which calibration was made will be negligible when T_t is only about 2.5 times T_j. Thus at normal room temperature, T_j may vary ±20 deg with no effect when T_t is above about 800 F, so far as the influx of radiant energy from the target is concerned.

A more troublesome effect of jacket temperature deviation lies in the variation of radiant energy transfer within the pyrometer, indicated by the factor $U_1T_j^3$, Equation [4]. When we strive for high output by evacuating the pyrometer jacket and making the thermoelectric elements as slender as possible (decreasing U_2 and U_3), we make the effect of jacket temperature change more and more critical. By retaining a high convection rate at a sacrifice of output, and by polishing the inner jacket wall to reduce U_1, we may render the effects of small deviations of jacket temperature negligible.

The coefficient K_0, Equation [4], is dependent upon the geometry of the arrangement and the "blackness" of the target surface. It may be increased by increasing the value of ϕ_t by the change of aperture size, usually entailing an undesirable increase in area of the target portion viewed, or by the use of a lens. A lens or window is never perfectly transparent, and contributes to the effective jacket temperature. A large lens external to the heavy jacket proper is therefore apt to present difficult temperature-control problems and is to be avoided. Constant K_0 may be further increased by pretreatment of the target surface as described under "Tests." Here repeatability of calibration is even more important than the increase in output.

The radiation transmission within the pyrometer may be reduced for increased output by polishing the inner-jacket-wall surface. The conduction factor U_2 has been commonly reduced by using extremely thin thermocouple wires or ribbons.

The convection factor U_3 is a function of the shape, size, and pressure of the jacket interior. In general, evacuating a pyrometer will increase its output by a ratio of 7 or 10 to 1. The

transition of sensitivity occurs rather suddenly over a limited range of pressures, through which the mean free path of the molecules of gas in the jacket appear to be of comparable order to the linear dimensions of the inner jacket chamber. It must be remembered that the tempting increase in output will demand much closer control and measurement of the pyrometer jacket temperature, unless the radiation factor $U_1T_j^3$ can be reduced proportionately. A more detailed analysis of the constants involved and their bearing upon the problem of jacket temperature deviation has been given elsewhere.[5]

A rigorous analysis of pyrometer performance will yield an equation very different from Equation [4], involving among other things an integration of a function of "monochromatic" radiation "density" in a differential spectral band at varying temperature with respect to the spectral distribution of a nonblack surface, further modified by the selective transmission of the particular window or lens material; hardly a feasible project in wartime. It is sufficient to say that the result is not a fourth-power function, nor is the pyrometer in any sense a "total radiation" pyrometer. Empirical formulas, involving a temperature exponent between 4 and 5, have been found to fit portions of the calibration curves, but they have contributed nothing to the usefulness of the project. Attempts to use Equation [4] for any but qualitative considerations should be avoided.

Tests of Commercial and Experimental Pyrometers

1 Tests of zero drift in a commercial vacuum-type pyrometer were conducted to determine the optimum arrangement of thermal resistance and capacitance between a water-cooled housing of fixed diameter and the pyrometer. No aperture or target was used. The criterion for zero drift was the pyrometer-output versus time curve of each arrangement, produced by a sudden drop of 30 deg F in water temperature. Fig. 6 shows the results obtained with seven such arrangements, which are numbered in

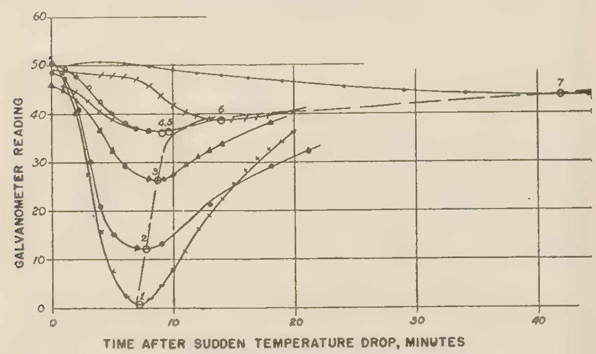


Fig. 6 Zero Drift in a Commercial Vacuum-Type Pyrometer

(Solid lines: zero-drift versus time curves produced by a sudden 30 F drop in cooling-water temperature with seven combinations of thermal resistance and capacitance confined in outer jackets of equal size. Dashed line: trend of magnitude and duration of zero drift with variation in jacket construction, indicated by the peaks, i.e., numbered points, of the solid lines. Point numbers go up with increase in "heat-mass" concentration of the jacket arrangements tested.)

order of increase of "heat-mass" concentration. The dashed line through the peaks of these curves shows the trend from high drift of short duration to low prolonged drift. Curve 7 was obtained with an arrangement similar to Fig. 2(*b*), which in service permitted zero-drift adjustment, a target observation, and a good zero check after observation.

Zero drift in the experimental pyrometer, Fig. 4, was determined by a sudden increase of 37 deg in the jacket temperature. The zero drift persisted for about 7 min, and its maximum value was about four galvanometer divisions at the end of 3 min. The effect of this zero drift is shown in Fig. 9 (*b*). Since no such sudden jacket transients occurred in service, the use of a zero adjuster and cold reference target was discontinued, eliminating the reference-target reflection error previously described.

2 Repeat calibrations at several jacket temperatures indicated the significance of jacket deviations in service. The commercial pyrometer required a correction of a little over 1 deg per deg deviation of jacket temperature at a target temperature of 1200 F, and an increase in the correction as the target temperature rose. The corresponding correction for the experimental pyrometer without vacuum was less than 1/4 deg per deg, with a

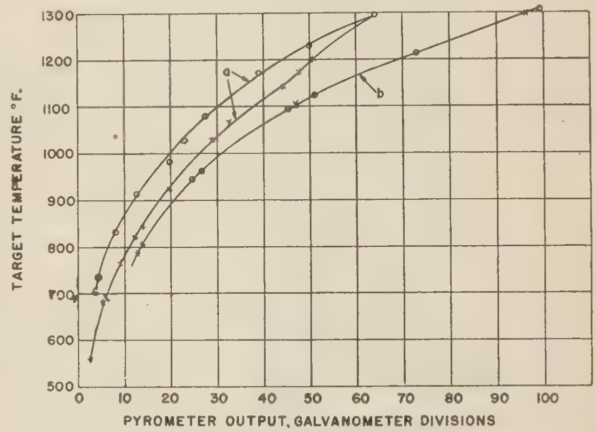


Fig. 7 Effects of Target Pretreatment on Thermoelectric Radiation Pyrometry

(*a*, Target polished before test. *b*, Target lightly sandblasted and preheated to 1400 F. Encircled test points indicate increasing temperature. Crossed test points indicate decreasing temperature. Point scatter, curve *b*, is typical of pyrometer repeatability.)

tendency to decrease with an increase in target temperature. This tendency is not apparent from Equation [4] and is best explained by the fact that the convection exponent *y*, Equation [1], is appreciably greater than unity, so that the value of U_2, Equation [4], increases with $(T_r - T_j)$. In words, the rate of heat transfer between the receiver and jacket by convection increases with target temperature faster than does the transfer by radiation. The effect upon the required correction is a real trouble saver.

3 Tests on the effects of target emissivity drift were conducted by using a calibrating target of test-target material, initially bright, which was gradually heated and cooled. Readings of pyrometer output with increasing and decreasing target temperature were observed. Fig. 7 shows the results obtained with two initial surface conditions. It was found that a light sandblast of the calibrating target rendered the emissivity drift negligible, Fig. 7(*b*), though in subsequent tests where turbine-wheel temperatures were to be observed, both wheel and calibrating target were preheated to the highest contemplated test temperature (or as closely as possible without metallurgical damage) after a light surface sandblast.

4 The "calibration-repeatability" tests were identical with the target emissivity variation tests where no visible effect of emissivity variation occurred, Fig. 7 (*b*). In addition, a series of closely spaced calibration points was obtained and plotted on a highly expanded scale. The scatter of the test points about a smooth curve drawn through them is shown in Fig. 9 (*a*) and (*c*), where the greatest deviation noted is 3 1/2 deg, or a spread of

[5] "An Improved Radiation Pyrometer," by T. R. Harrison and W. H. Wannamaker; "Temperature—Its Measurement and Control in Science and Industry," American Institute of Physics, Reinhold Publishing Corporation, New York, N. Y. 1941, pp. 1206–1224.

7 deg. It was expected that the point scatter would decrease at elevated target temperatures because of the decrease in slope of the calibration curve. In practice, it was found that at temperatures above 1000 F, target-temperature versus time curves taken with the radiation pyrometer were much smoother than those taken with the base-metal contact thermocouple peened into the surface of the target. It is justifiable, therefore, to blame the persistent scatter at elevated temperatures on the poor adaptability of the contact thermocouple to precise measurement of surface temperatures when air or gases at different temperatures are present.

In early wheel tests, vibration produced occasional shift between pyrometer and jacket, changing the solid angle of sight. Another source of difficulty was sooting of the pyrometer aperture, which was eliminated by the use of an air jacket outside the water jacket. This also eliminated the water film formed occasionally by condensation on the aperture. Water is extremely opaque to infrared radiation. All of these possibilities must be carefully guarded against in any application.

5 Tests on the effect of varying internal pyrometer pressure were conducted on the experimental pyrometer, Fig. 4, by sighting on an electric lamp of controlled voltage and connecting the evacuating tube to a good vacuum pump. Fig. 8 shows the result and is comparable to the characteristic curve of a thermocouple-type vacuum gage. Note that rough evacuation produces no appreciable change in sensitivity and that there is a point, presumably where conduction and radiation transfer swamp out the remaining convection transfer, where further refinement of the vacuum is useless. It is interesting to note that the inflection point of the curve in Fig. 8 occurred at a pressure corresponding to a mean-free-path for air molecules of approximately 0.1 in., while the actual distance from receiver to inner jacket wall was about $^1/_8$ in.

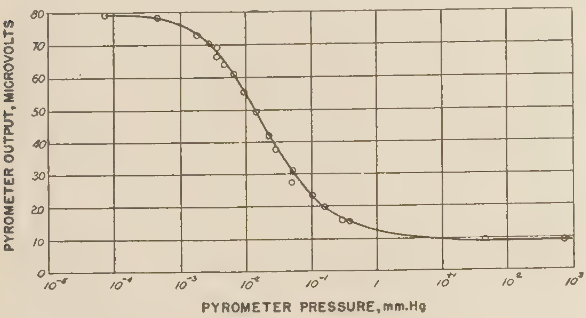


Fig. 8 Effects of Internal Pressure Variation on Thermoelectric-Radiation-Pyrometer Output With Constant Target Temperature

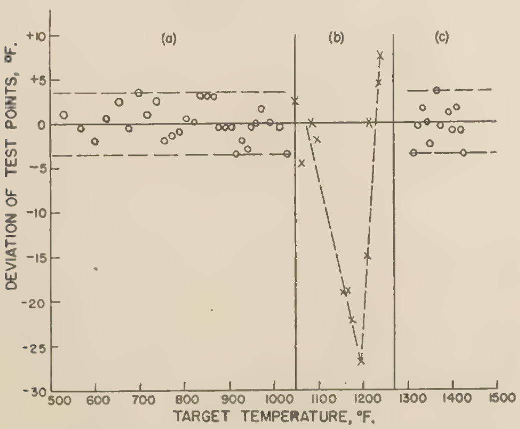


Fig. 9 Performance of Miniature Radiation Pyrometer Calibrated at Atmospheric Pressure

(*a*, *c*, Point scatter about a smooth curve. Pyrometer jacket held at 66 ± 2 F. *b*, Zero drift introduced by a sudden increase of 37 F in jacket temperature.)

To date, the device in Fig. 4 has presented sealing difficulties and has been applied in turbine-wheel temperature measurement at atmospheric pressure in mountings similar to that shown in Fig. 2, with considerable reduction of bulk. The resulting reduced sensitivity has been offset at low temperatures by the use of certain crystalline salt apertures, transparent to infrared, in place of glass. The relative freedom of this arrangement from the effects of small jacket-temperature deviation has reduced personal supervision of tests to a minimum.

Conclusions

The requirements for basically sound thermoelectric radiation pyrometry may be briefly stated as follows:

1 The target surface must be of constant "blackness" throughout calibration and application.

2 It is necessary to make a precise measurement of the difference in temperature between a receiver and its surrounding jacket, or the corresponding emf or galvanometer equivalent.

3 It is necessary to measure or know at least approximately, and in applications below about 800 F, precisely, the temperature of the jacket.

4 It is advantageous to sacrifice the high output associated with an evacuated pyrometer when freedom from the effects of small jacket-temperature variations is essential.

5 There must be interposed between the pyrometer cold junctions and a cooled outer jacket a high heat-capacity, high-conductivity jacket, surrounded by at least a thin layer of good thermal insulation.

Thermoelectric radiation pyrometers are normally free of the deteriorating influences to which high-temperature contact pyrometers of even the noble-metal thermocouple type are subject. A little regard for the thermodynamics governing their performance has lowered the minimum practical application temperature from 1500 F[6] to about 300 F, and there is no reason to state a minimum for further development. The field of application has been broadened from industrial-furnace applications, requiring about 1 sq in. of "target" area, to include local surface measurement with target diameters of $^1/_4$ in. or less, with repeatability equal to or greater than that of the contact thermocouple. Procedures for improved hot-gas temperature measurement utilizing the thermoelectric radiation pyrometer are in sight. The integration of the laws of monochromatic radiation over the broad spectral band of a suitable window or lens material to create a reproducible high-temperature standard of long life is not beyond the realm of postwar probability.

Acknowledgments

The pioneering in this project was made possible by the efforts and enthusiasm of Messrs. S. S. Stack, W. S. Thompson, N. F. Frischhertz, and R. H. Norris.

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[6] A.S.M.E. Power Test Codes: "Instruments and Apparatus:" Part 3, "Temperature Measurement," chapt. 2, "Radiation Pyrometers."

High-Speed Multiple-Point Potentiometer Recorder for Measuring Thermocouple Temperatures During Test-Plane Flights

By I. M. STEIN,[1] A. J. WILLIAMS, JR.,[2] and W. R. CLARK[3]

Measuring exhaust-gas temperatures in flight-testing large planes requires the use of thermocouples of special design and requires also an instrument capable of recording rapidly, accurately, and directly a large number of temperatures. This paper describes a multiple-point, null potentiometer for this purpose. It accommodates 140 thermocouples and records each temperature in 1.63 sec. Flexible arrangements are provided for switching the numerous thermocouples, and this switching may be operated manually by push-button control and also automatically by the recorder. Operating characteristics and performance in flight tests are discussed.

Introduction

In the early part of December, 1940, representatives of the National Advisory Committee for Aeronautics (N.A.C.A.) Special Subcommittee on Exhaust Gas Turbines and Intercoolers approached the A.S.M.E. Committee on Industrial Instruments and Regulators with a request for assistance in solving certain problems involved in the measurement of high temperatures in high-velocity gas streams. On December 4, 1940, the A.S.M.E. Committee on Industrial Instruments and Regulators appointed a Subcommittee on Exhaust Gas Temperature Measurement to co-operate with the N.A.C.A. subcommittee in solving these problems. The general problem assigned to this special A.S.M.E. subcommittee was made up of two parts, the first one involving the design of the sensitive elements, that is, the thermocouples, and the second one involving the instrumentation. A paper reporting on the first part of the problem has been presented by W. J. King.[4] The present paper deals with the instrumentation part of the problem.

Among the first steps taken by the special A.S.M.E. subcommittee was the circularization of all well-known domestic suppliers of thermocouples and measuring instruments for use with thermocouples, in an attempt to find what, if anything, might be available to meet the particular requirements of the exhaust-gas-temperature problem. In this way some information was obtained on sensitive elements, but all answers to the inquiries with reference to the availability of instruments were negative. As a result of the negative returns to the circular letters, the members of the special subcommittee were urged to encourage developments that might lead to the solution of the instrumentation problem.

The type of instrument needed was a high-speed multiple-point recorder, preferably of the potentiometer type. Fortunately, certain elements essential in the design of such an instrument were either fully or partially developed at the time the request was received from the N.A.C.A. subcommittee, and the availability of these elements shortened considerably the time needed for the development of the complete high-speed multiple-point potentiometer recorder described in the present paper. The paper is limited to a general description of the recorder and its adaptability to the exhaust-gas-temperature problem. A detailed discussion of the electrical circuits involved is reserved for a later paper, as are the applications of the recorder to other problems.

Recorder Requirements in Exhaust-Gas-Temperature Measurements

Number of Points. In each test installation, many thermocouples are involved, and it was felt that one instrument should be capable of recording temperatures of at least 50 thermocouples and preferably of 100 or more.

Ranges and Accuracy Required. The temperatures involved range from those approaching —100 F to those approaching 2000 F. A limit of error of 5 F was desired.

Recording Speed. A relatively high speed of recording was sought because the temperature measurements were to be made during flight tests of planes, and it was desired to record as many temperatures as possible while the plane was gaining or losing altitude at a rapid rate. If possible, the objective was to obtain 20 or more readings in 10 sec, that is, a recording speed of $^1/_2$ sec or less per point. However, it was desired to have the record appear immediately on a printed chart or tape. In other words, the time required for photographic development of charts or tapes was to be avoided. It was recognized that without resorting to photographic recording, it might be necessary to allow more than $^1/_2$ sec per point for recording.

Push-Button Operation. One requirement of the switching facilities was that the arrangement should be such that by means of push-button operation, the recorder could be promptly connected to any one of the many thermocouples and allowed to record continuously the temperature of that thermocouple.

Flight Requirements. Because its principal use would be in connection with flight-testing of planes, it was necessary for the recorder to meet the general requirements of flight instruments; namely, it must operate from the d-c power supply available on planes, its accuracy and reliability must not be unduly influenced by changes in level, by vibration, or by the wide range of ambient temperatures encountered in aerial navigation. In addition, the instrument should not produce radio interference and, naturally, should be as small and light as practicable. Because the instrument was to be used primarily in large planes, it was not considered necessary to reduce the size and weight to the extent that would be essential for use in smaller planes.

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[4] "Measurement of High Temperatures in High-Velocity Gas Streams," by W. J. King, Trans. A.S.M.E., vol. 65, 1943, pp. 421–431.

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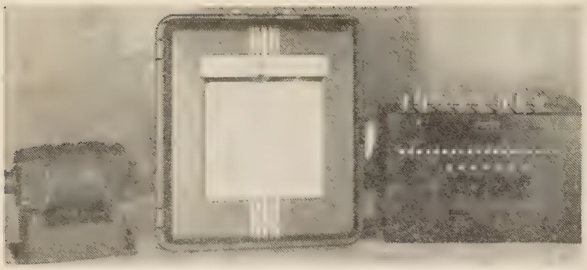

Fig. 1 Flight-Test Recorder and Accessories

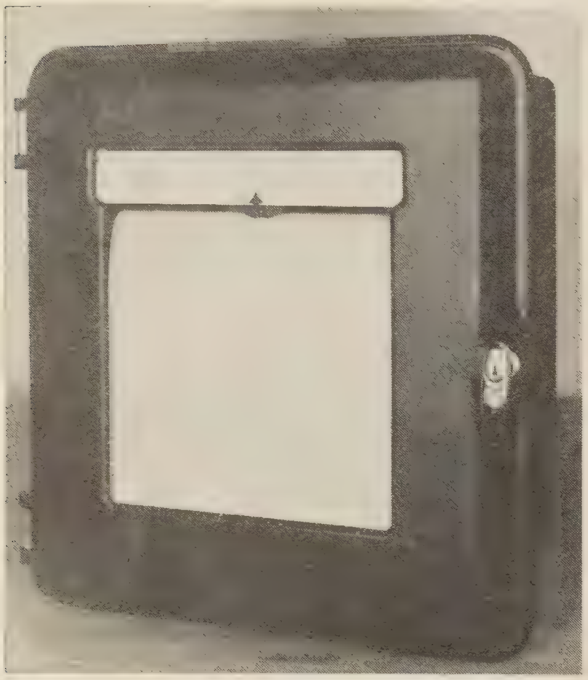

Fig. 2 Flight-Test Recorder, Showing Details of Scales

General Description

A general view of the recorder and its accessories is shown in Fig. 1. The accessories are a multiple-point switch box and a small rotary converter. The weight and dimensions of these items are as follows:

Recorder: Weight, 110 lb; size, $18^3/_8 \times 20^1/_8 \times 12^7/_8$ in. This weight is for a steel case. If aluminum were available, the weight would be 95 lb.

Switch box: Weight, 55 lb; size, $17^7/_8 \times 11^5/_{16} \times 9^{11}/_{16}$ in. This weight includes the 15 plug connectors which have a total weight of 4 lb.

Rotary converter: Weight 35 lb; size, $11^1/_2 \times 9^1/_4 \times 6$ in. This weight is for a 200 v-a unit. If available, a 150 v-a unit would be ample and the weight would be 26 lb.

Power Supply. The recorder proper is designed to operate from a 115-v, 60-cycle supply. The switch box and one or two auxiliary circuits in the recorder are designed to operate directly from the d-c supply available in the plane, usually 24 or 28.5 v. To provide for complete operation from the d-c supply in the plane, the small rotary converter is used to obtain the 115-v 60-cycle supply for the recorder proper. The total power requirement from the d-c supply is 220 w continuous, plus a maximum momentary requirement of 95 w when all stepping switches are being reset simultaneously.

Recorder. A large view of the recorder proper is shown in Fig. 2, and a block diagram of the circuits is shown in Fig. 3. From Fig. 2, it will be seen that the recorder is provided with two ranges, one spanning 800 F (—100 to +700 F) and the other spanning 950 F (1050 to 2000 F). The former is calibrated to correspond to the characteristics of copper-copnic thermocouples, and the latter to correspond to the characteristics of chromel-alumel thermocouples. These particular ranges and calibrations met the requirements of a particular problem. It will be noted that there is a gap of 350 F between the upper end of the lower scale and the lower end of the upper scale. If, in some other problem, temperatures were to be measured in the range where this gap exists, the matter could be handled in one of two ways: Either the two ranges could be expanded somewhat to close the gap, or a third range, closing the gap and overlapping the other two ranges, could be added. The former method would entail some sacrifice in accuracy whereas the latter would not. The range-changing is completely automatic in a

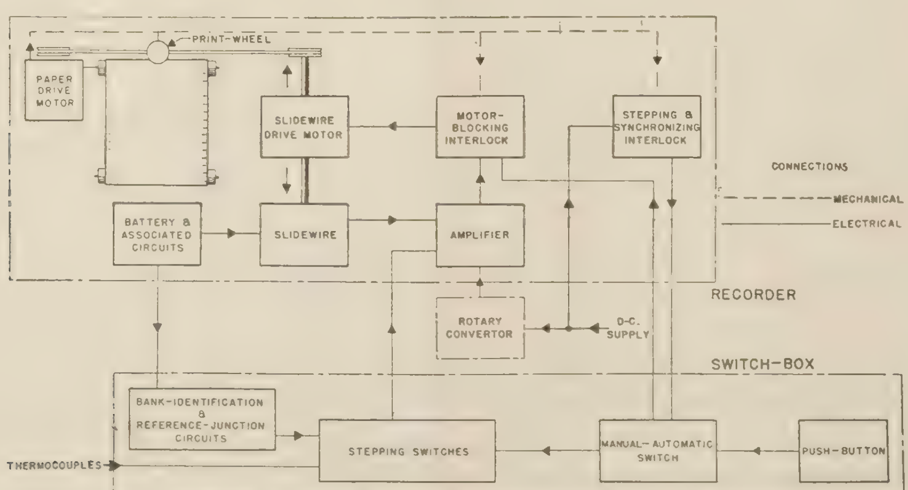

Fig. 3 Block Diagram of Flight-Test Recorder and Accessories

manner which will be described more fully in connection with a discussion of the operating characteristics.

The recorder circuit is that of a null potentiometer. The recorder contains no galvanometer and is not affected by changes in level or by acceleration. It is built to withstand the vibrational effects encountered in plane service, provided only that it is mounted in the usual antivibration frame. The thermocouple voltage is modulated by a vibrating element and then passes through an electronic amplifier. The output of the amplifier is sufficiently powerful to operate a reversible motor, which moves the balancing contact on the slide wire of the potentiometer circuit. Associated with the potentiometer is a resistance-capacitance network, which provides appropriate damping. The vacuum tubes used in the amplifier are all of standard types, thus facilitating replacement should this be necessary. Life expectancy on these tubes is at least 1000 hr.

Switch Box. An enlarged view of the switch box is shown in Fig. 4. The seven telephone-type keys, shown at the center of the panel, control the circuits of 140 thermocouples arranged in banks of 20. As many of the seven banks as desired may be cut out of service by throwing down the respective keys. These may be cut out of service in any order, and even if all but one of the banks are cut out of service, the recorder will continue to recycle on the 20 thermocouples connected to the remaining bank. The bank on which the recorder is operating at any moment is indicated clearly by a signal lamp immediately above the bank key. The longer row of signal lamps above the bank signal lamps indicates the particular thermocouple, in any bank, that is being recorded at the moment. It will be noted that this row of signal lamps contains 21 lamps, although there are only 20 thermocouples in each bank. The additional point on each bank is used for "bank identification" on the recorder chart, as will be discussed under "Operating Characteristics."

The connections to the thermocouples are made by means of the plug connectors at the top of the switch box. Each plug connector handles 10 thermocouples so that there are two plug connectors for each of the seven banks. The "power" plug connector, at the extreme left in Fig. 4, handles the interconnections between the switch box and the recorder proper.

A quick-acting push button, shown below the fifth and sixth bank keys, provides push-button control of the bank switches, as mentioned in outlining the requirements. The telephone key, shown below the first and second bank keys, is a selector switch to give control of the switching to the recorder mechanism or to the push button.

The switch-box housing is lined completely with copper for the purpose of eliminating radio interference. The relays, switches, and other parts used in the switch box have been chosen to withstand satisfactorily the vibration and other conditions encountered in plane service.

The selector switches were chosen carefully to insure reliable operation and long life. They are of a type which has undergone exhaustive development for use in automatic-telephone circuits, and all sliding contact surfaces are heavily gold-plated, Fig. 5.

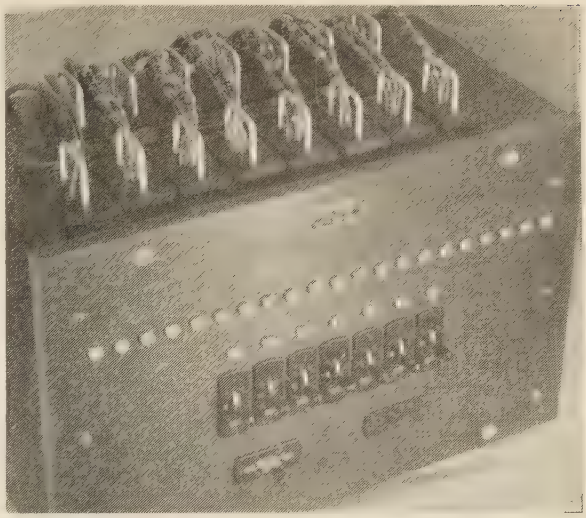

Fig. 4 Switch Box, Showing Panel Details

Fig. 5 Selector Switch

Operating Characteristics

Accuracy, Sensitivity, and Speed. The sensitivity of the recorder and the readability of the chart are better than ± 0.25 per cent of the scale range, and the limit of error is less than ±0.5 per cent of the scale range. In a range of 800 F, 0.25 per cent corresponds to 2 F and 0.5 per cent corresponds to 4 F. The time required for recording on each point is somewhat under 2 sec, specifically, 1.63 sec per point, so that all 147 points (140 thermocouples and 7 bank-identification points) are covered in a period of 4 min. The design of the recorder is such that the constant time-cycle of 1.63 sec is adequate to permit a complete null balance on each point regardless of whether successive points are at nearly the same temperature or are at the extreme ends of the 10-in. chart. The constant time-cycle makes it unnecessary to group the thermocouples according to their temperatures. Hence they may be grouped according to some other scheme as, for example, according to the units on which the temperature measurements are being made.

Fig. 6 is a reproduction of the recorder chart, showing a complete record of all 147 points. It will be seen that all 147 points are obtained without crowding on only an 8 in. length of chart. Because an 11 in. length of chart is always visible, at least one recording for each thermocouple is always visible. This is for a chart speed of 2 ipm. By throwing a lever, the speed of the chart may be reduced to 4 iph. This slower speed would be used in the event that it were desired to record continuously the temperature of one of the thermocouples. A careful examination of the chart will disclose a number of the operating features of the recorder. In examining the chart, it should be remembered that the normal motion is downward so that the record proceeds from the bottom to the top.

At the extreme left of the chart will be found, in order, the

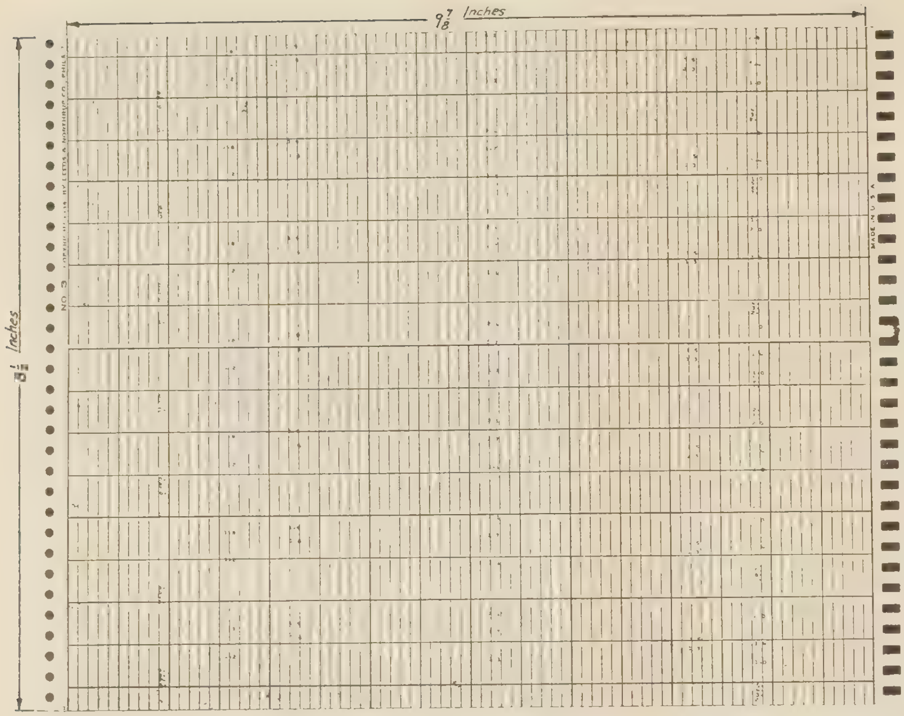

Fig. 6 Reproduction of Recorder Chart

numerals 1 to 7, inclusive; these will be discussed later. The remaining six columns of numerals on the chart correspond to six definite voltages, and the small dots associated with each numeral in any one column should all fall at the same position on the chart. It will be seen that this condition is approached very closely. Now, if the sequence of recording is examined, that is, if the recordings are examined in the order of their numbers, from 1 to 21, for each bank, it will be found that point 21, near the center of the chart, was recorded following a printing toward the upper end of the chart, that is, point 20. On the other hand, it will be found that the recording for point 17 in the same column was made following a printing on point 16, which is much lower on the chart. Hence, the printwheel came in from a higher point on the scale to record point 21 and came in from a lower point on the scale to record point 17, and yet the two points are in nearly perfect alignment. This is a reliable index of the sensitivity and reproducibility of the recorder mechanism. If the first and the last of the six columns are examined, it will be found that, in a number of cases, series of successive numerals appear in the same column. For instance, near the upper part of the last column, points 7, 8, 9, and 10 were recorded successively. The recording for point 6 will be found to be near the other end of the chart. Hence point 7 was recorded after a large travel of the printwheel, whereas points 8, 9, and 10 were recorded without any occasion for travel of the printwheel, but the recording for point 7 is in perfect alignment with the recordings for points 8, 9, and 10. This is an indication that the 1.63 sec allowed for recording each point is ample to provide for complete balance, even when the temperatures to be recorded are widely spaced on the chart.

The balancing characteristic of the instrument may be recorded automatically by adding a pen to the printwheel carriage and by using an increased chart speed. The normal balancing characteristic obtained in this way is shown in Fig. 7, in the curve marked "damped." The balancing characteristic with the damping removed also is shown in Fig. 7. These curves are for a chart traverse approximating 40 per cent of the full width of the chart, and the chart speed was more than 20 times the normal chart speed of 2 ipm. Referring to the damped curve, it will be seen that about $^{2}/_{10}$ of the total cycle of 1.63 sec is required for the linear traverse (motor running at constant speed), and that about $^{3}/_{10}$ of the total cycle is required by the residual oscillations. The time for the linear portion of the traverse increases in proportion to the length of the traverse, and the time for the residual oscillation is a constant. Hence, for a full-scale traverse, one half of the full time-cycle would be required for the linear portion of the traverse. Adding $^{3}/_{10}$ for the residual oscillations gives a total time corresponding to $^{8}/_{10}$ of the complete cycle. As may be seen from Fig. 8, more than 92 per cent of the complete cycle is available for balancing. Similar analysis would show that, without damping, the oscillations would be sustained beyond the end of the cycle and would cause erratic operation.

Referring to Fig. 8, the diagram marked "automatic operation" shows the sequence of operations and the times available for them, when the switching is being done automatically by the recorder. It will be seen that 1.51 sec, or more than 92 per cent of the total cycle of 1.63 sec, are available for balancing.

Bank Identification. The numerals 1 to 7, inclusive, at the left of the chart, Fig. 8, are used for bank identification. It will be noted that each of these identification points occupies a specific position on the chart. The fact that each of these identification markings appears in its own particular position on the chart

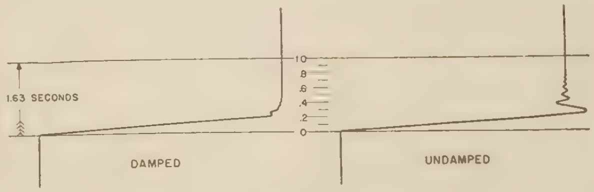

Fig. 7 Balancing Characteristic

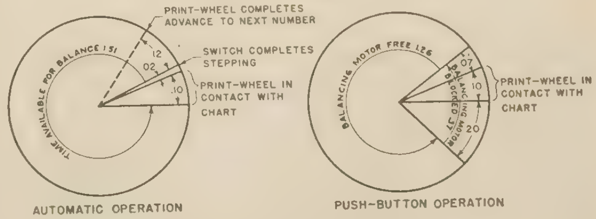

Fig. 8 Sequence of Operations
(Numbers on circle diagrams give time in seconds.)

indicates that there is complete synchronization between the switching mechanism and the printing mechanism. The bank identification is accomplished in the following manner: On bank No. 1 there is no thermocouple connected to the No. 1 switch point. Instead this switch point is connected to an electrical network so arranged that the recorder printwheel prints No. 1 at a specific location on the chart assigned for the identification of the No. 1 bank. Likewise, in bank No. 2 there is no thermocouple connected to the No. 2 point on the switch, the connections to the No. 2 point again being made to a network which causes the recorder printwheel to print the numeral 2 at a place on the chart specifically assigned for the identification of the No. 2 bank; and so on for all seven banks.

Automatic Range-Changing. In discussing the two or more ranges provided for the recorder, it was mentioned that range-changing is completely automatic. This is accomplished in the following manner: Each one of the seven banks of 20 thermocouples may be associated with a different network to provide a different recorder range for each bank, or several of the banks may be connected to provide the same range while the remaining bank or banks may be connected to provide another range. For instance, if most of the thermocouples were to be of the copper-copnic type for measuring temperatures in the range of —100 F to +700 F, the first six banks would be associated with that range on the recorder, and the seventh bank would be associated with the range of 1050 to 2000 F, calibrated for chromel-alumel couples. In that case, as part of the automatic cycling operation, the range remains unchanged when switching between banks, until bank No. 7 is reached and then is changed automatically. When bank No. 7 has been recorded and the recorder is switched to another bank, the range automatically changes back to the lower one. Automatic range-changing applies also to push-button operation.

Automatic Reference-Junction Compensation. The fourteen plug connectors, Fig. 4, which bring the thermocouple leads to the switch box connect to fourteen receptacles which are built into a "thermal-equilibrium" chamber, housed in the top of the switch box. This chamber is provided to keep all of the reference junctions at the same temperature regardless of what that temperature may be. This thermal-equilibrium chamber also houses one or more resistance-type compensators to provide automatic compensation for the temperature of the reference junctions.

Push-Button Operation. The push-button operation feature introduced certain problems that had to be met in a reliable way and without introducing undue complications. Obviously, when the switches are operated by push button, all synchronization between the switching and the recorder printing is lost so that, when the control of the switching is returned to the recorder mechanism, some provision must be made to resynchronize the switching and printing. This is accomplished by an electrical interlock between the printwheel and the automatic selector-switch advancing mechanism. Under automatic operation, the selector switches are always started from the No. 1 position and, by means of the interlock, the switching is not allowed to start until the printwheel is ready to print the numeral "1." By itself, this simple interlock feature is not entirely satisfactory because, even though the first active selector switch were thus synchronized, the remaining ones might be out of synchronism due to having been left in random positions as the result of push-button operation. Of course, after a full cycle, involving the automatic operation of all seven selector switches, the synchronization would be complete, but this might require an interval of 4 min, which is much too long. This delay is overcome by causing all selector switches to "clear" automatically at the time the switching control is transferred from "push button" to "automatic." The "clearing" operation sends all selector switches to their No. 1 positions in a maximum of 6 sec. The automatic synchronization of the printwheel, which follows immediately, requires a maximum of 34 sec, so that the total time for complete synchronization varies from a minimum of less than 1 sec to a maximum of 40 sec.

Although the constant-time printing cycle is advantageous in several respects, its use in combination with push-button operation introduced another problem. Under push-button operation, the selector switches are stepped quickly from one thermocouple to another without waiting for the recorder to balance and print. Hence the recorder might be attempting to rebalance at the instant the printwheel was in contact with the paper, thus causing damage to the chart, the printwheel, or both. This problem was met by interlocking the balancing motor with the printwheel, so that, at the moment the printwheel is in contact with the chart, the balancing motor is prevented from operating. This interlock is effective only when push-button operation is used. It is unnecessary during automatic operation from the recorder (see Fig. 8, diagram marked "push button").

"Grounding" of Thermocouples. In some applications, the thermocouples are not in direct contact with metallic structures and in other applications they are; that is, the thermocouples may be either "grounded" or "ungrounded." For this reason, it was necessary to arrange the measuring circuits so that satisfactory operation could be obtained under either condition. This is a matter of the detailed electric circuits, including their insulation and guarding, which is somewhat beyond the scope of the present paper.

Operating Experience—Availability

Operating experience with this recorder has necessarily been limited, but the observations so far made have been encouraging. During the many months involved in the development of the recorder, the selector switches were subjected to many thousands of operations and not a single failure was recorded. The automatic compensator for reference-junction temperature was subjected to ambient temperatures as low as 0 F, and no error outside the tolerance claimed for the instrument was observed. This test will be repeated with still lower ambient temperature, but the tests already made show a lag of 1 hr between the time a temperature change is made at the exterior and the time when one half of such change is effected in the interior of the "equilibrium chamber." This high degree of lagging is a fairly reliable indication that further lowering of the ambient temperature will not cause appreciable error, under expected service conditions.

The instrument has had approximately 12 hr of actual flight service at altitudes ranging up to 35,000 ft. The ambient temperature at the recorder was as low as —20 F, and this low ambient temperature did not appear to affect the instrument operation in any way. Accelerations caused no apparent effect on any part of the instrument. In fact, no difficulties attributable to the recorder or switch box were encountered during the whole flight-test period. During this flight test, the instrument was not provided with automatic compensation for the thermocouple reference junctions, the temperature of these being fixed by ice baths.

This new recorder is not now in commerical production and is available only in limited quantity for very urgent war problems.

Acknowledgment

Throughout the development work and the field testing, the authors have had the splendid co-operation of Messrs. W. J. King and S. F. Richardson, both of the General Electric Company, and their respective staffs, and wish to express their thanks for this assistance.

High-Temperature-Steam Corrosion Studies at Detroit

By I. A. ROHRIG,[1] R. M. VAN DUZER, JR.,[2] and C. H. FELLOWS[1]

A program was undertaken by The Detroit Edison Company to determine the rate of corrosion and the relative corrosion resistance in an unstressed condition of various materials that could be used in the construction of steam generators, piping, and turbines. Specimens were exposed in a steam atmosphere at 380 psi and at temperatures of 925 F and 1100 F. The investigation required 5 years for completion; during that time 46 different materials were studied, including nickel, nickel-copper alloys, chromium and chromium-nickel stainless steels, medium- and low-alloy steels, carbon steels, and alloy cast iron. The tests were conducted under plant-operating conditions, and samples were exposed in superheated station steam for periods ranging from 4000 to 16,000 hr. Weight-loss, hardness, and metallographic data were obtained after successive exposure periods for many of the samples. Trends in the corrosion rate were plotted for some of the materials. The test results, reported in this paper, indicate that the weight loss of plain-carbon steel exposed to 1100 F steam continues at a high rate, whereas the rate of loss of the alloyed materials decreases with time. The high chromium-nickel and the 12-chromium stainless steels were the most corrosion-resistant. The corrosion rate of 0.50 molybdenum and 1.0 chromium steels compared favorably with that of steels containing 5 chromium. Nonferrous materials were not as corrosion-resistant in high-temperature steam as the high-alloy ferrous materials. Materials which corroded rapidly at 1100 F corroded only slightly at 925 F, and at this lower temperature there was little difference between the rate of corrosion of plain carbon and alloy steels.

Introduction

Experience gained in the operation of two experimental installations utilizing steam at high temperatures located in plants of The Detroit Edison Company led to an investigation of the corrosion effects of high-temperature steam on a selected group of metals for superheaters, piping, and turbines (1, 2).[3] The results of an extended study of this subject are the basis for the present paper.

One of these experimental high-temperature installations was made at the Delray Plant in 1930; it comprised a turbogenerator and superheater. Materials in the superheater and piping were high-alloy stainless steels of high creep resistance that were known not to corrode significantly under the influence of high-temperature steam. In 1934, certain low-alloy steels were used in the tubing of the superheaters installed at the Connors Creek Plant to supply steam at 840 F. The alloys used at Delray, although highly corrosion-resistant, were too costly for this use. At Connors Creek, within the first 2 years of operation there were indications of corrosion resulting from the reaction between the high-temperature steam and the low-alloy steels used there. The experiences with both of these installations and concurrent development in high-creep-resistant low-alloy materials indicated that future installations for large-scale generation could be made with more economical and more suitable materials than those used in the original installations.

Several years earlier, in anticipation of the use of steam above 700 F, in the generation of electric power, some thought had been given to the reaction between steel and high-temperature steam. Laboratory-type studies were begun in 1927 that demonstrated the trend of the reaction between plain-carbon and high-alloy steel and steam at about 1100 F (3). These studies were not comprehensive enough to permit the data obtained to be used in engineering design, and moreover, by 1934, other alloys, especially low-alloy steels, were offered for high-temperature service. With the upward trend in the temperature of steam used in the generation of electricity, it became desirable, not only to examine the new alloys offered for high-temperature service and to evaluate them from a corrosion-resistance standpoint, but also to obtain data that would enable the usefulness of these alloys over long periods of time to be estimated.

Facilities were available for carrying out a comprehensive examination of this steel-steam reaction under what were practically power-plant operating conditions. Accordingly, a program of investigation was discussed with an interested steel manufacturer, the purpose of which was to determine what the effect of high-temperature steam would be on a group of low-alloy materials then being used or proposed for use in high-temperature equipment. The specific object was to determine the rate of weight loss over an approximately 2-year period and all materials would be tested in an unstressed condition.

Information then available gave only the total weight lost for samples exposed to furnace gases or air for short periods of time. Such data gave no indication of what rate of metal loss could be expected over a period of years, or was there any reason to believe that metal losses as the result of oxidation in furnace gases or air would be of the same order of magnitude as losses resulting from corrosion in a steam atmosphere.

At the time this program was being considered, A. A. Potter, H. L. Solberg, and G. A. Hawkins (4) started a similar investigation at Purdue University, utilizing high-temperature high-pressure steam equipment available in their laboratory, in which they planned to obtain fundamental engineering data related to the steel-steam reaction. The two investigations had as their objective the determination of the rate of weight loss of materials subjected to the action of high-temperature steam. They differed, however, in one important respect. The Detroit Edison Company's tests were to be conducted under average steam-power-plant conditions of fluctuating steam temperature and intermittent operation, whereas those at Purdue University were to be conducted under precise laboratory control for much shorter continuous-exposure periods.

The initial group of nine materials, with which the authors'

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[2] The Detroit Edison Company, Production Department, Detroit, Mich. Mem. A.S.M.E.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Research and Power Divisions and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

tests were begun, comprised steels suitable for superheater and piping use and were subjected to 1100 F steam. Additional materials were added during the course of the investigation, which continued for 5 years, from 1937 to 1942; and the scope of the work was enlarged by subjecting some of the materials to 925 F steam. All materials are described elsewhere in this paper.

The corrosion results, some of which were obtained during exposure periods up to 16,000 hr at 1100 F, show that in the case of steels of low alloy content, there is a definite retardation in the rate of metal loss with increase in the exposure period. The behavior of the carbon steels is entirely different, however; with continued exposure, the loss due to corrosion approaches a straight-line function. The results of the 925 F exposure tests do not indicate the marked differences between the alloy steels and the carbon steels that were evident in the tests at 1100 F. In the case of most of the materials tested at 925 F the losses were so small that accurate weight-loss determinations could not be made.

The results of this work agree with those published by Solberg, Hawkins, and Potter (5) for 2000-hr exposure in that the same rate-of-loss trends are shown for similar materials tested at the same temperatures, although the magnitude of the weight losses does not correspond because of the difference in testing procedure. Moreover, these results assume added significance because of the further findings of Solberg, Potter, Hawkins, and Agnew (6) that stress does not influence the corrosion rate of steels in steam at high temperatures. The information should be of value in design problems, inasmuch as the materials were tested under actual operating conditions of temperature variations and for sufficiently long time periods to establish definitely the trend of corrosion of a number of the materials studied.

Materials Investigated

The material studied in this investigation covered a variety of alloys that could be used for power-plant service, such as in superheater tubing, turbine-shell castings, turbine buckets, valves, and miscellaneous high-temperature equipment, such as flow nozzles, thermometer wells, and so forth. Included were both rolled and cast products. All samples were installed inside an experimental high-temperature steam line. In all, 46 separate materials were tested, varying in composition from relatively pure nickel through nickel-copper alloys, chrome-nickel stainless steels, medium- and

TABLE 1 COMPOSITION OF MATERIALS TESTED IN 925 F AND 1100 F STEAM

Sample No.	General type	Chemical composition: C	Si	Mn	P	S	Cr	Ni	Mo	Cu	Fe	Al	Sn	Cb	Pb	Structural condition	Heat-treatment[a]	Duration of test, hr: At 925 F	At 1100 F
	Nonferrous alloys:																		
1	Low-carbon nickel.....	0.03	0.03	0.22		0.005		99.60		0.03	0.06					Cast	A		7461
2	High-carbon nickel....	0.14	0.04	0.11		0.005		99.51		0.06	0.11					Cast	A		7471
3	Magnesium deoxidized nickel..............	0.11	0.08	0.08		0.005		99.62		0.03	0.05					Cast	A		7461
4	Inconel................	0.05	0.14	0.09		0.005	12.86	80.97		0.08	5.78					Rolled	A		7461
5	"K" Monel...........	0.17	0.28	0.38		0.005		66.79		28.59	0.31	2.93				Rolled	A		7461
6	"S" Monel............	0.14	4.14	0.81		0.003		63.91		29.31	1.64					Cast	A		7461
7	Nickel-copper-tin alloy.	..	0.66	0.59	0.69			52.66		32.51	1.50		11.10		0.09	Cast	None	4351	
8	Copper-nickel alloy....	..	2.47	2.39				47.70		46.38	0.61					Cast	OQ, D	4351	
9	Copper-nickel-iron alloy	..	..	0.43				31.12		64.44	3.81					Cast	None	4351	
	Stainless Steels:																		
10	35 Nickel-15 chrome....	0.14	0.70	0.86			15.46	35.56								Cast	A		7461
11	20 Nickel-25 chrome-2 columbium..........	0.39	0.51	0.75			24.00	19.44						2.43		Cast	A		7461
12	20 Nickel-25 chrome, type 310............	0.12	1.09	0.90			25.06	20.18								Cast	A		7461
13	12 Nickel-25 chrome, type 309............	0.09	0.59	0.79			22.46	12.75								Cast	A		7461
14	18 Chrome-8 nickel-2 columbium, type 347.	0.13	0.46	0.71			17.63	8.73						2.08		Cast	A		7461
15	18 Chrome-8 nickel, type 304............	0.07	0.16	0.14			17.60	10.20								Rolled	WQ		15143
16	12 Chrome-2 molybdenum..............	0.05	0.30	0.13	0.024	0.011	12.18	0.10	2.01							Rolled	A		16526
17	12 Chrome, type 420...	0.32	0.22	0.36	0.014	0.008	13.57									Rolled	OQ, D	13165	16526
18	12 Chrome, type 403...	0.10	0.20	0.35	0.017	0.022	12.33	0.18								Rolled	A	13165	16526
19	12 Chrome, type 416...	0.11	0.25	0.35	0.016	0.308	12.27									Rolled	A		16526
20	12 Chrome, type 410...	0.08	..	..			12.23									Cast	OQ, D	4351	
21	12 Chrome, type 403...	0.13	0.24	0.29		0.01	11.87	0.37		0.15						Cast	OQ, D		7461
	Alloy and Carbon Steels:																		
22	5 chrome-moly-silicon..	0.10	1.55	0.30	0.010	0.016	4.83		0.51		Ti					Rolled	A		15143
23	5 Chrome-moly-titanium	0.06	0.41	0.36	0.010	0.008	5.18	0.19	0.58		0.46					Rolled	A		16526
24	5 Chrome-moly........	0.13	0.32	0.45	0.010	0.016	4.96		0.52							Rolled	A	13165	15143
25	2.50 Chrome-moly-silicon	0.11	0.78	0.41	0.012	0.013	2.50		0.50							Rolled	A		15143
26	1.25 Chrome-moly-silicon	0.09	0.78	0.41	0.010	0.013	1.21		0.58							Rolled	A	13165	15143
27	1.25 Chrome-moly-silicon	0.10	1.40	0.41	0.010	0.013	1.24		0.58		V					Rolled	A		15143
28	1 Chrome - vanadium, S.A.E. 6120.........	0.24	0.28	0.65	0.020	0.021	0.94				0.18					Rolled	A		15143
29	I Chrome-moly........	0.10	0.25	0.36	0.011	0.014	0.97		0.55							Rolled	A	13165	
30	Nitrided nitralloy......	0.35	0.25	0.40	0.020	0.020	1.25		0.20			1.20				..	..	13165	16526
31	3 Nickel-moly.........	0.37	0.17	0.65				3.03	0.49							Cast	A		7461
32	1.25 Nickel-chrome-moly	0.41	0.28	0.77			0.73	1.25	0.59							Cast	A		7461
33	1 Nickel-chrome-moly..	0.21	0.26	0.55	0.029	...	0.61	1.04	0.39							Cast	N, D	4351	
34	Silicon-moly..........	0.11	1.35	0.19	0.010	0.012			0.50							Rolled	A		15143
35	Carbon-moly..........	0.15	0.34	0.40	0.023	0.015			0.49							Cast	A, D		16526
36	Carbon-moly..........	0.18	0.35	0.70					0.51							Cast	N, D		16526
37	Carbon-moly..........	0.17	0.15	..					0.56							Rolled	A	13165	
38	Carbon-moly..........	0.26	0.40	0.71	0.035				0.49							Cast	N, D	4351	
39	Low-carbon, S.A.E. 1010	0.11	..	..												As rolled	None	13165	15143
40	Low-carbon, aluminum-killed...............	0.18	0.15	0.51			0.01	0.05	0.01	0.04						Cast	A		7461
41	Low-carbon, high phosphorus..............	0.12	0.10	0.50	0.17											As rolled	None	4351	
42	Medium-carbon, S.A.E. 1035...............	0.34	..	..												As rolled	None	13165	16526
	Cast Iron:																		
43	NiResits cast iron.....	3.13	1.77	1.18			3.80	13.29		6.00						Cast	None		7461
44	NiResist cast iron.....	2.18	2.86	1.54			1.92	15.52		6.05						Cast	None		7461
45	NiResist cast iron.....	2.62	1.83	1.17			2.12	20.07		0.10						Cast	None		7461
46	Invar cast Iron.......	2.04	1.88	0.92			3.58	35.55		0.20						Cast	None		7461

[a] A, annealed; OQ, oil-quenched; D, drawn; WQ, water-quenched; N, normalized.

low-alloy steels to plain-carbon steels, and alloy cast irons. The materials included 9 nonferrous alloys, 12 stainless steels, 17 medium- and low-alloy steels, 4 carbon steels, and 4 alloy cast irons.

Table 1 gives a tabulation of the materials and their chemical composition. This table includes information concerning the structural condition of the materials, that is, whether cast or rolled, the heat-treatment, the temperature at which they were tested, and the total hours of exposure.

Test Apparatus

The test apparatus consisted of an experimental high-temperature steam line supplied with steam from a separately fired superheater of the radiant type, installed in 1929 as experimental equipment at the Trenton Channel Power House. Station steam was supplied to the superheater at 380 psi and 700 F, and it was capable of superheating to 1100 F 6000 lb per hr of steam which passed into a 65-ft experimental section of $5^1/_2$-in-OD tubing. A schematic view of the superheater and piping is shown in Fig. 1. Advantage was taken of the various flanged joints in this piping system to insert test cages holding the specimens where they were subjected to the action of high-temperature steam.

The temperature of steam leaving the superheater was so controlled that at joint *B* Fig. 1 its temperature was 1100 F $\pm$50 F, as indicated by a thermocouple installed at that point. The specimens tested at 1100 F were placed in the steam line between the outlet joint *A* and joint *B*. At joint *C* the temperature was reduced so that the temperature of the steam, as measured by a thermocouple adjacent to joint *D* was 925 F $\pm$20 F. The specimens for test at 925 F were placed in the steam line between joints *C* and *D*. The steam velocities in the two sections of the steam line were low, varying from approximately 8 fps in the 1100 F section to approximately 6 fps in the 925 F section. At joint *E* the steam temperature was further reduced to 700 F at which temperature the steam was used in a turbine.

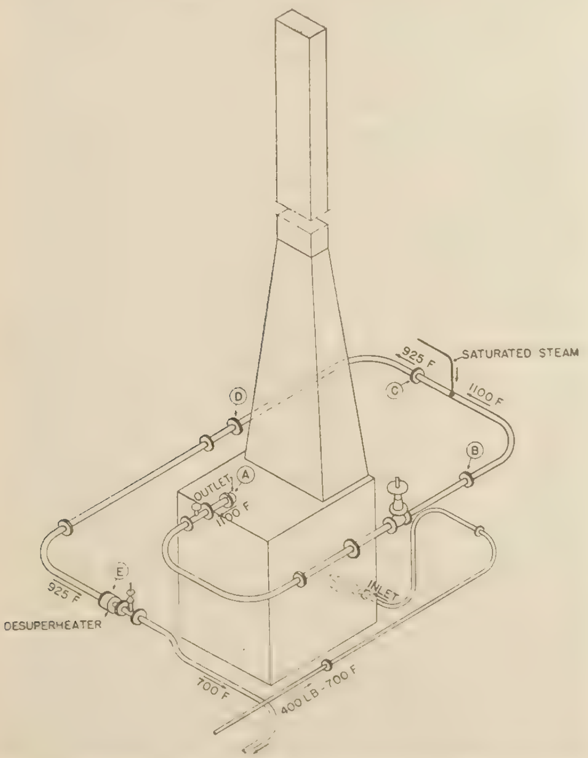

Fig. 1 Isometric View of High-Temperature Superheater and Pipe System, Trenton Channel Plant

Testing Procedure

Corrosion-test specimens and specimens used for initial-weight determinations were made from each material, according to the detail shown in Figs. 2 and 3, respectively. In the preparation of these specimens, each was given a ground finish and every precaution was taken to insure that the surfaces of all specimens were uniform and that the dimensional variations were small. These precautions were necessary since the weight loss of the specimens after exposure was determined by a comparison of the weight of a selected portion of each, after descaling, with the weight of an initial-weight specimen, 1 in. long, similar to that shown in Fig. 3. The diameter of each test specimen was carefully measured and its weight per unit length was determined by reference to the weight and diameter of the initial-weight specimens, which as noted was 1 in. in length.

The specimens were assembled into test cages made according to the detail shown in Figs. 4 and 5. The materials were tested in groups made up of individual cages welded together with short spacer rods. Each such group was placed in the steam line and held in place by tack-welding. At the conclusion of exposure periods of approximately 4000 hr, the group was withdrawn from the steam line, one cage detached, and the remaining cages replaced in the steam line for continued test.

When, at the conclusion of exposure periods, cages or assem-

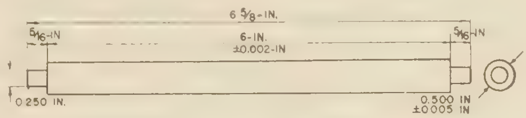

Fig. 2 Corrosion Test Specimen

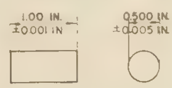

Fig. 3 Specimen for Initial-Weight Determination

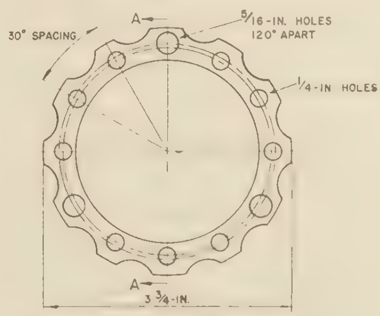

Fig. 4 Detail of Cage Ring

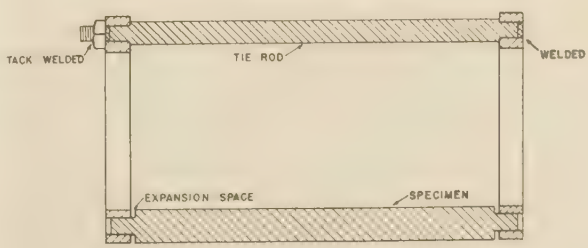

Fig. 5 Cage Assembly, Section A-A

blies of specimens were removed from the steam line, the assembly was taken apart, and the specimens prepared for examination. Two pieces, each approximately $^1/_2$ in. long, one for metallographic examination and the other for hardness determination, were cut from one end of each test specimen. The remaining portion of the specimen, that is, approximately a 5-in. length, was used for weight-loss determinations. In order to insure that the weight per unit length of the weight-loss specimens could be measured with extreme accuracy, the ends of these and of the initial-weight specimens were cut off exactly perpendicular to the longitudinal axis.

The specimens were treated in the following manner in order to determine the weight lost through corrosion.

The weight-loss specimens were first cleaned of corrosion products and then weighed. During the early stages of the investigation, the specimens were descaled with a solution of 100 parts by weight of concentrated hydrochloric acid, 5 parts by weight of stannous chloride, and 2 parts by weight of antimony oxide as inhibitors. Some of the stainless-steel specimens were descaled with concentrated hydrochloric acid containing 1 per cent by weight of quinoline ethiodide as an inhibitor. Later, most of the specimens were descaled electrolytically in a solution of 10 per cent by volume of sulphuric acid with 0.1 per cent by weight of quinoline ethiodide inhibitor. In the electrolytic method, the descaling was done in a lead container that served as the anode, the specimen acting as the cathode. The current was 6v d-c and the current density was about 0.5 amp per sq in. of specimen surface. The weight per unit length of these specimens, after removal of the corrosion products, was then compared with the original unit weights of the specimens.

Hardness readings were made on all the specimens after they were tested as well as on specimens representing the initial condition of all the materials before test. These readings were obtained with the appropriate Rockwell penetrator.

The station steam used in the tests contained no carbon dioxide and less than 0.01 ppm of oxygen. It was considered to be of high purity.

Losses Due to Corrosion

The losses due to corrosion are reported in Table 2, in terms of penetration and as per cent loss in weight, and as penetration only in Figs. 6 and 7.

In reviewing the data based upon the loss in weight, it should be remembered that rather small specimens and small values are being dealt with. In those cases where the materials scaled considerably no difficulty developed in determining the weight lost; but for such materials as did not scale, for example, the stainless steels, the weight losses were very slight and in some instances could not be measured. The relative weight losses of such materials were estimated by comparing the condition of the specimens after test with the condition of other materials of similar composition for which the weight losses resulting from longer exposure periods were known, Table 2 and Fig. 7.

For nine of the materials tested at 1100 F, weight-loss data were obtained after each of three successive exposure periods; for another set of nine materials, weight-loss data were obtained after four exposure periods; and for the last group, containing 19 materials, tested at 1100 F, data were obtained after one exposure period only. Data were obtained for the materials tested at 925 F, after three exposure periods for nine materials, and after one period only for seven materials.

Sufficient data were obtained over these relatively long exposure periods, approximately 4000 hr each, to establish the general trend of the corrosion losses, as is shown in Fig. 8, for 1100 F exposure. Although the curves included are for only 18 of the materials tested, the three general classes of materials investigated, namely, carbon, alloy, and stainless, are represented and their relative rates of corrosion, as established by these tests, are illustrated.

Corrosion at 1100 F. The curves are typical for the ferrous materials tested at 1100 F, in that the high-chromium alloys were the most corrosion resistant. In referring to the 12 per cent chromium steels, it is interesting to note that the free-machining grade sample 19 (type 416) had suffered little surface attack during 16,526 hr of exposure to steam at 1100 F, whereas the other steels in the same class, i.e., sample 17 (type 420) and sample 18 (type 403), had suffered considerable surface pitting. The 18 chromium-8 nickel steel, sample 15 (type 304) was pitted more than the straight-chromium steels.

The 12 per cent chromium-2 per cent molybdenum steel, sample 16, had a higher weight loss than the other samples of the general group in which it would normally be placed, that is, the stainless steels. The higher weight loss of this sample is also shown in Figs. 6 and 7. Improved corrosion resistance in air is reported (7) for 12 per cent chromium alloys containing 1 per cent of molybdenum over that of similar alloys without the molybdenum addition. The results of this test show, however, that in high-temperature steam a 12 per cent chromium alloy containing 2 per cent of molybdenum will scale considerably, whereas a straight 12 per cent chromium steel will exhibit only a slight tendency to scale. Pitting can be expected, however. The behavior of sample 19, which contained 0.30 per cent sulphur, indicates that an increase in the sulphur content of 12 per cent chromium alloys will further improve their corrosion resistance to the extent that pitting may be prevented in high-temperature-steam atmospheres. These tests, however, do not show conclusively that a high sulphur content will improve the corrosion resistance in steam of 12 per cent chromium steels.

Progressive weight losses were not obtained for the nonferrous alloys, samples 1 to 9, inclusive, since only two specimens of each were tested and both specimens were removed from the steam line at the same time. The relative weight losses of these samples in comparison with the ferrous materials tested at the same temperatures and for similar exposure periods are also shown in Fig. 7. The nickel samples 1, 2, and 3 had suffered intergranular attack and the two nonferrous alloys containing a high silicon content, samples 6 and 8, had scaled, sample 6 at 1100 F and sample 8 at 925 F. Furthermore, a layer of copper was found on the surface of both of these alloys after descaling. The high chromium-nickel-ferrous alloys, 10, 11, 12, 13, and 14, together with the chromium stainless steel, sample 19, had the lowest weight losses of the samples tested.

The corrosion losses of the medium- and low-alloy steels did not differ very much in that those steels containing 5 per cent chromium were not greatly superior to those containing 1 per cent chromium; in fact, the losses did not differ greatly from those for 0.50 per cent molybdenum steels. In general, those alloys having silicon contents higher than normal, 0.15 to 0.25 per cent by weight, had approximately the same, and in some cases higher, weight losses than similar alloys of low silicon content. This is contrary to the findings reported for air oxidation, where a high silicon content is beneficial (8).

The highest weight losses were obtained for the carbon steel, 39 (S.A.E. 1010) and 42 (S.A.E. 1035). Sample 40, which was approximately an S.A.E. 1020 steel that had been aluminum-killed had a much lower weight loss when exposed to steam at 1100 F than either the 1010 or the 1035 steels. At 925 F, however, the weight losses of the carbon steels were approximately the same as the weight losses of the low-alloy steels. Materials 5 and 30, which had relatively high aluminum contents, had better corrosion resistance in steam than similar alloys to which only small amounts of aluminum may have been added for deoxidization

TABLE 2 LOSSES RESULTING FROM EXPOSURE TO 925 F AND 1100 F STEAM

Sample No.	General type	Penetration during test periods, in. — 1100 F Test — 4000 hr	8000 hr	12000 hr	15000 hr	925 F Test 13000 hr	Calculated penetration per 10,000 hr, inches at conclusion of test[d] At 1100 F	At 925 F	Per cent loss in weight at conclusion of test[d] At 1100 F	At 925 F
1	Low-carbon nickel		0.00348				0.00467		2.77	
2	High-carbon nickel		0.00257				0.00345		2.06	
3	Magnesium deoxidized nickel		0.01647				0.02208		13.09	
4	Inconel		0.00070				0.00094		0.57	
5	"K" Monel		0.0011				0.00148		0.88	
6	"S" Monel		0.00541				0.00725		4.32	
7	Nickel-copper-tin alloy	...[a]								
8	Copper-nickel alloy	...[a]								
9	Copper-nickel-iron alloy	...[a]								
10	35 Nickel-15 chrome		0.00006				0.00009		0.05	
11	20 Nickel-25 chrome-2 columbium		0.00034				0.00046		0.27	
12	20 Nickel-25 chrome, type 310		0.00008				0.00011		0.06	
13	12 Nickel-25 chrome, type 309		0.00006				0.00009		0.05	
14	18 Chrome-8 nickel-2 columbium, type 347		0.00006				0.00008		0.045	
15	18 Chrome-8 nickel, type 304	0.00074	0.0005	0.00002	0.0003		0.00020		0.20	
16	12 Chrome-2 molybdenum	0.00094	0.0011	0.00186			0.00152		1.50	
17	12 Chrome, type 420	...	0.00004	...		0.00004	0.00005	0.00003	0.32	0.0055
18	12 Chrome, type 403	...	...	0.00002		0.00002	0.00001	0.00002	0.13	0.0050
19	12 Chrome, type 416	...	0.00003	...			0.00003		0.21	
20	12 Chrome, type 410	...[a]								
21	12 Chrome, type 403		0.00047				0.00062		0.32	
22	5 Chrome-moly-silicon	0.0032	0.0035	0.0039	0.0044		0.0029		3.55	
23	5 Chrome-moly-titanium	0.0012	0.0028	0.0034			0.0028		2.74	
24	5 Chrome-moly	0.0022	0.0031	0.0038	0.0036	...[b]	0.0024		2.96	..[b]
25	2.50 Chrome-moly-silicon	0.0037	0.0042	0.0050	0.0051		0.0034		4.16	
26	1.25 Chrome-moly-silicon	0.0034	0.0034	0.0036	0.0041	0.00047	0.0027	0.00035	3.24	0.39
27	1.25 Chrome-moly-silicon	0.0045	0.0054	0.0059	0.0083		0.0055		6.71	
28	1 Chrome-vanadium, S.A.E. 6120	0.0034	0.0038	0.0043	0.0047		0.0031		3.77	
29	1 Chrome-moly	...[c]				0.00048		0.00036		0.38
30	Nitrided nitralloy	0.0017	0.0025	0.0031		0.00089	0.0025	0.00067	2.53	0.72
31	3 Nickel-moly		0.0028				0.0037		2.21	
32	1.25 Nickel-chrome-moly		0.0032				0.0045		2.59	
33	1 Nickel-chrome-moly	...[a]								
34	Silicon-moly	0.0037	0.0041	0.0045	0.0048		0.0032		3.86	
35	Carbon-moly	0.0021	0.0035	0.0041			0.0034		3.29	
36	Carbon-moly	0.0022	0.0033	0.0038			0.0031		3.03	
37	Carbon-moly	...[c]				0.00046		0.00035		0.37
38	Carbon-moly	...[a]								
39	Low-carbon, S.A.E. 1010	0.0107	0.0223	0.0274	0.025	0.00027	0.017	0.00021	20.04	0.22
40	Low-carbon, aluminum killed		0.0032				0.0042		2.51	
41	Low-carbon, high phosphorus	...[a]								
42	Medium-carbon, S.A.E. 1035	0.0041	0.0116	0.0156		0.00043	0.013	0.00033	12.46	0.35
43	NiResist cast iron		0.0033				0.0044		2.60	
44	NiResist cast iron		0.0035				0.0046		2.75	
45	NiResist cast iron		0.0028				0.0037		2.21	
46	Invar cast iron		0.0045				0.0060		3.58	

[a] Tested at 925 F only. Total exposure period was 4351 hr. Weight loss was not measured.
[b] Specimen damaged during cleaning, weight loss not determined.
[c] Tested at 925 F only.
[d] See Table 1 for total time of exposure.

purposes. This is in accord with the findings of Houdremont and Bandel (9) who pointed out that aluminum, like chromium, improves corrosion resistance in an atmosphere of high-temperature steam.

In the case of the chromium-molybdenum high-silicon, the chromium-molybdenum, the chromium-molybdenum-titanium, the silicon-molybdenum, and the carbon steels at 1100 F, the outer scale was brittle and tended to flake off, exposing new surfaces. In general the scale was more brittle on those steels having the higher silicon contents, which explains their relatively higher corrosion rates in steam than in air and indicates also that the mechanism of corrosion is different in steam from that in air. This has been reported also by Solberg, Hawkins, and Potter (5).

The 0.50 per cent molybdenum steels, at 1100 F, compared favorably with the 5 per cent chromium - 0.50 per cent molybdenum and the nitrided nitralloy steels. They were far superior to the plain-carbon steels.

At 1100 F, all of the cast-iron samples had approximately the same weight loss irrespective of their alloy content, Fig. 7.

Corrosion at 925 F. At 925 F, the 12 per cent chromium stainless steels were again the least affected. At this temperature there appeared to be no great difference between the 5 per cent chromium - 0.50 per cent molybdenum steel, the 0.50 per cent molybdenum steels, the plain-carbon steels, the high-phosphorus steel, the chromium-molybdenum-silicon steels, the chromium-nickel-molybdenum steel, and the nitrided nitralloy steel, as each had scaled only slightly, Fig. 11. Although the actual weight losses of the nonferrous materials, 7, 8, and 9, tested at 925 F, could only be estimated (see Fig. 7), metallographic examination indicated that they had corroded more than the 12 chromium steels tested at the same temperature.

Surface Characteristics. The weight losses were so slight, in the case of many of the materials tested, particularly at 925 F, that it is better to consider similar materials, for example, the high chromium-nickel stainless steels, as a group. In view of this, the results of the metallographic examinations are important. On that basis a distinction can be made between the samples when no such distinction can be made on the basis of weight-loss measurements. The photomicrographs, Fig. 9, show that sample 19, the free-machining grade of 12 per cent chromium stainless steel, had greater surface stability than the other steels of the same general type. The 18-8 stainless steel, sample 15, had pitted more than any of the 12 per cent chromium steels but the columbium-stabilized 18-8 stainless steel, sample 14, had not pitted. In general, the steels of this class had good surface stability and corrosion resistance in high-temperature steam.

A uniform and adherent scale formed on all the steels at 925 F except the chromium stainless steels, which had pitted slightly at this temperature. Fig. 12 illustrates the surface condition of representative steels at the conclusion of test exposure periods. The scale found on the specimens, exposed to 925 F steam, is shown to be adherent, Fig. 12 (*a*), whereas in the case of similar steels exposed to 1100 F steam, Fig. 12 (*b*), much of the scale, especially the outer layers, had flaked off. Fig. 12 (*c*) illustrates typical adherent scales formed at 1100 F.

This apparent greater adherence of the scale at 925 F than at 1100 F is one reason for less severe corrosion at the lower tempera-

Fig. 6 Penetration Due to Corrosion of Metals Exposed to High-Temperature Steam for Different Exposure Periods

FIG. 7 PENETRATION DUE TO CORROSION OF METALS EXPOSED TO HIGH-TEMPERATURE STEAM

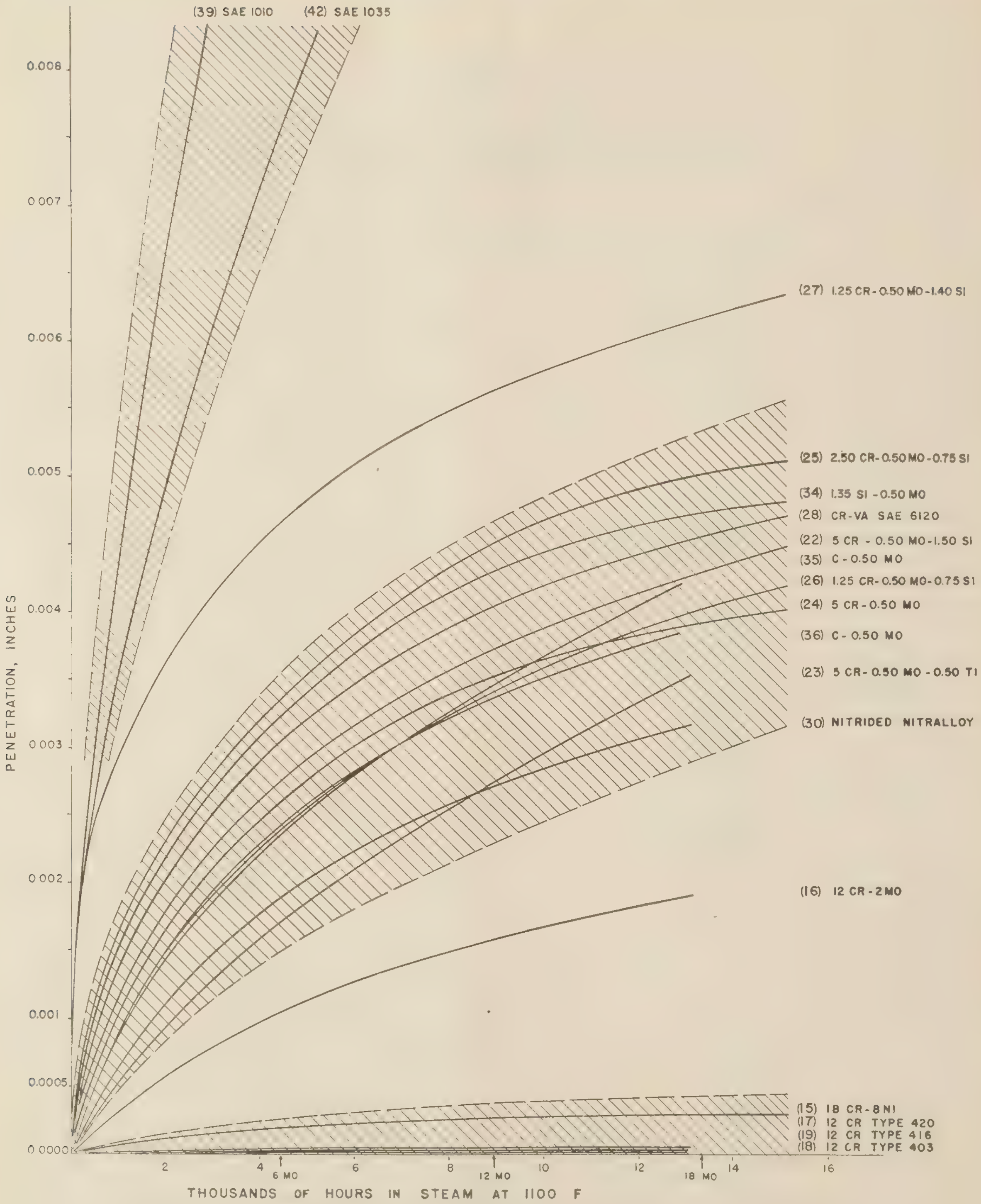

Fig. 8 Effect of Long-Time Exposure to Steam at 1100 F Upon Carbon- and Alloy-Steel Bars

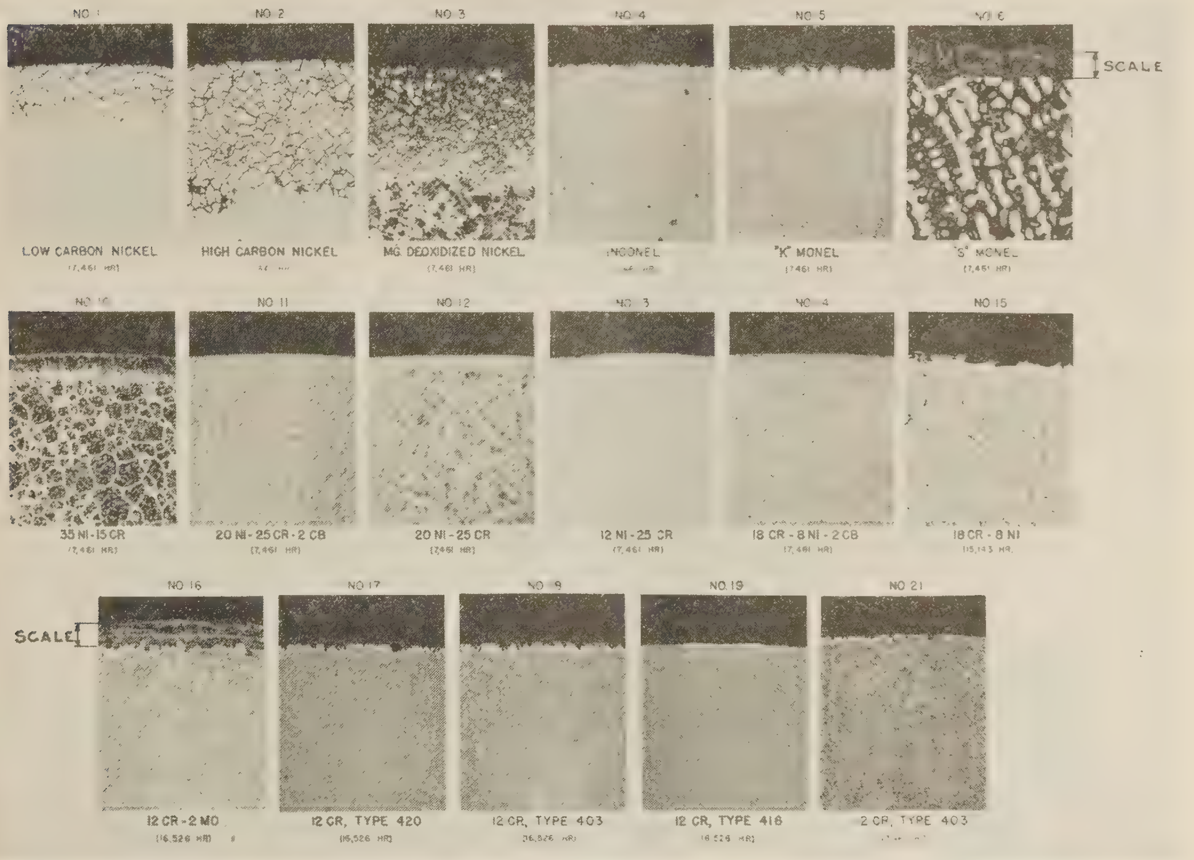

Fig. 9 Photomicrographs Showing Effect of 1100 F Steam on Metals

ture. The greater reaction rate between steam and the materials at 1100 F than at 925 F is a further reason for the difference in corrosion at the two temperatures.

For certain of the steels tested, the scale formed in distinct layers and tended to flake off as a result of temperature changes when the superheater was cooled to room temperature. This may offer an explanation for the relatively high weight loss of these samples in spite of their alloy content. The scale on the carbon steels, except sample 40, formed in thick layers. The scale on certain of the samples was tight and adherent at 1100 F, in spite of temperature variation.

The ability of a metal to maintain an adherent protective scale during high-temperature service is important and apparently can be controlled by using or limiting certain elements in an alloy; as, for example, 0.50 per cent of molybdenum added to a plain-carbon steel greatly improves the corrosion resistance in steam at 1100 F, but 2 per cent of molybdenum added to a 12 per cent chromium steel lowers the corrosion resistance of that steel. Silicon also can be considered in the same way, in that a high silicon content does not improve the corrosion resistance of steels in steam as it does in air.

Hardness. The hardness of all materials tested is given in Table 3. The hardness of most of the nonferrous alloys was reduced by exposure at 1100 F, but at 925 F two of the materials, 7 and 9, increased in hardness owing to the fact that they were precipitation-hardening alloys. Increases in hardness after 1100 F exposure also occurred in several of the stainless steels, namely, 10, 12, 13, 14, and 15.

In general the hardness of the low-alloy, medium alloy, and plain-carbon steels was reduced through exposure to the high-temperature-steam atmosphere. In some instances, however, as shown in Table 3, practically no change occurred. This may be attributed to the heat-treatment that the materials had received.

Indications of slight precipitation reactions were shown by the hardness changes of steels 22, 35, and 37, and hardness increases were found for three of the cast irons, 43, 44, and 46.

Concluding Comment

Although this investigation has been comprehensive in that in it were included a fairly large number of different materials in the several classes of metals proposed for high-temperature service, the conclusions drawn from the results must be qualified by reason of the fact that in many instances only one specimen was examined for a given material at any one temperature and exposure period. To have done otherwise would have provided many more data, but would have required much larger quantities of material and an enormously greater expenditure of man-hours than it appeared could be justified at the time of the inception of the work.

In summarizing the results of this investigation, the following statements can be made:

1 In general, ferrous alloys containing high percentages of chromium alone are corrosion-resistant in steam at 1100 F.

2 Ferrous alloys containing high percentages of chromium and nickel are corrosion-resistant in steam at 1100 F.

3 Nonferrous alloys containing high percentages of nickel

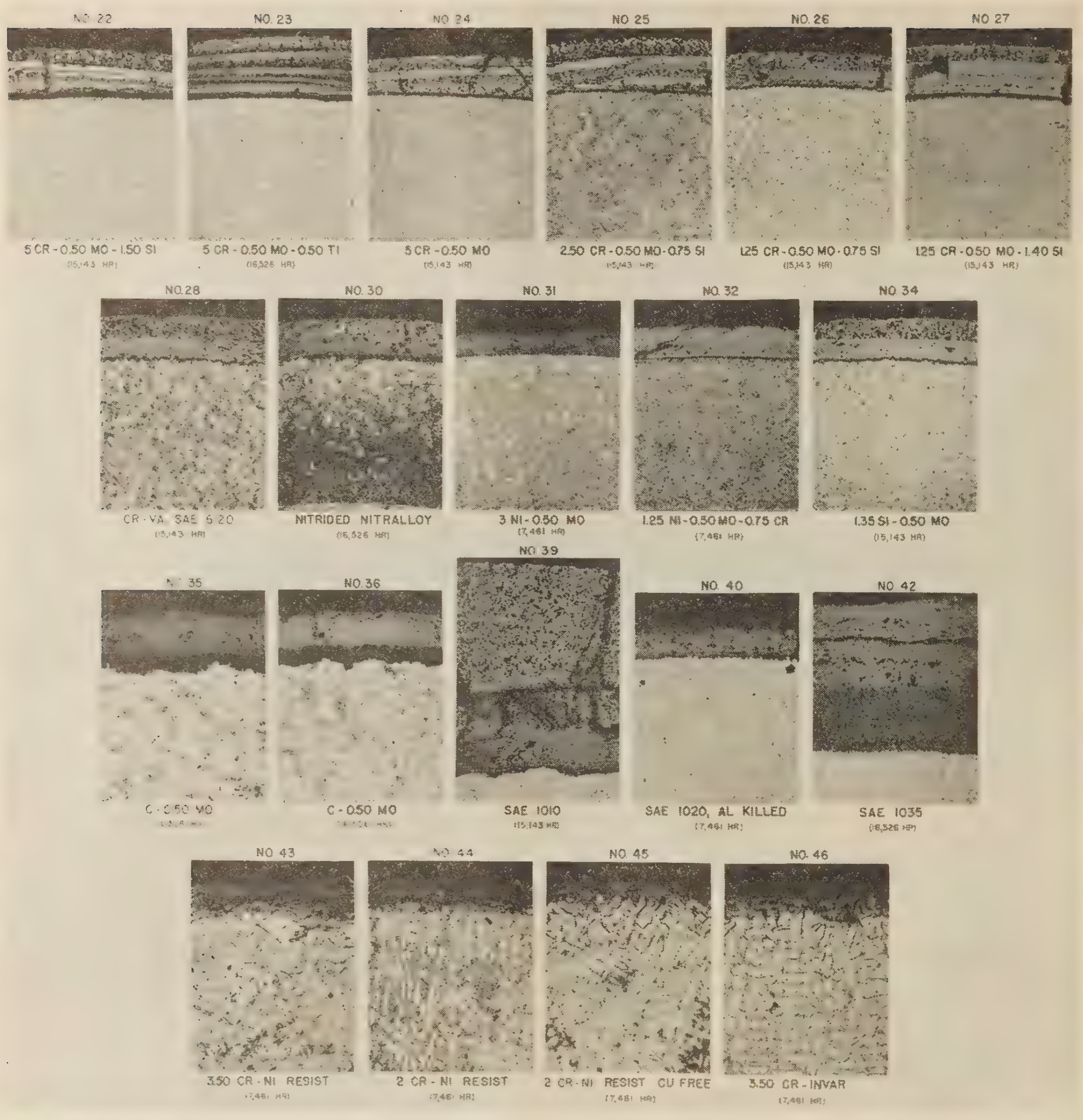

Fig. 10 Photomicrographs Showing Effect of 1100 F Steam on Metals

and copper that were tested in either 925 F or 1100 F steam can be expected to corrode in steam at those temperatures.

4 Although significant differences exist between the corrosion rates of carbon steels and medium- or low-alloy steels at 1100 F, at 925 F only a slight difference exists.

5 At both 925 and 1100 F, many steels that have a relatively low alloy content (0.5 molybdenum, samples 35, 36, 37, and 38) compare favorably in corrosion resistance with steels of medium alloy contents (5 chromium, samples 22, 23, and 24).

6 The formation and maintenance of an adherent scale materially increases the corrosion resistance of materials exposed to high-temperature steam.

7 The trends in the reaction between steam at high temperatures and the various steels for which weight-loss rates were determined, the authors believe, have been satisfactorily demonstrated, and on the basis of these trends the user of such steels may be assured that their rates of corrosion in steam atmospheres do not continue to increase. They flatten out and, as has been shown in the case of many of the low-alloy steels, the rates tend to be materially reduced after 15,000 to 16,000 hr at 1100 F. Some reasons for this have been suggested, but the exact mechanism which underlies these reasons has not been fully explored

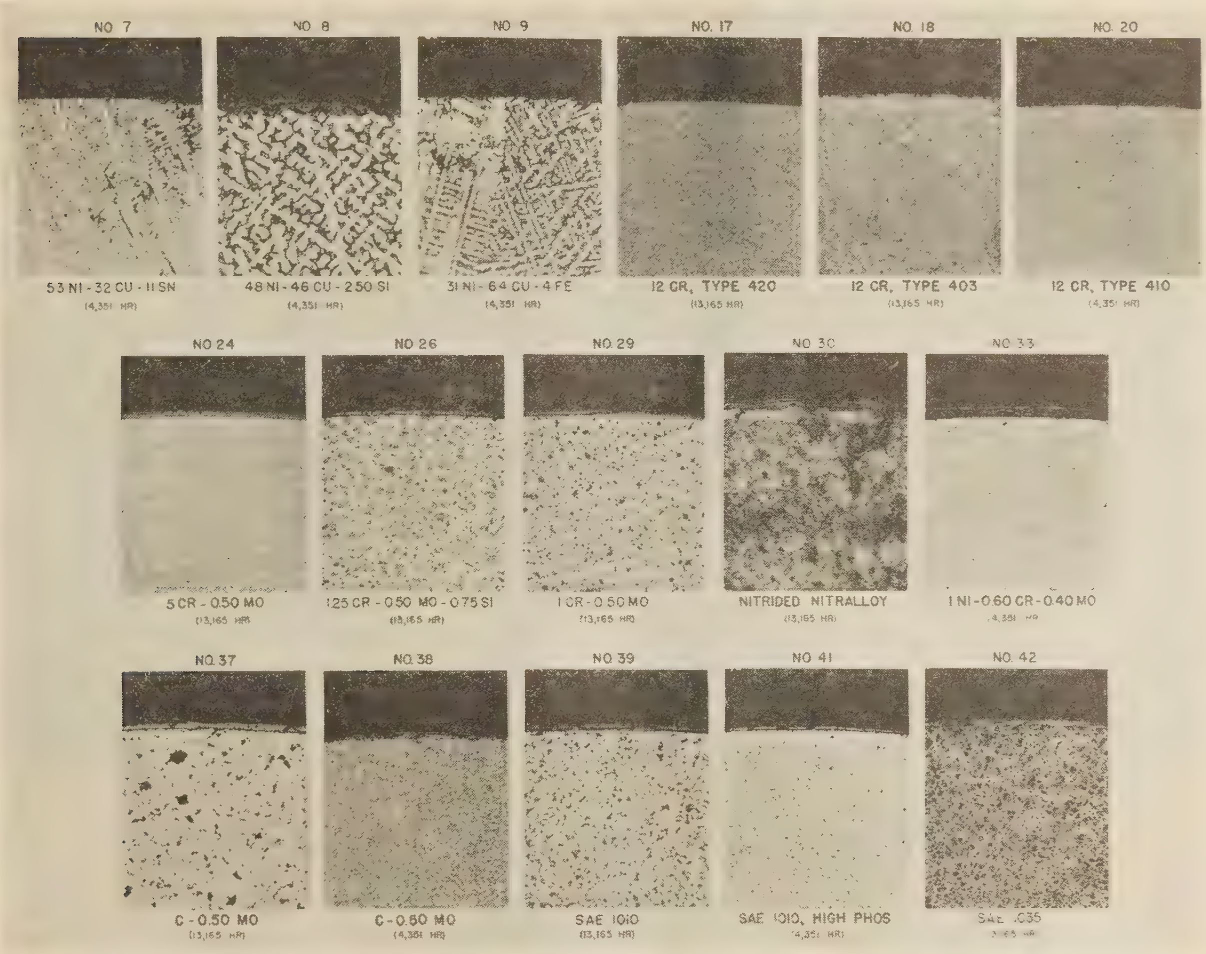

Fig. 11 Photomicrographs Showing Effect of 925 F Steam on Metals

Acknowledgment

The authors wish to acknowledge the co-operation from the Steel and Tube Division of the Timken Roller Bearing Company, the Lunkenheimer Company, the National Tube Company, the International Nickel Company, and the General Electric Company. These companies supplied most of the materials studied. Acknowledgment is made also of the co-operation of the Climax Molybdenum Company in permitting the use of its laboratory equipment for cutting the specimens after each exposure period.

The authors are indebted also to their associates in the company who worked on this study and assisted in the many activities incidental to its progress.

[Fig. 12 and Table 3 will be found on the following page.]

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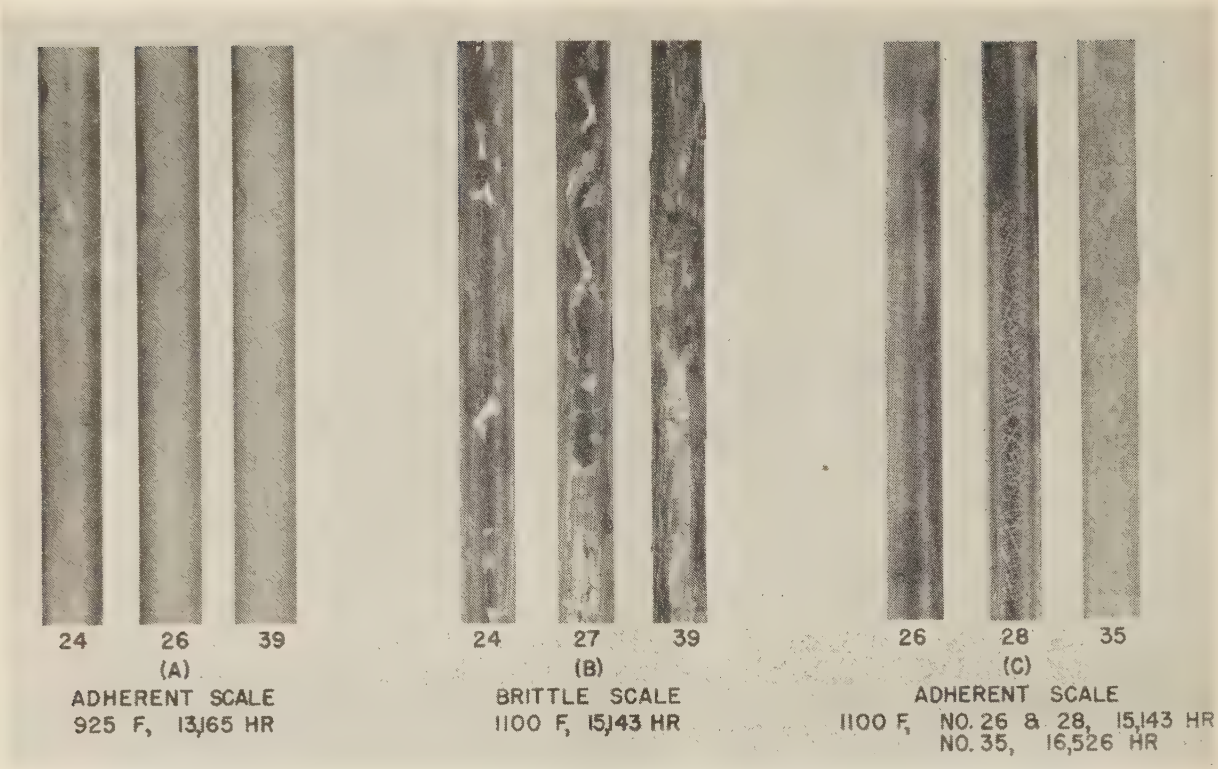

Fig. 12 Surface Condition of Typical Specimens After Exposure to 925 F and 1100 F Steam

TABLE 3 BRINELL HARDNESS BEFORE AND AFTER EXPOSURE TO 925 F AND 1100 F STEAM

		1100 F Tests						925 F Tests				
Sample no.	General type	Before test	After 4000 hr	After 8000 hr		After 12000 hr	After 15000 hr	Before test	After 4000 hr	After 8000 hr	After 13000 hr	
1	Low-carbon nickel	93		90	93							
2	High-carbon nickel	124		87	88							
3	Magnesium deoxidized nickel	157		130	95							
4	Inconel	152		152	155							
5	"K" Monel	326		247	248							
6	"S" Monel	316		274	292							
7	Nickel-copper-tin alloy							247	335			
8	Copper-nickel alloy							331	325			
9	Copper-nickel-iron alloy							97	158			
10	35 Nickel-15 chrome	133		190	190							
11	20 Nickel-25 chrome-2 columbium	171		169	176							
12	20 Nickel-25 chrome, type 310	142		185	186							
13	12 Nickel-25 chrome, type 309	159		183	184							
14	18 Chrome-8 nickel-2 columbium, type 347	149			162							
15	18 Chrome-8 nickel, type 304	121	143	143		147	144					
16	12 Chrome-2 molybdenum	240	194	185		181	174					
17	12 Chrome, type 420	482	207	199		199	193	321	257	234	248	251
18	12 Chrome, type 403	227	214	207		204	188	221	221	221	241	234
19	12 Chrome, type 416	239	207	203		204	204					
20	12 Chrome, type 410							263	217			
21	12 Chrome, type 403	212		220	219							
22	5 Chrome-moly-silicon	158	168	163		160	159					
23	5 Chrome-moly-titanium	123	122	123		119	117					
24	5 Chrome-moly	141	144	146		161	139	138	129	139	132	133
25	2.50 Chrome-moly-silicon	274	174	172		166	166					
26	1.25 Chrome-moly-silicon	161	146	148		143	144	161	174	176	181	186
27	1.25 Chrome-moly-silicon	174	165	158		158	159					
28	1 Chrome-vanadium, S.A.E. 6120	241	190	172		165	159					
29	1 Chrome-moly							129	133	126	126	130
30	Nitrided nitralloy[a]	573	432	415		377	372	534	524	515	472	496
31	3 Nickel-moly	270			187							
32	1.25 Nickel-chrome-moly	277		211	219							
33	1 Nickel-chrome-moly							192	188			
34	Silicon-moly	166	166	161		161	156					
35	Carbon-moly	120	133	135		121	122					
36	Carbon-moly	162	143	134		128	124					
37	Carbon-moly							95	101	102	102	101
38	Carbon-moly							174	179			
39	Low-carbon, S.A.E. 1010	148	78	83		77	68	139	136	124	124	125
40	Low-carbon, aluminum-killed	111		86	85							
41	Low-carbon, high-phosphorus							140	138			
42	Medium carbon—S.A.E. 1035	156	132	125		123	124	157	163	159	158	159
43	NiResist cast iron	174		224	226							
44	NiResist cast iron	145		199	184							
45	NiResist cast iron	133		128	134							
46	Invar cast iron	146		153	155							

[a] The values given are for the nitrided case.

Discussion

E. N. Skinner,[4] Jr., and J. T. Eash.[4] Through their extensive study of the corrosion of metals and alloys by high-temperature steam, the authors have obtained data undoubtedly of considerable practical worth to those concerned with power-plant design, as well as valuable information with respect to the little understood question of high-temperature corrosion.

With regard to their results on the three samples of malleable nickel, it may be useful to state here that experience has made it apparent that nickel which is used at temperatures between 750 and 1400 F should be low in carbon. In this temperature range, it has been found that, regardless of the atmosphere, nickel containing carbon may become embrittled with time due to an intergranular precipitation of graphite. In any application involving the use of wrought nickel at these temperatures, the commercial carbon-free nickel which contains a maximum of 0.02 per cent carbon should be specified.

The results of some laboratory tests that we have conducted on the oxidation of nickel in tap-water steam suggest that the presence of minor quantities of residual elements associated with the production of malleable nickel for widely diverse fields of application has a marked effect on its oxidation resistance in steam. Carbon is particularly harmful in promoting rapid intergranular corrosion such as found by the authors.

Our tests indicate, in addition, that a nickel which does not contain small amounts of residual elements has a low rate of oxidation in steam. Such a product is found in electroplated nickel. Specimens of this type of nickel exposed at 1100 F for 600 hr oxidized at approximately one seventh the rate of a wrought nickel containing about 0.08 per cent carbon; furthermore, this electroplated nickel was completely free from the intergranular attack noted by the authors.

The practical value of electro-nickel in steam applications is a matter which designers of equipment may wish to consider.

H. L. Solberg,[5] G. A. Hawkins,[6] and J. T. Agnew.[7] Through the courtesy of The Detroit Edison Company and the authors the Engineering Experiment Station of Purdue University has been provided with copies of progress reports covering the extensive high-temperature corrosion studies which have been carried on by the company over the past several years. The writers have greatly appreciated the fine spirit of co-operation which has made this information available to us, and wish to congratulate the authors on the excellent job they have done in condensing the contents of these numerous and extensive reports into this paper.

The writers reported[8] in an earlier paper upon certain tests which were carried on with steam at 1200 F, and with the specimens subjected to sharp fluctuations in temperature, in order to determine the resistance of the scales formed on the various alloy steels to spalling under fluctuating-temperature conditions. It was found that, in general, as the chromium content was increased to 4–6 per cent, the scale which was formed became more dense and brittle. The addition of either silicon or aluminum in more than normal amounts tended to aggravate the tendency of the scales to spall under temperature fluctuations. These results are confirmed by The Detroit Edison Company tests. Steels containing more than 7 per cent of chromium formed a tightly adherent thin scale which showed no tendency to spall under the test conditions.

Tests with specimens of various shapes indicated that the spalling action of low-chrome steels was severe on the outside of bars and on flat surfaces but was negligible on the inside surfaces of a tube section of 1-in-ID tubing. It is probable that the scale which is formed on a concave surface is in compression and, consequently, resists spalling caused by temperature fluctuation and furnishes better protection than does the scale which is formed on the outside of bar stock. It is, therefore, possible that the long-time corrosion rates determined in The Detroit Edison tests on $^1/_2$-in. round bars are higher than those which would be encountered in superheater tubing in which the shape of the tube might tend to hold the scale in place.

The Detroit Edison tests show a marked reduction in the corrosion rate of carbon-moly steel, as compared with that of low-carbon steel. The results obtained at Purdue University with a fairly large number of specimens of each type of steel do not indicate a very significant difference in the corrosion rates of these steels. Both cast and rolled specimens were tested at temperatures of 1100 F and higher, under conditions of steady as well as fluctuating temperatures for periods of time which were shorter than those employed by The Detroit Edison Company. Table 5 of this discussion has been compiled from previously published data[8] and summarizes some of these results. All specimens were given the heat-treatment recommended by the producers of the steels used in these tests.

TABLE 4 COMPARISON OF CORROSION OF LOW-CARBON AND CARBON-MOLY STEELS

Length of test, hr	Reference	Temperature, deg F	Low carbon: No. of spedimens	Low carbon: Average penetration, in.	Carbon-moly: No. of specimens	Carbon-moly: Average penetration, in.	Ratio of penetration, carbon-moly to low carbon
500	Table 9[a]	1100	16	0.001317	16	0.00122	0.926
570	Table 10[a]	1200	2	0.0118	2	0.01252	1.06
570	Table 10[a]	1200	2	0.01139	2	0.01049	0.92
1300[b]	Table 12[a]	1200	4	0.0156	4	0.0152	0.975

[a] From reference (8) of this discussion.
[b] Constant temperature for 500 hr followed by intermittent operation for 800 hr.

F. N. Speller.[9] The experimental data from the 5-year period of testing by The Detroit Edison Company is a welcome addition to our limited knowledge of the corrosion of metals in steam at 925 and 1100 F. It would be useful to know whether the corrosion found is uniform or localized in the form of pitting. Considering the relative cost, the favorable corrosion rate of 0.50 molybdenum plus 1 per cent chromium steels is encouraging. In 1933, we published data showing the favorable physical and corrosion-resistant properties of 3 per cent Cr compared with 5 per cent Cr steels. For oil-cracking tubes at 1100 F, the 2-$^1/_4$ per cent Cr, 1 per cent molybdenum steel has since then shown

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[8] "Corrosion of Unstressed Steel Specimens and Various Alloys by High-Temperature Steam," by H. L. Solberg, G. A. Hawkins, and A. A. Potter, Trans. A.S.M.E., vol. 64, 1942, pp. 303–316.

[9] Metallurgical Consultant, Pittsburgh, Pa.

very promising results in practice. In fact, experience in high-temperature oil-cracking around 1100 F affords experience on the relative strength of these steels over long periods at elevated temperatures that should be quite useful in the design of high-pressure steam boilers.

Authors' Closure

Respecting the comments of Messrs. Skinner and Eash, we are glad to have an explanation of the intergranular deterioration that was observed in the three samples of malleable nickel. The explanation that this was due to intergranular precipitation of graphite agrees with the results of our tests in that the lesser amount of intergranular deterioration was found in the sample which had the lowest carbon content.

It is interesting also to learn that the corrosion rate of electroplated nickel exposed to 1100 F steam for 600 hr was approximately one seventh that of wrought nickel containing 0.08 per cent carbon. The weight loss for electroplated nickel, as calculated from the factor given by Skinner and Eash in their 600-hr test, i.e., one seventh the value for wrought nickel, would indicate that, although the corrosion rate appeared to compare favorably with that found for the chromium and chromium-nickel stainless steels, on the basis of a 7461-hr test, the corrosion rate of electroplated nickel might not be found to compare so favorably with that for these stainless steels.

The comment of Messrs. Solberg, Hawkins, and Agnew that scale on a concave surface is more protective than on a convex surface and, consequently, the results given in the paper may be somewhat high, as compared with corrosion rates that would be encountered in the inner surface of tubes in service, is probably true. With respect to the relative rates of corrosion instead of the actual rates under service conditions of the 46 different materials, it is felt that, since each material was tested by the same type of specimen, i.e., $^1/_2$-in-diam bar specimens, the comparative tendency of these materials to corrode and scale has been established. To interpret such data in terms of corrodibility of tube forms in service, will require more such studies as those carried on at Purdue University. There are many applications in power plants, such as turbine buckets, various sections of turbine shells, valve-trim materials, and other uses in which the service conditions would be similar to those under which our tests were made.

In contrast to the Purdue test results, our tests have shown a marked reduction in the corrosion rate of carbon-molybdenum steel as compared with that of carbon steel. Our finding is based upon consistent test results obtained over a period of approximately 16,000 hr of exposure to 1100 F steam, during which time four different specimens of each sample were examined. Our shortest exposure period was approximately 4000 hr, whereas the Purdue results are based upon exposure periods of only 500 hr in 1100 F steam. To compare the test results of the two separate investigations, the shape of the "rate-loss" curve would have to be determined between 0 and 4000 hr of exposure in steam at 1100 F. This point is significant. Furthermore, the metallographic examination made of our test specimens at the conclusion of each 4000-hr exposure period showed a marked difference in the thickness of the scale on carbon steel, as contrasted with carbon-molybdenum steel. The carbon-steel specimens consistently had a much thicker and less adherent scale than the carbon-molybdenum steels. Two different carbon steels and two different carbon-molybdenum steels were each tested at 1100 F. At 1200 F there may be very littly difference in the scaling resistance of carbon steel and carbon-molybdenum steel as shown in Table 4, given in the Purdue comments. At 925 F we found practically no difference between the scaling rate of carbon and carbon-molybdenum steels.

Our test results indicate that carbon content is a significant factor in the corrosion rate of carbon steels exposed to steam at 1100 F. Consistently higher weight losses and thicker scale were found for low-carbon steel than for medium-carbon steel. We did not study the effect of carbon content as related to the corrosion resistance of carbon-molybdenum steel in steam at 1100 F.

Mr. Speller's question concerning whether the corrosion found is uniform or localized in the form of pitting is answered as follows: In general the corrosion found on the carbon steels and the low-and medium-alloy steels was quite uniform. Whenever the surface of the specimens was completely covered with a scale of measurable thickness, which could be seen easily at a magnification of 100 diam in a cross-sectional specimen, we prefer to think of the corrosion as being of a uniform type provided that no intergranular attack has occurred. In the case of the cast materials, considerable surface roughness was observed beneath the scale which is believed to be due to segregation in the cast alloy. Similar materials when tested in the wrought condition did not show such roughening of the surface beneath the scale.

Pitting of the surface as a result of exposure to 1100 F steam was found in the stainless materials, particularly in the 18 Cr-8 Ni without columbium, and in the 12-Cr steels with the exception of Sample 19 which was a Type 416, 12-Cr steel and which had shown very little surface attack after 16,526 hr of exposure. Pitting of the surface was also found in Inconel and "K" monel after 7461 hr of exposure. Intergranular attack at the surface was found in the nickel samples and, as mentioned by Messrs. Skinner and Eash, this was caused by intergranular precipitation of graphite at the test temperature.

No tests were made of 1 per cent chromium, 0.50 per cent molybdenum steels in 1100 F steam. Two steels of that general type were tested but contained a high silicon content and are, therefore, not comparable to such steels with normal silicon. A 1 per cent chromium, 0.50 per cent molybdenum steel was tested in 925 F steam for 13,165 hr. At that temperature its corrosion rate appeared to be about the same as that of 0.50 per cent molybdenum steel and plain-carbon steel, all of which had scaled approximately the same amount. In steam at 1100 F, however, the beneficial effect of 1 per cent of chromium would probably enhance the corrosion resistance of 0.50 per cent molybdenum steels. No tests were made of steel containing $2^1/_4$ per cent chromium, 1 per cent molybdenum, but from its favorable showing in oil-cracking applications at 1100 F, as reported by Mr. Speller, it probably would be suitable for many steam-power-plant applications.

The Corrosion of Alloy Steels by High-Temperature Steam

By G. A. HAWKINS,[1] J. T. AGNEW,[2] and H. L. SOLBERG[3]

This paper presents additional test results obtained at the Engineering Experiment Station of Purdue University dealing with the relative resistance to corrosion by steam of unstressed specimens of various alloy steels at temperatures between 1000 F to 1800 F. All of the steels tested, except for the very high chromium-nickel alloys, show rapid corrosion beyond a limiting temperature which increases with chromium content. Data are presented relative to the chemical composition of the scale layers formed during tests at 1500 F and 1800 F.

Inasmuch as steam temperatures in modern power plants and in some of the process industries are approaching and in some cases exceeding those which may be used for the production of hydrogen by reaction between steam and iron, an investigation was undertaken at Purdue University of the corrosion by steam of the various steels which are available for high-temperature service. Apparatus has been constructed and techniques developed for measuring the amount of corrosion on unstressed specimens due to steam temperatures up to approximately 1800 F.

Description of Apparatus

A complete description of the apparatus used in this investigation together with the temperature-control circuit has been published (1).[4] Briefly, steam was passed through a counterflow gas-fired steel-tube superheater, after which it flowed through an electrically heated superheater constructed from 25-20 stainless-steel pipe $^5/_8$ in. OD $\times$ $^3/_8$ in. ID. It was then admitted to a reaction chamber made from a piece of 2-in-OD double-extra-heavy 25-20 stainless-steel pipe 10 ft long. A plate was welded over one end of the reaction chamber and was drilled and tapped for the superheater-inlet pipe. The other end of the chamber was threaded and closed with a pipe cap made of 7-Cr steel to facilitate insertion and removal of the test specimens. The reaction chamber was heated externally to the same temperature as the steam by means of two main and two guard heaters constructed from chromel wire. The reaction chamber was suitably insulated. Pyod-type chromel-alumel thermocouples were used to measure the temperatures along the reaction chamber. The steam discharging from the reaction chamber was condensed in a coiled-copper-tube heat exchanger and weighed.

Test Specimens and Testing Procedure

The specimens were machined to a length of 6 in. and a diameter of $^1/_2$ in. After the specimens had been machined, the surfaces were sandblasted in order to obtain identical surface conditions. After sandblasting, the specimens were weighed and placed in small cages made of stainless-steel welding wire and inserted into the reaction chamber, which had been brought up to the desired operating temperature. After remaining in the reaction chamber in contact with flowing steam for the desired length of time, the samples were withdrawn. The outer scale layers were removed mechanically in order to preserve the scale for chemical analysis. All of the remaining scale was removed by making the specimen the cathode in an electrolytic cell containing a 10 per cent solution of sulphuric acid with 1 g per liter of quinoline ethiodide as an inhibitor and with a current density of approximately 1 ampere per sq in. (2). The stripped samples were reweighed and the loss in weight determined by difference was taken as the measure of the amount of corrosion.

Corrosion Results

Samples of S.A.E. 1010, 3 Cr-Moly, 4-6 Cr-Moly, 7 Cr-Moly, 9 Cr-Moly, 12 Cr, 18-8 Cb (Stabilized), 21 Cr, 27 Cr, 25-20, and 25-15-2W steels were subjected to steam temperatures ranging from 1400 F to approximately 1800 F for 500 hr. The results for the various tests are given in Table 1. These data are represented in Fig. 1 by smooth curves drawn through the average of the test points. These curves are based upon the data presented in this paper as well as the results which have been reported in earlier papers (1, 2, 3).

The S.A.E. 1010, 3 Cr-Moly, 4-6 Cr-Moly, 9 Cr-Moly, 12 Cr-Moly, and 18-8 steels show a definite breaking point at a particular temperature. This same fact would probably have been observed for the 21 Cr, 27 Cr, 25-20, and 25-15-2W if the tests had been carried out at high temperatures. Only two points are available for the 7 Cr-Moly steel at the higher temperatures; hence no general curve is presented. The available data on the 7 Cr-Moly steel lie close to the 9 Cr-Moly curve.

An attempt was made to establish a simple single equation which could be used to compute the loss in weight for the steels tested for 500 hr at the various test temperatures. This was not possible due to the fact that the high-chromium steels show little corrosion until a certain temperature is reached above which an abrupt increase in the corrosion rate occurs at higher temperatures, while the low-chromium steels show a gradual increase in the rate of corrosion with increasing temperature. In order to use one equation, it was necessary to employ a very complex relation which would not be satisfactory from an engineering standpoint. As a result, a family of equations was established of the form

$$C = a(x - b)^d \qquad [1]$$

wherein the constants, a, b, and d differ for each steel and are given in Table 2. The symbol C is the corrosion or loss in weight as a percentage of the initial weight. The term x is equal to the following

$$x = \frac{t - 1000}{100} \qquad [2]$$

where t is the temperature under consideration in degrees F. This family of equations represents the test results within the

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[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Special Research Committee on Critical-Pressure Steam Boilers and the Power Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

TABLE 1 CORROSION OF STEEL BARS IN CONTACT WITH STEAM FOR 500 HR

Steel	Chemical analysis, ladle, per cent												Steam temp, F	Loss in weight, per cent of original weight
	C	Mn	P	S	Si	Cr	Ni	Mo	Cb	W	Cu	N		
S.A.E. 1010	0.08	0.30	0.017	0.034	....	.....	.....	....	....	....	....	.....	1736	100.0
													1498	62.4
													1400	34.1
3 Cr-Moly	0.11	0.51	0.014	0.017	0.36	2.95	.....	0.98	....	....	....	.....	1736	74.8
													1501	44.3
													1400	21.7
4-6 Cr-Moly	0.11	0.33	0.020	0.027	0.28	5.66	0.22	0.50	....	....	....	.....	1772	77.7
													1736	70.0
													1501	28.8
													1400	19.1
7 Cr-Moly	0.11	0.43	0.012	0.011	0.92	7.33	.....	0.50	....	....	....	.....	1772	57.4
													1506	12.2
9 Cr-Moly	0.11	0.38	0.010	0.016	0.27	9.00	.....	1.22	....	....	....	.....	1772	62.7
													1728	60.1
													1680	41.3
													1550	28.3
													1506	14.6
													1475	1.9
12 Cr	0.10	0.52	0.014	0.015	0.32	12.92	0.12	....	....	....	....	.....	1680	23.6
													1550	0.0
													1508	0.0
													1475	0.0
18-8	0.06	0.50	0.03	0.03	0.61	17.75	9.25	....	....	....	....	.....	1675	0.0
													1560	0.0
													1470	0.0
18-8 Cb (Stabilized)	0.07	0.36	0.015	0.012	0.39	18.62	9.90	....	1.11	....	....	.....	1765	11.5
													1728	12.3
													1675	0.0
													1560	0.0
													1508	0.0
													1470	0.0
21 Cr	0.11	0.39	0.016	0.016	0.42	21.45	.....	....	....	....	1.18	.....	1670	0.0
													1570	0.0
													1480	0.0
27 Cr	0.11	0.48	0.014	0.014	0.38	26.02	0.20	....	....	....	....	0.101	1670	0.0
													1570	0.0
													1480	0.0
25-20	0.07	1.62	.....	.....	0.34	24.45	20.30	....	....	....	....	.....	1751	0.0
25-15-2W	0.10	1.75	.....	.....	0.56	24.18	14.34	....	....	2.06	....	.....	1751	0.0

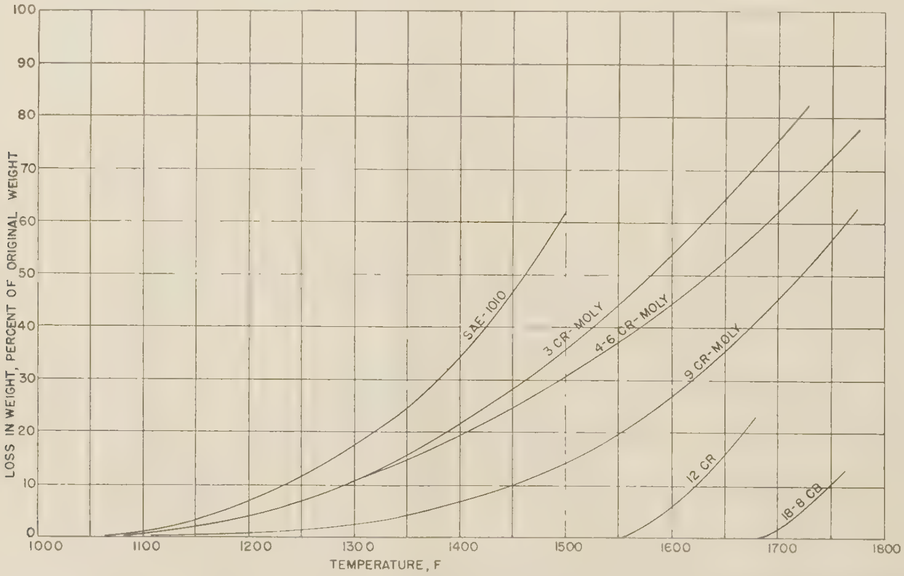

Fig. 1 Corrosion of Steel Bars in Contact With Steam for 500 Hr at Various Temperatures

range of temperatures investigated and should not be used for temperatures in excess of the test limits.

TABLE 2 CONSTANTS FOR USE IN CORROSION EQUATION $C = a(x - b)^d$

Steel	Constants		
	a	b	d
S.A.E 1010	1.1	0	2.5
3 Cr-Moly	0.93	0	2.27
4-6 Cr-Moly	1.0	0	2.13
9 Cr-Moly	0.07	0	3.33
12 Cr	16.45	5.5	1.34
18-8 Cb (Stabilized)	15.40	6.75	1.33

Chemical Analysis of Scale Layers

The steels listed in Table 3 were tested at temperatures in excess of 1700 F for a period of 500 hr.

An extremely thin tightly adhering layer was found on all of the scales next to the parent metal. It was found impossible to remove enough of this inner layer to analyze it.

The scale formed on the 3 Cr-Moly steel consisted of a thick porous and brittle outer layer and a dense brittle middle layer The outer and middle layers were separated mechanically by using a sharp probe. In many cases sections of both layers stuck together. Although the color of the middle and outer layers was

TABLE 3 CHEMICAL COMPOSITION OF VARIOUS SCALE LAYERS FORMED ON STEEL SPECIMENS IN CONTACT WITH STEAM AT APPROXIMATELY 1700 F FOR 500 HR

	Chemical analysis, ladle, per cent								Chemical composition of scale layers, per cent: Middle layer			Outer layer			Steam temp,
Steel	C	Mn	P	S	Si	Cr	Ni	Mo	Cr	Si	Mo	Cr	Si	Mo	F
3 Cr-Moly	0.11	0.51	0.014	0.016	0.36	2.95	....	0.98	7.44	0.75	2.34	0.33	0.03	0.05	1736
4-6 Cr-Moly	0.11	0.33	0.020	0.027	0.28	5.66	0.22	0.50	11.61	0.57	1.12	0.40	0.03	0.03	1736
9 Cr-Moly	0.11	0.38	0.010	0.016	0.27	9.00	....	1.22	11.94	1.48	0.97	0.34	0.01	0.03	1728

TABLE 4 CHEMICAL COMPOSITION OF VARIOUS SCALE LAYERS FORMED ON STEEL SPECIMENS IN CONTACT WITH STEAM AT APPROXIMATELY 1500 F FOR 500 HR

	Chemical analysis, ladle, per cent								Chemical composition of scale layers, per cent: Middle layer			Outer layer			Steam temp,
Steel	C	Mn	P	S	Si	Cr	Ni	Mo	Cr	Si	Mo	Cr	Si	Mo	F
3 Cr-Moly	0.11	0.51	0.014	0.016	0.36	2.95	....	0.98	5.08	0.58	1.65	1.07	0.06	0.15	1501
4-6 Cr-Moly	0.11	0.33	0.020	0.027	0.28	5.66	0.22	0.50	9.38	0.49	0.84	0.32	0.03	0.025	1501
7 Cr-Moly	0.11	0.43	0.012	0.011	0.92	7.33	....	0.59	8.86	1.15	0.70	1.07	0.17	0.025	1506
9 Cr-Moly	0.11	0.38	0.010	0.016	0.27	9.00	....	1.22	12.35	0.26	1.54	2.06	0.09	0.29	1506

the same, it was very easy to distinguish between the two. The dividing plane between the layers was very sharp, due to the fact that the exposed surfaces of the particles which composed the middle layer reflected ordinary light differently from those contained in the outer layer. Samples of the outer and middle layers were analyzed for Cr, Mo, and Si. The results are given in Table 3.

The 4-6 Cr-Moly showed a scale-layer formation similar to the scales formed on the 3 Cr-Moly steel. The outer layer was very brittle and readily separated from the middle layer. The outer layer of scale formed on the 9 Cr-Moly steel was quite brittle and was easily separated from the adjacent layer. The middle layer was very dense and hard to remove. It was not possible to distinguish the middle layer from a layer next to the parent metal, if such a layer existed. The percentage of chromium in the outer scale layer for the 4-6 Cr-Moly and 9 Cr-Moly steel was larger than for the 3 Cr-Moly steel.

The steels listed in Table 4 were held in a steam atmosphere at 1500 F for 500 hr. From visual observations the scales formed on the 3 Cr-Moly, 4-6 Cr-Moly, and 9 Cr-Moly samples appeared to be quite similar to those formed in the same type of steel specimens tested at 1700 F for 500 hr. The 7 Cr-Moly and 9 Cr-Moly scales were quite similar. The results differ materially from those obtained on the 500-hr test at 1700 F.

The data presented in Tables 3 and 4 indicate that a concentration of chromium, silicon, and molybdenum has occurred in the middle scale layer. The increase in concentration for the middle layer may be clearly brought out by studying the ratio of the percentage of the alloying element in the scale layer to the percentage in the base steel. These data are presented in Tables 5 and 6. For the chromium the relative concentration increase ranged from 1.21 to 2.52, for silicon from 0.96 to 5.48, and for molybdenum from 0.79 to 2.39.

In every case the alloy content in the outer layer is lower than in the middle layer. This agrees with many of the oxidation theories which have been advanced, in that since the outer layer is composed chiefly of iron, the iron diffused through the alloy-rich middle layer.

These results tend to indicate that in all probability the steam-oxidation resistance of the alloy steels tested is due to thin and very dense tightly adhering inner layer of scale and a dense middle layer which contains a higher concentration of the alloying agents than is present in the base steel. An outward diffusion of the alloying agents from the base steel to and into the inner and middle layers must occur. In order to have a steam-corrosion-resisting steel there must be sufficient amounts of the alloying agents to combine with the oxygen diffusing inward to form the dense protective inner and middle layers. These general conclusions are in agreement with those found by White, Clark, and McCollam on air-oxidation tests (4).

It is thus possible that long-time tests would show less difference in the total amount of corrosion for medium- and high-alloy steels than has been found by conducting tests of from 500 to 2000 hr duration.

TABLE 5 RELATIVE CONCENTRATION OF ALLOYING ELEMENTS IN MIDDLE SCALE LAYER FOR SPECIMENS IN CONTACT WITH STEAM AT APPROXIMATELY 1700 F FOR 500 HR

	Ratio of percentage of alloying agent in middle layer and base steel		
Steel	Chromium	Silicon	Molybdenum
3 Cr-Moly	2.52	2.08	2.39
4-6 Cr-Moly	2.05	2.03	2.24
9 Cr-Moly	1.33	5.48	0.79

TABLE 6 RELATIVE CONCENTRATION OF ALLOYING ELEMENTS IN MIDDLE SCALE LAYER FOR SPECIMENS IN CONTACT WITH STEAM AT APPROXIMATELY 1500 F FOR 500 HR

	Ratio of percentage of alloying agent in middle layer and base steel		
Steel	Chromium	Silicon	Molybdenum
3 Cr-Moly	1.72	1.61	1.68
4-6 Cr-Moly	1.75	1.75	1.69
7 Cr-Moly	1.21	1.25	1.18
9 Cr-Moly	1.37	0.96	1.26

Conclusions

The data obtained on the steels tested together with previously reported data (1, 2, 3) which are graphically presented in Fig. 1 show the effect of temperature on corrosion by high-temperature steam. All of the steels tested except the 25-20, 25-15-2W, 21 Cr, and 27 Cr specimens start to corrode rapidly at some temperature less than 1675 F. After the break occurs the rise in corrosion rate is much more rapid for the steel containing 12 per cent chromium and the 18-8 stainless steel than for the steels containing less chromium. The 25-20 and 25-15-2W steels which were tested at temperatures of 1751 F show no corrosion at the end of 500 hr. The temperature at which rapid corrosion begins increases with the chromium content. The 18-8 Cb steel shows the same tendency toward rapid corrosion above some limiting temperature that the S.A.E. 1010 steel shows at a much lower temperature.

Relatively simple empirical equations have been established to aid in computing the amount of corrosion produced in 500 hr at temperatures within the test range for the various steels tested.

Chemical analysis of the various scale layers produced on the 3 Cr-Moly, 4-6 Cr-Moly, 7 Cr-Moly, and 9 Cr-Moly steels during 500-hr tests at temperatures of 1500 and 1730 F are presented. The chemical analysis of the inner scale layer indicated a build-up in concentration of chromium, silicon, and molybdenum. The diffusion of the alloying agents and the inward diffusing oxygen combine to form the dense protective inner layer.

From the results obtained, the chromium content of a steel is a major factor in controlling the amount of corrosion produced by high-temperature steam.

Acknowledgment

The authors wish to express their appreciation to the members of the A.S.M.E. Special Research Committee on Critical-Pressure Boilers for their valuable assistance during the course of the investigation.

Part of this investigation was made possible through the financial support of the Engineering Foundation, The Babcock & Wilcox Company, The Timken Roller Bearing Company, the General Electric Company, the National Tube Company, The Globe Steel Tubes Company, and The Combustion Engineering Company.

The authors are particularly indebted to Dr. C. L. Clark and The Timken Roller Bearing Company, Steel and Tube Division, for making the chemical analyses of the scales reported in this paper.

Appreciation is also expressed to the various alloy-steel manufacturers who have contributed steel samples from time to time for use in this investigation.

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Discussion

R. C. Corey.[5] Since its inception, the work at Purdue University on the corrosion of metals and alloys by high-temperature steam has been followed with considerable interest. No other laboratory studies of this important subject, either here or abroad, have been as comprehensive as the present series, and the authors are to be commended for the excellent data that they have obtained.

It is gratifying to note that the authors conducted their tests for 500 hr. All too frequently one finds data in the literature on the rate of corrosion, intended for general use, but which are based upon extremely short test periods. In a recent article,[6] it was shown that with everything else essentially equal, the corrosion rate varies markedly with the duration of test. It may be argued that regardless of the test period, the relative rate of corrosion of different metals and alloys studied simultaneously under the same conditions will remain the same. Considering, however, that the rate of corrosion of most metals and alloys is greatest initially (except in the unusual case where corrosion increases linearly with time), it is possible that an otherwise satisfactory metal will appear to be unfavorable if the test period is too short.

It is not clear why the authors changed the unit for expressing corrosion from that of penetration, as used in previous reports, to percentage loss of original weight. In order that data of this kind may be compared with those of other investigators, the rate of corrosion is expressed more rationally in terms of the amount of metal lost per unit area in unit time, or as the depth of penetration in unit time. The literature on high-temperature corrosion studies is confusing and difficult to correlate as the result of the use of units which cannot be reduced to a common basis.

If the authors have the data available, it might be of interest to plot the loss of metal against time and thereby determine the specific type of scaling that occurs. Such information would be valuable for interpreting the corrosion mechanism of the various alloys that were studied. Recent papers[7,8] on this subject indicate that scaling may occur linearly, parabolically, or logarithmically with time, depending upon the temperature, conditions, and the metal under study. Thus, the sharp break at a certain temperature noted by the authors for certain of the alloys studied may be the result of recrystallization[9] of the oxide above a certain temperature, which may lead to logarithmic scaling, a factor determined only by the type of plot just mentioned.

The chemical analyses of the scales are of great interest as such data are the first step toward developing a mechanism for corrosion. It is not apparent why the Si increased and the Mo and Cr decreased in amount in the middle oxide layer with increasing chromium content of the alloy at both 1500 F and 1700 F. It is suggested that if possible the authors should have an X-ray diffraction study made of these scales to determine how these elements are distributed, that is, as solid solutions, stoichiometric compounds, or spinels. Such information would afford them valuable supplementary data.

The data of Table 1 of the paper show that Cb-stabilized 18-8 steel loses about 12 per cent of its weight between 1728 F and 1765 F. It might be of interest to study plain 18-8 steel at the same temperature for 500 hr in order to determine if the Cb decreases the corrosion resistance of this alloy.

With reference to the remarks of the authors concerning diffusion of oxygen through the oxide, the following is offered as an alternate explanation: Assuming for the moment that the gas phase diffuses through the oxide to meet and react with metal ions, it is probable that steam and not oxygen, as the authors state, would be involved as it is not until oxidation takes place that oxygen is released, and then it probably combines immediately as an oxide. On the other hand, it is doubtful that a steam molecule could diffuse interstitially through a compact oxide lattice because of the large size of the gas molecule with respect to the interatomic spacing of the lattice. However, if the oxide was porous or fissured the gas could reach the metal ions readily. Also there are cases where a gas like oxygen may be taken into solid solution in an oxide which is deficient in oxygen atoms, i.e., an oxide that is not of stoichiometric composition. The oxide FeO and certain of the spinel type of oxides like $FeO \cdot Fe_2O_3$, $FeO \cdot Al_2O_3$, and $FeO \cdot Cr_2O_3$ are capable of taking a small amount of oxygen into solid solution, but the quantities involved are too small to be significant, thus the rate of scaling probably depends primarily upon the rate at which the iron ions reach the oxide-gas interface.

W. Trinks.[10] On account of being a furnace engineer, the writer's interest in high temperature begins where the power engineer quits. In furnace work we have learned that water vapor scales iron and steel more rapidly than air does. The writer attributes this phenomenon to dissociation. The professors of thermodynamics tell us that dissociation at the temperature

[5] Research and Development Department, Combustion Engineering Company, New York, N. Y.

[6] "Corrosion by Hot Gases," by R. C. Corey, *Combustion*, vol. 15, 1943, pp. 34–39.

[7] "The Transition State Theory of the Formation of Thin Oxide Films on Metals," by E. A. Gulbransen, preprint 83–4, 1943, The Electrochemical Society.

[8] "Laws Governing the Growth of Films on Metals," by U. R. Evans, preprint 83–10, 1943, The Electrochemical Society.

[9] "Oxidation of Metals and the Formation of Protective Films," by N. F. Mott, *Nature*, vol. 145, 1940, pp. 996–1000.

[10] Professor of Mechanical Engineering (Emeritus), Carnegie Institute of Technology, Pittsburgh, Pa.

in question is negligible. They are right, if we limit ourselves to permanent dissociation. They are wrong if we consider instantaneous dissociation. Collision of the molecules causes some dissociation at all temperatures. Ordinarily, the atoms reassociate immediately without leaving any trace of their dissociation; but if a material is present for which oxygen has a greater affinity than it has for hydrogen, it goes after that material, because the oxygen is atomic (in the nascent state) and leaves the unwanted hydrogen to look out for itself.

Water vapor is triatomic and, for that reason is dissociated or cracked more easily than the diatomic oxygen molecule.

The variations in the composition of the scale are what we should expect. Iron oxide has a vapor pressure. Every steel-furnace engineer knows that fact. In a superheater tube the heat comes from the outside. The scale next to the tube is hotter than the scale next to the steam. For that reason the iron oxide gets away from the tube toward the cooler steam. It is a migration which some people call diffusion.

As regards cracking of the scale, we know from the sheet industry that a thin scale adheres and that a thick scale cracks. If we could chromium-plate the inside of the tubes and have the plating stick, then we should have a thin retentive greenish chromium scale which would not crack.

Authors' Closure

The authors are indebted to Messrs. Corey and Trinks for their written discussions and for their excellent comments concerning the mechanism of scale formation.

Mr. Corey has suggested that the data on corrosion be presented in terms of penetration rather than per cent loss in weight. In their earlier reports (1, 2, 3),[11] the authors presented the data in terms of inches of penetration. However, the high-temperature tests reported in this paper produced very thick layers of scale on some of the specimens which were originally 6 in. long and $^1/_2$ in. in diam. This resulted not only in a significant reduction in the diameter of the parent metal during the course of the tests but also produced a shortening of the specimens and a tapering of the parent metal at the ends of the specimens.

Mr. Corey has made some pertinent comments on the effect of time upon the usefulness of corrosion data. The tests which are reported in Table 1 are for 500 hr, and data for shorter periods of time are not available. However, the authors have published data showing the effect of time upon the corrosion of various steels for periods up to 2000 hr at a steam temperature of 1100 F (2).

In conclusion, the authors wish to thank all of those who presented discussions for their interest, comments, and constructive suggestions.

[11] Numbers in parentheses refer to Bibliography at end of original paper.

Surface Fatigue of Plastic Materials

Progress Report No. 16 of the A.S.M.E. Special Research Committee on Strength of Gear Teeth

By EARLE BUCKINGHAM,[1] CAMBRIDGE, MASS.

GREATER knowledge of the surface-fatigue characteristics of materials will serve many useful ends. Surface fatigue is a phenomenon largely responsible for one type of wear, commonly known as pitting. Definite information will permit the more intelligent and effective choice of materials for specific service conditions. Again, as noted in our preceding report,[2] if the surface-fatigue characteristics of the materials used are known and the surfaces of a mechanism are carefully watched, dynamic loads that cause surface failure can be detected before any harm is done to any other part of the mechanism, and a reasonably close measure of their intensity can be made from the condition and appearance of the surface failure of the specific part. Thus such information will give us a tool, or weighing scale, by which to measure the intensity of existing loads on all types of mechanisms in actual operation, and also to make possible more accurate comparisons between many laboratory test results and actual service conditions.

A special testing machine, shown in Fig. 1, has been built, and tests have been run almost continuously during the past 6 years. In brief, the testing machine consists of two shafts running in plain bearings, one mounted in a heavy fixed frame and the other mounted in a substantial swinging frame. The test rolls, with a maximum face width of one inch and from 2.200 in. to 4.00 in. in diameter, are mounted on the shafts. A load is set up between them by a calibrated spring acting against the swinging frame. Revolution counters are connected to each shaft. Gears may be mounted on the ends of the shafts in such a manner as to obtain any desired amount of slipping or rubbing between the two test rolls. Many of the tests under rolling conditions are run without gears, the rolls driving each other by friction. The majority of tests to date have been made under rolling conditions only.

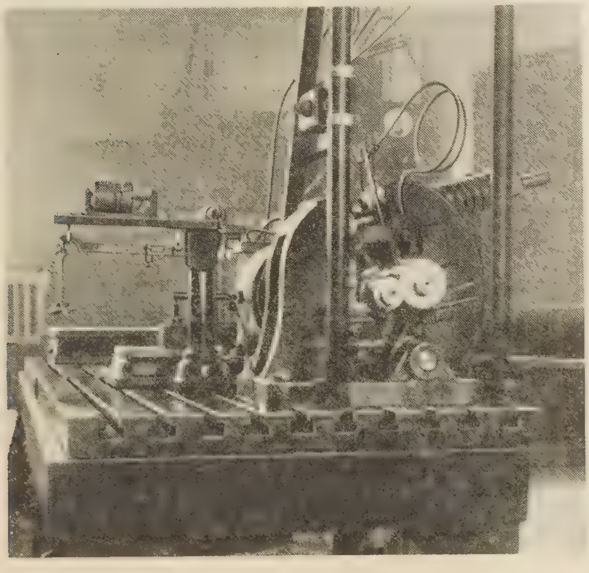

Fig. 1 Testing Machine

Plain bearings were used to avoid the possible dynamic effects of ball or roller bearings. An oil pump provides oil for the bearings and the gears. The testing machine is driven by a small Sprague dynamometer, and records are kept of the torque input during the runs.

The physical properties and the structure of the materials under test appear to have a pronounced influence on the nature of the surface failure. The following is an attempt to put into words a detailed account of the several phenomena that occur concurrently under test conditions of rolling contact. Considering only the plastic materials, the following have been observed.

Plastic Flow of Surface Lamina and Work-Hardening

The action of one roll upon the other carries a plastic and elastic wave ahead of the contact area between them, setting up conditions quite similar in many respects to the conditions existing when cold-rolling metals. This action cold-works the surface lamina, increasing its hardness and raising the physical properties of the surface of the material. This action appears to be progressive, tests indicating that the depth of this cold-working (which might well be called "mechanical case-hardening") increases with the number of repetitions of stress up to a maximum depth, this maximum depth probably depending upon the pressure. The hardness also increases with the number of repetitions of stress, as well as with increase of load; but depth of penetration appears to depend mostly upon the number of repetitions of load.

This increase of surface hardness has been observed on practically all samples of plastic materials which have been tested. Special tests to study this phenomenon were made by E. L. Bartholomew, Jr., in 1937, using rolls of stainless steel (18-8 steel). The three following tests were run on rolls of 2.3 in. diam, 1 in. face width, against a hardened and ground steel roll of the same size.

Test no.	Load, lb	Number of repetitions of stress	Maximum specific compressive stress, psi	Maximum shear stress, psi
110	2362	500000	146000	44400
111	2362	2069000	146000	44400
112	3150	4940000	184000	56000

The stresses are computed as though loaded under static conditions. Photoelastic tests indicate that, under the combined stress conditions existing in rolling contact, the stress distribution is changed and that the stresses are greater than those existing under static conditions of simple radial loading. In the absence

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[2] "Qualitative Analysis of Wear," by Earle Buckingham, 15th Progress Report of the A.S.M.E. Special Research Committee on Strength of Gear Teeth, *Mechanical Engineering*, vol. 59, 1937, pp. 576–578.

Contributed by the Special Research Committee on Strength of Gear Teeth, and the Production Engineering Division, and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

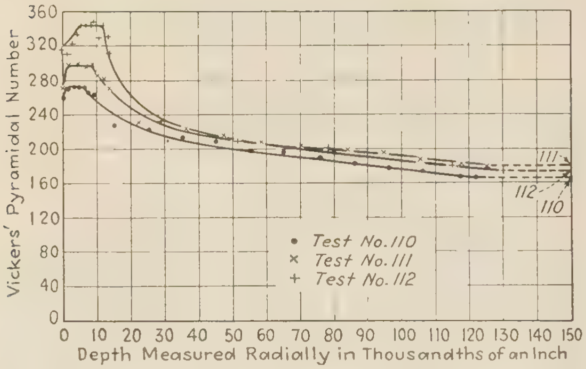


Fig. 2 Results of Tests to Determine Effect of Minimum Torque Conditions

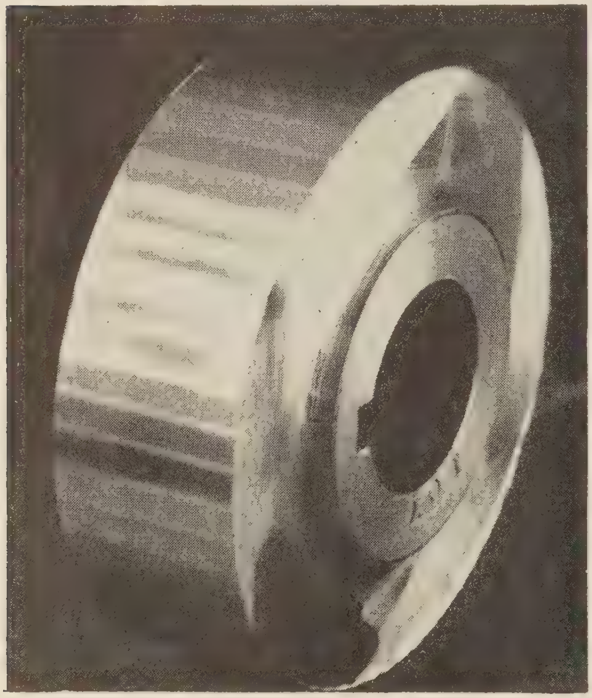


Fig. 3 Test Roll No. 109, Carbon Machine Steel
(3.30-in-diam roll, 0.50 carbon steel, with 8 per cent sliding, 1940-lb load, 67,800 cycles.)

of more exact methods, conditions are compared on the basis of static radial loading only.

On several tests, it has been noticed that the input torque tended to decrease to a minimum as the test progressed, and then to rise slightly after a long period of running under minimum torque conditions. This increase of torque takes place before any indication of destructive pitting can be detected by an examination of the unetched surface. Destructive pitting then occurs in from one or more million cycles after the increase of torque has been detected. Test No. 112 was run until the torque was observed to rise from a minimum of about 1.90 to 2.06 lb at a radius of 1 ft. The results of these tests are plotted in Fig. 2, showing the Vickers hardness numbers against the depth in thousandths of an inch below the surface. The dotted lines at the ends of the graphs show the hardness of the core.

It will be noted that the maximum-hardness reading is obtained at a distance of a few thousandths of an inch below the surface; also, that the depth of maximum hardness is greater for test No. 111 than for test No. 110. It is possible that there is a tendency for the surface material to start to disintegrate, perhaps because of the reversed bending action on the surface as the elastic wave travels ahead of the contact. Photomicrographs taken just below the surface show slip lines, indicative of cold-working. In that of test No. 112, the structure had a semblance of having been more severely worked than either of the other two samples. The grains had the appearance of having shattered in this case. The structure as a whole showed a general precipitation at the grain boundaries and many slip planes.

With very plastic materials, such as soft steel and brass, under heavy loads, this plastic flow of the surface material results in definite waves or corrugations on the surface, much the same as may be found on gravel roads, particularly on the uphill grades. One possible explanation is that when the plastic flow has hardened the material sufficiently, the plastic flow is stopped or reduced at that point; then the rolls ride over and start to build up another wave. Sliding accentuates this condition greatly. Similar waves or corrugations are often found on railroad rails, particularly on curves where sliding between the wheel and rail is greater than on straight stretches.

Fig. 3 shows a 3.3-in-diam test roll of 0.50 carbon machine steel (substantially S.A.E.-1050) which was run with a hardened-steel roll of 2.2 in. diam under a load of 1940 lb for 67,800 cycles. The hardness of the core material was 77 Rockwell B; that of the crest of the waves 89 Rockwell B, and that of the trough of the wave 86 Rockwell B. The pitch diameters of the gears used on the ends of the shafts were 3.20 in. and 2.30 in., thus introducing a relative sliding action between the rolls of slightly less than 8 per cent. Another sample of the same material, 3.5 in. diam, was run with a 2.3-in-diam roll of hardened steel, with rolling action only, under a test load of 3500 lb for 3,000,000 cycles without destructive pitting or waves becoming apparent on the surface.

It is believed that the change in torque observed as the test progresses is some measure of the work done in the plastic deformation of the test samples. For example, on test No. 112, the initial scale load (at a radius of one foot) shortly after the test was started was 3.5 lb. After about one hour, it had fallen to 2.30 lb. After 24 hours continuous running, it had fallen to 2.25 lb. Then it showed a continuous decrease to 1.9 lb at the end of about four million cycles before it started to rise again to 2.06 lb when the test was stopped for examination of the structure of the surface material. The first reduction of from 3.5 lb to 2.25 lb was largely the influence of the warming up of the testing machine. The further reduction of torque is believed to be a measure of the plastic working. When the torque has reached a minimum, and continues there without further change, it is felt that this is reliable evidence that the material has been cold-worked to its limit under the specific test conditions, and that any deformations, other than those resulting from failure of the material, are elastic ones only.

Influence of Elastic Deformation

At the same time that the surface lamina is being cold-worked by the plastic flow, and even after the surface has been cold-worked to the limit, the elastic wave traveling ahead of the contact imposes reversed bending conditions on this surface lamina. The result appears to be, in many cases, the development of microscopic cracks at right angles to the direction of rolling. Microscopic examination of the stainless-steel samples showed evidence, as already noted, of disintegration of the surface material at the grain boundaries.

Incidentally, the number of cycles of stress required to establish the surface-endurance limits is very much greater than the number of cycles required to establish the flexural or bend-

ing endurance limit. Where, in general, from one to five million cycles will establish the flexural endurance limit of the softer steels, from twenty to thirty million cycles appear to be necessary to establish the surface-endurance limit.

In some cases, tests have been run to twenty-five or thirty million cycles without any surface indications (without etching) of destructive pitting, and the test has then been stopped. To make sure that the material is still sound, a light lathe cut has been made on the surface. In some cases, the material has crumbled away in front of the cutting tool, showing the disintegration of the surface material, although it was not evident from a careful examination of the surface. In other cases, a clean chip was obtained, which indicated that the surface material was still sound, yet the appearance of the surfaces which gave either types of chip conditions appeared to be the same. In all cases where a chip has been taken on the surface of rolls which have failed by destructive pitting, the material crumbled away in front of the cutting tool.

In all cases where the loads are appreciable, whether the stresses are above or below the surface-endurance limits of the material, small pits are evident on the surface. These are shallow, possi-

Fig. 4 Blister in Brass Test Roll Running With Hardened-Steel Roll

(Brass roll, 3.5 in. diam; 3500-lb load; 305,000 cycles. Thickness of flake, 0.018–0.021 in.)

Fig. 5 Surface Failure of Nickel-Cast-Iron Test Roll

(Test No. 23: 4.00-in-diam roll; 3500-lb load; 406,000 cycles.)

bly up to 0.005 in. deep at most, generally much less, and the shapes appear to depend upon the structure of the material. Some are microscopic, only a few thousandths of an inch across; others are one sixteenth of an inch or more, and of irregular shapes. This we have called "incipient pitting." Various explanations have been suggested to account for this phenomenon. Dr. Stewart Way[3] ascribes it to the formation of microscopic sur-

Fig. 6 Heat-Treated Nickel-Cast-Iron Roll After Run of 400,000 Cycles Under 3435-Lb Load

(Test No. 34: 3.80-in-diam roll; thickness of flake, 0.033–0.036 in.)

face cracks followed by the penetration of oil which lifts these small sections out because of the hydrostatic pressure developed at the region of contact. It has also been called "corrective pitting" and ascribed to the high local pressures developed on the ridges or peaks of surface irregularities left by the cutting tool. It has also been suggested that the plastic flow of the surface lamina builds up local shearing stresses that shear these particles out after the surface cracks appear. Possibly any or all of these explanations may be true, depending upon the conditions and nature of the material. At all events, this incipient pitting does not appear to be the cause of any great concern. If the loads are below the surface-endurance limits, this incipient pitting appears to progress to a certain extent and then to cease. It is our belief that the elastic wave traveling ahead of the contact is a contributing factor to this phenomenon.

Destructive Pitting

Simultaneously with the foregoing phenomena, and probably with many others not yet observed, shear stresses are repeatedly imposed upon the material below the surface. When the loads imposed develop stresses beyond the surface-endurance limits of the material, particles or flakes will be sheared out of the surface of the material; and thus far in these tests, with very few exceptions, the thickness of these flakes or the depth of the pits has been equal to or greater than the depth to the point of maximum shear.

In the case of a phenolic laminated test roll (bakelite) of 4 in. diam running with a 2.3-in-diam hardened-steel roll under a load of 1720 lb, both rolls of 1 in. face width, at the end of 107,000 cycles of the bakelite roll, a blister appeared on the surface with

[3] Research Engineer, Westinghouse Electric & Manufacturing Company, Research Laboratories, East Pittsburgh, Pa. Jun. A.S.M.E.

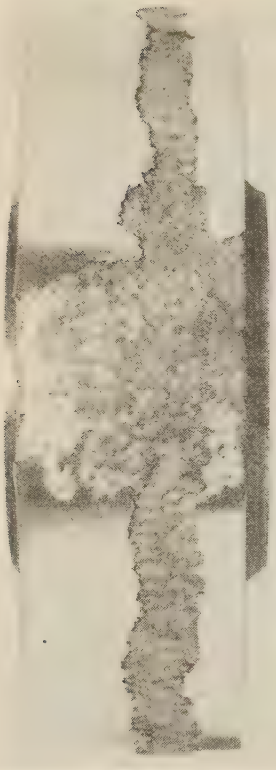


FIG. 7 CHROME-NICKEL-CAST-IRON ROLL AFTER RUN OF 661,000 CYCLES UNDER 2500 LB LOAD
(Test No. 25: 3.70-in-diam roll.)

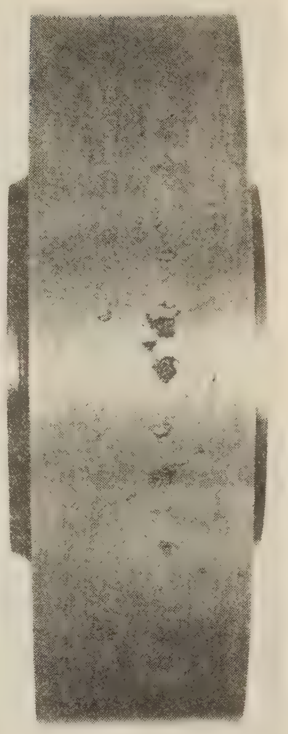


FIG. 8 CHROME-NICKEL-CAST-IRON ROLL AFTER RUN OF 13,090,000 CYCLES AT 1500 LB LOAD
(Test No. 21: 3.70-in-diam roll.)

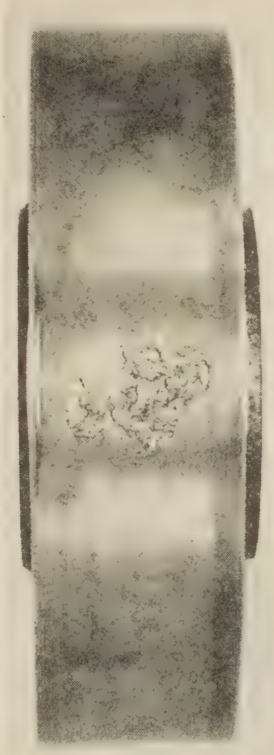


FIG. 9 PHOSPHOR-BRONZE ROLL AFTER RUN OF 1,125,000 CYCLES UNDER 3500 LB TEST LOAD
(Test No. 26: 4-in-diam roll.)

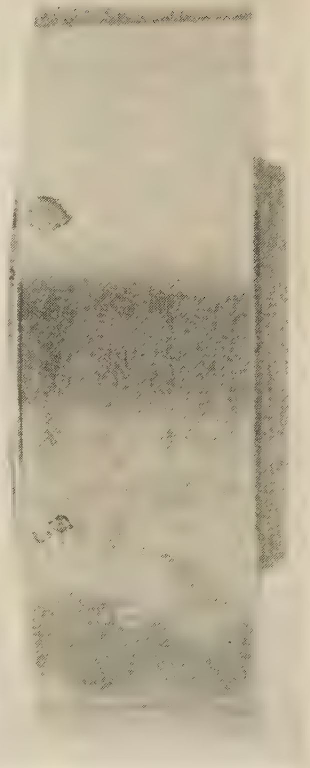


FIG. 10 MACHINE-STEEL ROLL AFTER RUN OF 808,000 CYCLES UNDER 4750 LB LOAD
(Test No. 103: 3.50-in-diam roll, 0.20 carbon steel, 4750-lb load, 808,000 cycles.)

no cracks or other signs of failure on the surface. When the sample was split to examine the blister, a crack about one quarter of an inch long about 0.15 in. below the surface was found to have developed under the blister. In addition, a circular band of the typical dark-brown color of overheated phenolic resins, about 0.030 in. wide and 0.150 in. below the surface, was present, running entirely around and concentric with the surface. A similar dark-brown band of about one half this width was also observed, extending from the surface to slightly below it. Here, the material itself made an automatic record of the heat concentration built up by the internal friction created by the stresses imposed. The testing of these materials is thus further complicated by these thermal conditions. The material has high internal friction, low heat conductivity, and appreciable loss of strength at higher temperatures. Thus in the case of a test roll of this material, 2.3 in. diam, running with a hardened-steel roll of the same diameter, under a test load of 1181 lb running at 950 rpm, the test roll blistered after 7000 cycles. Another roll of the same size and material, under the same test load, running at 350 rpm, blistered after 122,000 cycles.

As noted in Progress Report No. 15,[2] on a brass sample, a blister of similar character to that on the bakelite extended around nearly three quarters of the circumference. This roll is shown in Fig. 4. The brass roll was 3.50 in. diam running with a hardened-steel roll 2.30 in. diam under a load of 3500 lb. The failure occurred at the end of 305,000 cycles; the thickness of flake varied from 0.018 to 0.021 in. The surface hardness before test was 51 Rockwell B and after the test was 87 Rockwell B.

On the softer and more plastic cast-iron alloys, large flakes sheared out of the surface, some one quarter of an inch wide and as much as one inch in length. On the harder heat-treated cast-iron alloys, these flakes were sometimes as small as one eighth of an inch and of irregular shape. These flakes varied in thickness from about 0.010 in. to 0.035 in., depending largely upon the intensity of the load and the number of cycles. Fig. 5 shows the failure of the surface of a 4-in-diam test roll of nickel cast iron, running with a 2.30-in-diam hardened-steel roll under a load of 3500 lb, at the end of 406,000 cycles.

Unless otherwise noted, all tests were run against a hardened-steel roll of 2.3 in. diam. Fig. 6 shows a heat-treated nickel-cast-iron roll, 341 Bhn, 3.80 in. diam, after a run of 400,000 cycles under a load of 3435 lb. The flakes measured from 0.033 to 0.036 in. in thickness.

Fig. 7 shows a chrome-nickel-cast-iron roll, 3.7 in. diam, after a run of 661,000 cycles under a load of 2500 lb. Fig. 8 shows another roll of the same size and material after a run of 13,090,000 cycles under a load of 1500 lb.

Fig. 9 is a phosphor-bronze roll, 4 in. diam, after a run of 1,125,000 cycles under a load of 3500 lb.

Fig. 10 illustrates a machine-steel roll (substantially S.A.E. 1020), 3.5 in. diam, after a run of 808,000 cycles under a load of 4750 lb. The original hardness was 22 Rockwell C, and the surface hardness at the end of the test had increased to 32 Rockwell C.

These illustrations show some of the different types of destructive pitting; on some, the shallow cavities left by the incipient pitting may be seen. There does not appear to be any con-

nection between the incipient pitting and the destructive pitting.

The exact sequence of the different phenomena is a matter of question. Probably in some cases a crack may start under the surface in the region of high shear stresses before the disintegration of the surface material progresses far. This appears to be the case with the phenolic-laminated materials and brass. In other cases, the reverse may be true. Again, both types of failure may progress together. Probably the specific physical characteristics of the materials influence these conditions.

Tests on Cast-Iron Alloys

The most extensive series of tests to date has been on several cast-iron alloys. All of these tests were with rolls, 1 in. face width and running with a hardened-steel roll, 2.30 in. diam, unless otherwise noted. The first test rolls were 4 in. diam and were then turned down after one test and used again in a succeeding test. Here it was noted that the surface material crumbled away ahead of the cutting edge of the tool, so the diameter was reduced until the nature of the chip indicated sound material. The results of some of these tests indicated that the properties of the material might be changing as the diameter was reduced more and more from the original size. Therefore, on later tests, the rolls were cast to a size to finish 2.3 in. diameter. The smaller rolls were used to permit higher speeds of the test disks within the same range of surface velocities as before, to save time, particularly after it became evident that the tests must run to thirty million cycles or more to establish the surface-endurance limits of these materials.

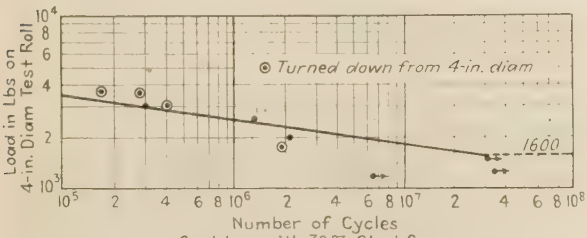
The following tables give representative test results. The actual test data are given first, followed by the equivalent load (based upon static stress conditions) on a 4-in-diam test roll, and the equivalent Brinell hardness number for purposes of comparison.

Semisteel. Table 1 gives the test results on gray iron with

TABLE 1
CAST IRON WITH STEEL SCRAP (SEMI-STEEL)

TEST No.	ROLL DIA.	LOAD IN LBS.	HARDNESS CORE	HARDNESS SURFACE	THICKNESS OF FLAKES	NUMBER OF CYCLES	EQUIV. LOAD 4 IN. ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
32	3.50*	3330	22-C	24-C	—	173,000	3620	240	248
47	3.80*	3435	96-B	102-B	.012-.014	279,000	3500	216	256
131	2.30	2362	190	209	.013	300,000	3000	190	209
48	3.80*	2940	95-B	98-B	—	400,000	3000	210	228
129	2.30	1968	180	190	.007	1,330,000	2500	180	190
128	2.30	1575	—	—	.010	2,093,000	2000	—	—
18	3.70*	1720	26-C	30-C	—	1,920,000	1770	260	283
126	2.30	118.	—	—	NO FAILURE	32,530,000	1500	—	—
125	2.30	1024	—	—	NO FAILURE	36,240,000	1300	—	—
17	3.70*	1210	94-B	97-B	NO FAILURE	6,499,000	1245	205	222

* TURNED DOWN FROM ORIGINAL 4" DIAMETER AFTER EARLIER TESTS



Cast Iron with 30% Steel Scrap
(Table 1)

30 per cent steel scrap, sometimes known as "semisteel." The chemical analysis of a sample of this material is as follows: Silicon, 1.84; sulphur, 0.136; manganese, 0.65; phosphorus, 0.387; total carbon, 3.25; graphite, 2.80; combined carbon, 0.45.

The results of tests of the physical properties of this material were as follows:

	As cast	Heat-treated
Ultimate strength, psi	35200	45950
Elastic limit, psi	15000	35750
Brinell hardness number	223	255
Flexural endurance limit	21000	25000

The heat-treatment (tests Nos. 18 and 32) was as follows: Heat to 1500 F, and quench in oil; draw to 950–1000 F.

After some tests, the surface was so thoroughly destroyed that no reliable hardness measurements could be made. In many cases, the flakes were so broken that no measurement of their thickness could be made.

The surfaces were turned on the samples that did not fail, and the nature of the chips indicated sound material.

As none of the samples under rolling contact failed after 30,000,000 cycles, the intersection of the line representing the load against the number of cycles on a logarithmic chart with the 30,000,000-cycle point has been used to estimate the surface-endurance limit. Thus, the surface-endurance limit of this material has been estimated as between 1500 and 1600 lb load on a 4-in-diam test roll of 1 in. face width in rolling contact with a hardened-steel roll of 2.30 in. diam. The equivalent maximum specific compressive stress, on the basis of static-stress conditions of load, is equal to 87,500 psi.

Nickel Cast Iron. Table 2 A gives the test results on gray iron

TABLE 2A
NICKEL IRON -AS CAST

TEST No	ROLL DIA	LOAD IN LBS.	HARDNESS CORE	HARDNESS SURFACE	THICKNESS OF FLAKES	NUMBER OF CYCLES	EQUIV LOAD 4 IN. ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
33	3.90*	3500	97-B	100-B	—	360,000	3530	222	242
93	2.30	2760	250	266	.010-.012	410,000	3500	250	266
50	4.00	3500	98-B	100-B	—	610,000	3500	228	242
39	3.50*	2900	99-B	99-B	—	275,600	3150	235	235
51	4.00	2500	98-B	100-B	—	1,670,000	2500	228	242
46	3.70*	2435	93-B	95-B	—	314,000	2500	200	210
44	3.80*	2450	94-B	98-B	—	243,000	2500	205	228
52	4.00	1500	98-B	98-B	—	2,663,000	1500	228	228
53	4.00	1100	101-B	101-B	NO FAILURE	15,651,000	1100	250	250

* TURNED DOWN FROM ORIGINAL 4" DIAMETER AFTER EARLIER TESTS

alloyed with nickel, with the rolls as cast. The chemical analysis of a sample of this material is as follows: Silicon, 1.42; sulphur, 0.117; manganese, 0.37; phosphorus, 0.448; total carbon, 3.36; graphite, 2.50; combined carbon, 0.86; nickel, 1.52.

The results of tests on physical properties tested were as follows:

Ultimate strength, psi	35400
Elastic limit, psi	12600
Brinell hardness number	217
Flexural endurance limit, psi	16000

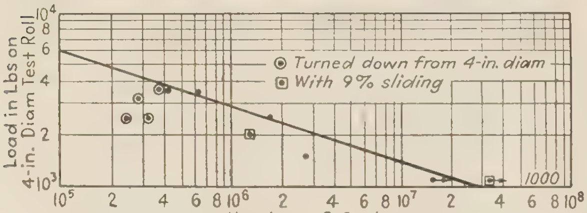
The roll used for test No. 53 did not fail. The surface could be turned after test and the nature of the chips indicated sound material.

On the early tests, the rolls were turned down after previous tests. The results seemed to indicate a possible change in the material away from the original cast surface. Some of the tests were repeated with rolls cast to finish 4.00 in. and 2.300 in. diam. The results showed longer life. The tests were not continued further because heat-treated rolls appeared to give much better results. The surface-endurance limit under the rolling-test conditions

TABLE 2 B

NICKEL IRON - AS CAST WITH ABOUT 9% SLIDING ACTION

TEST No	ROLL DIA.	LOAD IN LBS.	HARDNESS CORE	HARDNESS SURFACE	THICKNESS OF FLAKES	NUMBER OF CYCLES	EQUIV. LOAD 4 IN. ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
140	2.20	1575	—	—	.013-.018	1,220,000	2000	—	—
141	2.20	860	—	—	NO FAILURE	32,250,000	1100	—	—



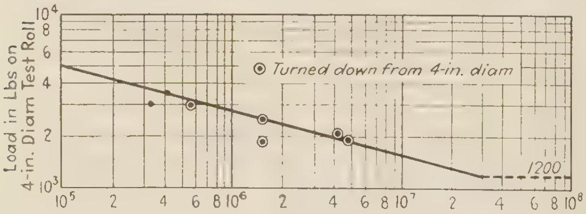
Nickel Iron as Cast
(Tables 2A and 2B)

TABLE 2C

HEAT TREATED NICKEL CAST IRON ABOUT 300 BRINELL

TEST No	ROLL DIA	LOAD IN LBS	HARDNESS CORE	HARDNESS SURFACE	THICKNESS OF FLAKES	NUMBER OF CYCLES	EQUIV. LOAD 4 IN. ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
23	4.00	3500	33-C	—	—	406,000	3500	305	—
34	3.80*	3435	341	—	.033-.036	400,000	3500	341	—
24A	4.00	3000	32-C	—	—	320,000	3000	296	—
36	3.80*	2940	32-C	—	.029-.031	558,000	3000	296	—
45	3.50*	2375	30-C	—	.025-.027	1,630,000	2580	283	—
63	3.80*	2058	32-C	32-C	.016-.020	4,070,000	2100	296	296
67	3.50*	1805	32-C	32-C	.013-.014	4,802,000	1960	296	296
66	3.20*	1740	29-C	—	.025-.033	1,650,000	1898	276	—

* TURNED DOWN FROM ORIGINAL 4" DIAMETER AFTER EARLIER TESTS



Nickel Iron - Heat Treated to 300 Brinell
(Table 2C)

was estimated at about 1000 lb load on the 4.0-in-diam roll. The equivalent maximum specific compressive stress was about 69,200 psi.

Two tests were conducted, introducing about 9 per cent sliding by using a 2.40-in-diam hardened-steel roll, and a 2.20-in-diam nickel-cast-iron roll with gears of 2.3 in. pitch diam on the end of each shaft. These results are given in Table 2B.

The roll used for test No. 141 did not fail. The turned surface indicated sound material. The surface endurance limit with about 9 per cent sliding appears to be about the same as for rolling action, although it appears to fail sooner with test loads greater than the endurance-limit loads.

Another series of tests was made with this nickel cast iron heat-treated as follows: Heat to 1500 F, and quench in oil; draw to 980 F.

The physical properties tested as follows:

Ultimate strength, psi	41700
Elastic limit, psi	24000
Brinell hardness number	246

Table 2C gives the results of rolling tests on this heat-treated material.

TABLE 2D

HEAT TREATED NICKEL CAST IRON ABOUT 350 BRINELL

TEST No	ROLL DIA	LOAD IN LBS.	HARDNESS CORE	HARDNESS SURFACE	THICKNESS OF FLAKES	NUMBER OF CYCLES	EQUIV. LOAD 4 IN. ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
22	4.00	3500	341	—	—	2,202,000	3500	341	—
24	4.00	3200	341	—	—	2,200,000	3200	341	—
35	3.80*	2940	38-C	—	—	2,940,000	2970	352	—
38	3.50*	2375	38-C	—	.024-.029	3,312,000	2580	352	—
49	3.20*	2108	36-C	—	.024-.029	5,600,000	2300	333	—
62	3.80*	2058	39-C	39C	.020-.025	9,192,000	2100	362	362

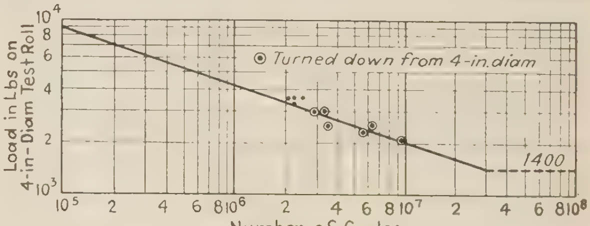
* TURNED DOWN FROM ORIGINAL 4" DIAMETER AFTER EARLIER TESTS

TABLE 2E

HEAT TREATED NICKEL CAST IRON ABOUT 400 BRINELL

TEST No	ROLL DIA.	LOAD IN LBS.	HARDNESS CORE	HARDNESS SURFACE	THICKNESS OF FLAKES	NUMBER OF CYCLES	EQUIV. LOAD 4 IN ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
27	400	3500	44-C	—	—	2,513,000	3500	415	—
29	400	3500	—	—	—	2,165,000	3500	—	—
42	380*	2940	44-C	—	.028-.032	3,177,000	2970	415	—
43	3.80*	2950	42-C	—	.023-.026	6,100,000	2500	393	—

* TURNED DOWN FROM ORIGINAL 4" DIAMETER AFTER EARLIER TESTS



Nickel Iron-Heat Treated to 350 and 400 Brinell
(Table 2D and 2E)

The tests on these heat-treated rolls, which were turned down from earlier tests, seemed to indicate that the material nearer the original cast surface was better than that away from it, much the same as in the case of the materials as cast. These tests were not continued further because a more effective heat-treatment for this material had been developed. The surface-endurance limit of this heat-treated material under rolling action is estimated at about 1200 lb load on a 4-in-diam test roll. The equivalent maximum specific compressive stress is equal to about 75,800 psi.

Another series of tests were made with this nickel cast iron, heat-treated as before, but drawn to about 350 Bhn. The results of these tests are given in Table 2D.

These tests were not continued further because a more effective heat-treatment had been developed for this material. The surface endurance limit under rolling action was estimated to be about 1400 lb test load on a 4-in-diam test roll. The equivalent maximum specific compressive stress is equal to about 81,900 psi.

Another series of tests was made with this nickel cast iron, heat-treated as before, but drawn to about 400 Bhn. The results of these tests are given in Table 2E.

The surface endurance limit under rolling contact was estimated to be about 1400 lb test load on a 4-in-diam roll. The equivalent maximum specific compressive stress is equal to 81,900 psi. When the test points are plotted, they group together with those of the tests on this same material heat-treated to 350 Bhn.

While the foregoing tests on heat-treated nickel cast iron were being made, experiments were being conducted by the company

supplying the material on different heat-treatments. The following heat-treatment was developed which changed the structure of the material, particularly the form and dispersion of the graphite, which gave much higher physical properties to the material (patents have been applied for on this heat-treatment): Heat to 1500 F and hold until thoroughly heated; quench to 650 F and hold until heat is uniform; cool in boiling soda water. On the test rolls, the parts are held about 30 min.

This interrupted-quench treatment gave the following properties to a test sample:

Ultimate strength, psi	46500
Elastic limit, psi	25000
Brinell hardness number	287
Flexural endurance limit, psi	19000

The results of the tests for surface endurance are given in Table 2F.

Rolls used on tests Nos. 122 and 123 did not fail; the surface being turned after test showed sound material.

Test No. 121, being out of line with the others, led to suspicion of the material. A microscopic examination disclosed an improper structure, believed to be evidence of too short a time of heat-treatment. Untested rolls, later used on tests Nos. 122, 123, and 124 were also examined and found to be of the same improper structure. These rolls were heat-treated again, and test No. 124 gave results beyond the average, a possible indication that the double heat-treatment improved the material still further. The surface-endurance limit under rolling action is estimated to be about 2400 lb on a 4-in-diam test roll. The equivalent maximum specific compressive stress is equal to 107,300 psi.

Three additional tests were made, introducing about 9 per cent sliding by using a 2.4-in-diam hardened-steel roll and a 2.2-in-diam nickel-cast-iron roll, with gears of 2.30 in. diam on each shaft. The results of these tests are given in Table 2G.

With 9 per cent sliding action, the estimated surface-endurance limit appears to be about 1800 lb on a 4-in-diam test roll. This is about 75 per cent of the value under rolling conditions. The equivalent maximum specific compressive stress is equal to 92,900 psi.

Chrome-Nickel Cast Iron. Table 3 gives the test results on chrome-nickel cast iron. The chemical analysis of a sample of this material is as follows: Silicon, 1.24; sulphur, 0.130; manganese, 0.44; phosphorus, 0.297; total carbon, 3.39; graphite, 2.50; combined carbon, 0.89; nickel, 1.44; chrome, 0.50.

The physical properties of this material tested as follows:

	As cast	Heat-treated
Ultimate strength, psi	39000	44730
Elastic limit, psi	15100	31600
Brinell hardness, Bhn	234	243

The heat-treatment was as follows: Heat to 1500 F; quench in oil; draw to desired hardness.

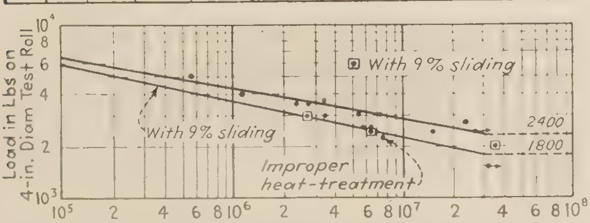
The heat-treatment employed did not appear to have very

TABLE 2F
INTERRUPTED QUENCH - NICKEL CAST IRON

Test No.	Roll Dia.	Load in Lbs.	Hardness Core	Hardness Surface	Thickness of Flakes	Number of Cycles	Equiv. Load 4 in Roll	Equivalent Brinell Core	Equivalent Brinell Surface
71.	3.20	4580	32-C	——	.025-.030	550,000	5000	296	——
97	2.30	3150	303	325	.015-.017	1,120,000	4000	303	325
68	3.50	3330	30-C	30-C	——	3,383,000	3620	283	283
91	2.30	2760	300	——	.021	2,731,000	3505	300	——
95	2.30	2760	302	363	.017-.025	2,480,000	3505	302	363
86	2.30	2362	280	328	.014-.015	3,410,000	3000	280	328
96	2.30	2362	305	363	.021-.035	7,150,000	3000	305	363
99	2.30	2362	255	——	——	5,500,000	3000	255	——
124	2.30	2126	——	——	.016-.025	25,300,000	2700	——	——
98	2.30	2047	268	307	.009-.018	6,370,000	2600	268	307
120	2.30	2047	——	——	.008-.014	5,900,000	2600	——	——
89	2.30	1968	312	345	.027-.031	13,630,000	2500	312	345
123	2.30	1968	——	——	No Failure	28,080,000	2500	——	——
121	2.30	1890	——	——	——	7,740,000	2200	——	——
122	2.30	1181	24-C	27-C	No Failure	31,160,000	1500	248	265

TABLE 2G
INTERRUPTED QUENCH - NICKEL CAST IRON - WITH ABOUT 9% SLIDING ACTION.

Test No.	Roll Dia.	Load in Lbs.	Hardness Core	Hardness Surface	Thickness of Flakes	Number of Cycles	Equiv. Load 4 in. Roll	Equivalent Brinell Core	Equivalent Brinell Surface
130	2.20	2362	264	269	.019-.025	2,730,000	3005	264	269
177	2.20	1968	——	——	.014-.016	6,240,000	2500	——	——
138	2.20	1575	——	——	.018-.028	34,280,000	2005	——	——

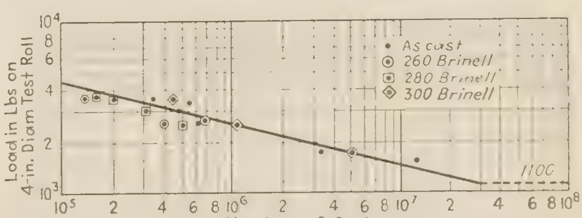


Nickel Cast Iron - Interrupted Quench (Tables 2F and 2G)

TABLE 3
CHROME NICKEL CAST IRON

Test No.	Roll Dia.	Load in Lbs.	Hardness Core	Hardness Surface	Thickness of Flakes	Number of Cycles	Equiv. Load 4 in Roll	Equivalent Brinell Core	Equivalent Brinell Surface
64	4.00	3500	21-C	23-C	——	334,000	3500	235	245
30	3.50*	3138	21-C	24-C	——	542,000	3410	235	248
40	3.20*	2749	100-B	101-B	.018	481,000	3000	242	250
65	4.00	2500	20-C	22-C	——	620,000	2500	230	240
19	3.90*	1720	21-C	21-C	——	3,384,000	1735	235	235
21	3.70*	1500	21-C	22-C	——	13,092,000	1545	235	240
			Heat Treated to about 260 Brinell						
58	4.00	3500	26-C	25-C	.016	140,000	3500	260	255
25	3.70*	2500	27-C	——	——	661,000	2575	265	–
61	4.00	2500	26-C	25-C	.014-.020	390,000	2500	260	255
20	3.90*	1720	27-C	27-C	——	4,950,000	1735	265	265
			Heat Treated to about 280 Brinell						
31	3.50*	3330	30-C	——	——	168,000	3620	283	
57	4.00	3500	28-C	31-C	.017	198,000	3500	272	290
41	3.20*	2749	28-C	30-C	.020-.023	306,000	3000	272	283
60	4.00	2500	28-C	29-C	.016-.020	501,000	2500	272	276
			Heat Treated to about 300 Brinell						
56	4.00	3500	32-C	32-C	.022-.027	435,000	3500	296	296
59	4.00	2500	33-C	33-C	.026-.028	1,070,000	2500	305	305

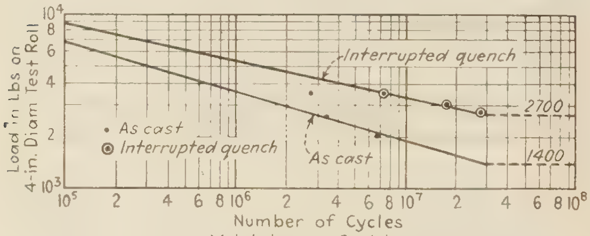
* Turned down from 4 inch diameter after earlier tests



Chrome Nickel Cast Iron (Table 3)

TABLE 4
MOLYBDENUM CAST IRON

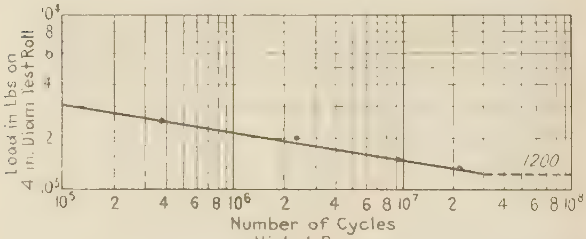
TEST No	ROLL DIA	LOAD IN LBS	HARDNESS CORE	HARDNESS SURFACE	THICKNESS OF FLAKES	NUMBER OF CYCLES	EQUIV. LOAD 4 IN ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
			AS CAST						
133	2 30	2760	215	240	——	2,750,000	3505	215	240
134	2 30	1968	213	240	——	3,200,000	2500	213	240
135	2 30	1575	212	247	——	6,700,000	2000	212	247
			HEAT TREATED, - INTERRUPTED QUENCH						
136	2 30	2760	107-B	113-B	014-.018	7,365,000	3505	291	333
132	2 30	2362	106-B	108-B	——	18,690,000	3000	284	298
137	2 30	2204	106-B	109-B	010-014	28,000,000	2800	284	305



Molybdenum Cast Iron
(Table 4)

TABLE 6
NICKEL BRONZE

TEST No.	ROLL DIA.	LOAD IN LBS.	HARDNESS CORE	HARDNESS SURFACE	DEPTH OF PITS	NUMBER OF CYCLES	EQUIV. LOAD 4" ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
174	2.3	1968	43-B	46-B	.012-.014	381,000	2500	83	87
173	2 3	1575	44-B	53-B	.017-.018	2,320,000	2000	85	97
172	2.3	1181	39-B	46-B	008	9,110,000	1500	79	87
175	2 3	1024	48-B	46-B	.006-.008	22,060 000	1300	90	87



Nickel Bronze
(Table 6)

much influence on the surface endurance, which was estimated at about 1100 lb on a 4-in-diam test roll under rolling action.

Molybdenum Cast Iron. Table 4 gives the test results on molybdenum cast iron. The heat-treatment used was the same interrupted quench as was used on the nickel cast iron and reported in Table 2F. The chemical analysis of a sample of this material is as follows: Silicon, 1.77; manganese, 0.59; total carbon, 3.12; molybdenum, 0.66.

The physical properties of this material, heat-treated, are as follows:

Ultimate strength, psi	49000
Elastic limit, psi	22400
Brinell hardness, Bhn	290
Flexural endurance limit, psi	22000

The estimated surface-endurance limit under rolling action on a 4-in-diam test roll was about 1400 lb for material as cast, and about 2700 lb for interrupted quenched material. The equivalent maximum specific compressive stress for the heat-treated material is equal to 113,700 psi.

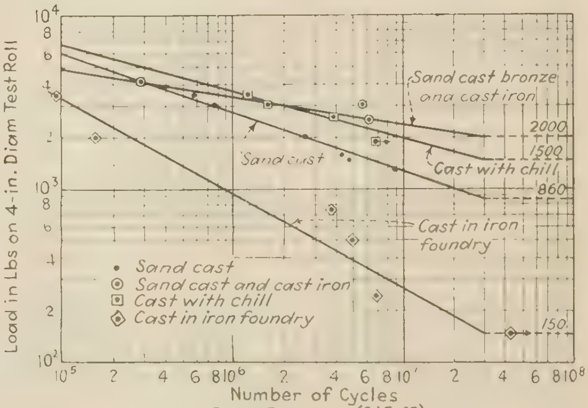
TABLE 5
PHOSPHOR GEAR BRONZE - SUBSTANTIALLY SAE 65

TEST No.	ROLL DIA.	LOAD IN LBS	HARDNESS CORE	HARDNESS SURFACE	THICKNESS OF FLAKES	NUMBER OF CYCLES	EQUIV. LOAD 4 IN ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
			CAST AGAINST A CHILL						
26	4.00	3500	48-B	92-B	——	1,125,000	3500	90	195
54	3.80*	2940	107-V	187-V	.023	1,696,000	3000	107	187
37	4.00	2500	48-B	92-B	——	3,842,000	2500	90	195
56A	3.70*	1845	——	——	No. FAILURE	6,789,000	1900	——	——
			CAST IN SAND WITHOUT CHILL						
78	2 30	2756	43-B	85-B	019-.028	570,000	3500	83	165
79	2.30	2362	45-B	84-B	——	760,000	3000	86	162
80	2 30	1575	47-B	72-B	——	2,600,000	2000	88	130
84	2 30	1332	68	——	——	4,100,000	1695	68	——
85	2 30	1175	62	——	——	4,750,000	1495	62	——
88	2 30	1023	82	——	.010-.015	8,550,000	1300	82.	——
			TESTS WITH 2.30 DIA. CAST IRON ROLLS - 341 BRINELL						
75	2.30	3214	42-B	92-B	BRONZE FAILED	280,000	4080	82	195
77	2.30	2362	47-B	76-B	BRONZE FAILED	5,820,000	3000	88	139
76	2 30	1970	49-B	81-B	.028	+6,016,000	2500	92	153

* TURNED DOWN FROM 4 INCH DIAMETER AFTER EARLIER TESTS.
\+ BOTH ROLLS FAILED.

TABLE 5A
PHOSPHOR GEAR BRONZE — CAST IN IRON FOUNDARY

TEST No.	ROLL DIA.	LOAD IN LBS.	HARDNESS CORE	HARDNESS SURFACE	DEPTH OF PITS	NUMBER OF CYCLES	EQUIV. LOAD 4" ROLL	EQUIVALENT BRINELL CORE	EQUIVALENT BRINELL SURFACE
166	2.3	2760	51-B	86-B	.010-.014	99,000	3500	95	169
167	2 3	1575	30-B	79-B	.007-.010	168,000	2000	72	47
168	2 3	590	41-B	65-B	.009	3,841,000	750	81	16
169	2 3	394	43 B	51-B	.008-.011	4,936,000	500	83	95
170	2 3	197	30-B	44-B	0035-.0045	6,849,000	250	72	85
171	2.3	118	23-B	36 B	DID NOT FAIL	42,842,000	150	66	76



Gear Bronze (SAE-65)
(Tables 5 and 5A)

Phosphor Bronze. Tests were made on two lots of phosphor gear bronze, both of them being substantially the same as S.A.E. 65 bronze. One lot was cast against a chill, the other was not. Three tests were run with the bronze rolls against heat-treated nickel-cast-iron rolls. The results of these tests are given in Table 5.

The test roll on test No. 56A did not fail. When turned after test, the material crumbled ahead of the cutting tool, showing the disintegration of the surface material. The surface-endurance

limit of material cast against a chill was estimated to be about 1500 lb on the 4-in-diam test roll under rolling action. The equivalent maximum specific compressive stress was equal to 82,800 psi.

The estimated surface-endurance limit for sand-cast material under the same conditions is about 1000 lb; equivalent maximum specific compressive stress, 67,600 psi.

The estimated surface-endurance limit for sand-cast material, running with a cast-iron roll under the same conditions, is about 2000 lb; equivalent maximum specific compressive stress, 83,300 psi.

Another lot of similar material was tested, which was cast in a cast-iron foundry with little regard for the technique of handling bronze. The results are given in Table 5A.

This material was evidently badly overheated when it was melted before pouring. The alloy was obtained in pigs from a reliable bronze company. The differences in original hardness would indicate that the samples, identified by the test numbers, were poured in the following order from the ladle: Nos. 171, 167, 170, 168, 169, 166.

The samples from tests Nos. 166 and 167 were etched, and also broken to examine the fracture. Very large crystals were apparent in No. 167, while they were much smaller in No. 166. On the outside surface, where destructive pitting was present, no crystal structure could be seen, indicating amorphous material where the surface had started to disintegrate. The fractures were in keeping with the porous overheated material. When proper foundry practices are followed, this material should have a surface-endurance limit of over 800 lb on a 4-in-diam test roll under rolling action. The estimated surface-endurance limit on these overheated rolls is about 150 lb.

Summary

Although the complex behavior of plastic materials under the repeated surface stresses set up by rolling and sliding is still far from being understood, and also because the actual intensities of the several stresses, compressive, tensile, and shear, are thus far indeterminate, yet the actual test-load results may be applied safely to design. These tests were started in 1931, and the test results have been successfully applied in the design of cams and their roller followers, and in the design of gears since then. The comparative results between different materials as found in these tests are confirmed by the behavior of these materials in service as elements of automatic machines operating in production. The limiting loads, as determined from these tests, have thus far resulted in designs with no appreciable wear.

For the purpose of setting up a load-diameter-stress factor, we will start from the Hertz equation for the stresses set up between two loaded cylinders in contact. When

s = maximum specific compressive stress, psi
w = load on cylinders, lb per in. of length
r_1 and r_2 = radii of cylinders, in.
E_1 and E_2 = modulus of elasticity of materials

$$s^2 = \frac{0.35w(1/r_1 + 1/r_2)}{(1/E_1 + 1/E_2)}$$

We will now introduce an experimental factor of load and stress concentration, based upon the test values; whence K_1 = experimental load-stress factor for two cylinders

$$K_1 = w(1/r_1 + 1/r_2) \quad \text{(by definition)}$$

and

$$w = K_1/(1/r_1 + 1/r_2)$$

Referring now to Table 1, for the test results on a roll of 4.00 in. diam of semisteel running with a hardened-steel roll of 2.300 in. diam, both with a face of 1.00 in., we have an experimental surface-endurance-limit load of 1600 lb. This gives the following values for use in the foregoing equation:

$$w = 1600 \text{ lb}$$
$$r_1 = 1.150 \text{ in.}$$
$$r_2 = 2.000 \text{ in.}$$

$$K_1 = 1600\ (1/1.150 + 1/2.00) = 2191$$

These factors (K_1) are used to determine the limiting surface loads between two curved surfaces. These limiting loads are the ones which can be carried indefinitely without appreciable wear. (If abrasive particles are present, these values do not apply.) For example, if the minimum radius of curvature of a cam is 4 in., and the cam roll is 2.00 in. diam., the cam being made of semisteel and the roll of hardened steel, then

$$r_1 = 1, \qquad r_2 = 4.00$$

$$w = 2191\ (1/1 + 1/4.00) = 1.25 \times 2191 = 2738 \text{ lb}$$

For involute-spur-gear teeth, where

K = load-stress factor for gear teeth
ϕ = pressure angle
D_1 = pitch diameter of pinion, in.
D_2 = pitch diameter of gear, in.
N_1 = number of teeth in pinion
N_2 = number of teeth in gear
F = face width of gears, in.
W_w = limiting load for wear, lb
Q = ratio factor

$$r_1 = D_1 \sin \phi/2$$
$$r_2 = D_2 \sin \phi/2$$

$$(1/r_1 + 1/r_2) = (2/\sin \phi)\ (1/D_1 + 1/D_2)$$

Let

$$K = s^2 \sin \phi\ (1/E_1 + 1/E_2)/(4 \times 0.35)$$
$$Q = 2N_2/(N_1 + N_2) = 2D_2/(D_1 + D_2)$$
$$W_w = D_1FKQ$$

The value of (K_1) for cylinders is

$$K_1 = w(1/r_1 + 1/r_2) = s^2(1/E_1 + 1/E_2)/0.35$$
$$K/K_1 = \sin \phi/4$$

$$K = K_1 \sin \phi/4$$

Hence, for this combination of semisteel and hardened steel, we have the following:
For $14^1/_2$-deg gears

$$K = (2191 \times 0.25038)/4 = 137$$

For 20-deg gears

$$K = (2191 \times 0.34202)/4 = 187$$

As an example, we will use a pair of 6 DP gears, 20-deg pressure angle, 2 in. face, of 24 and 36 teeth. This gives the following values for the limiting wear-load equation:

$$N_1 = 24 \qquad N_2 = 36 \qquad D_1 = 4 \qquad K = 187 \qquad F = 2$$
$$Q = 72/60 = 1.2$$
$$W_w = 4 \times 2 \times 187 \times 1.20 = 1795 \text{ lb}$$

The following values of (K_1) and of (K) have been computed in this manner from the experimental values. Except where noted

Table no.	Material	Cylinder K_1	Gears 14½ deg K	Gears 20 deg K
1	Semisteel	2191	137	187
2(*a* and *b*)	Nickel cast iron	1369	85	117
2(*c*)	Nickel iron, heat-treated, 300B	1643	102	140
2(*d* and *e*)	Nickel iron, heat-treated to 350–400	1917	120	163
2(*f* and *g*)	Nickel iron, interrupted quench	3286	205	280
2(*f* and *g*)	Nickel iron, interrupted quench, with 9 per cent sliding	2465	154	210
3	Chrome-nickel cast iron	1506	94	128
4	Molybdenum cast iron	1917	120	163
4	Molybdenum iron, interrupted quench	3697	231	316
5	Phosphor bronze, sand cast	1177	73	100
5	Phosphor bronze, chilled	2054	128	175
5	Phosphor bronze, overheated when cast	205	12	17
5	Phosphor bronze, sand cast and nickel iron, interrupted quench	2739	171	234

otherwise, a hardened-steel test roll has been used in contact with the material specified.

Discussion

G. B. Warren.[4] Professor Buckingham is to be congratulated upon making another fundamental contribution to gear information. The writer hopes that the author will be able to extend the excellent data contained in this paper to a larger variety of steels with different hardness values, since hardened steels are the more common gear materials used in high-duty machines today. Fundamental data of this character are required in order to extend these designs still further, or to design with greater assurance of reliable performance.

Some of the tests reported in the paper were run with sliding between the rollers. A few of the tabulations indicate a significant increase in the tendency to pit with sliding when the member under consideration is running at a slower speed than its mating roller. There is considerable evidence in actual gear performance and in some tests which have been run by Brown-Boveri along similar lines to indicate that the pitting resistance of the higher-speed roller, of two rollers that are rolling and sliding on each other, is greatly increased. The writer would like to discuss this matter briefly and to draw certain conclusions which may have considerable significance in connection with gear design and performance.

While not posing as a gear expert, it so happened that some gear-pitting difficulties in connection with a gear-reduction job were brought to the writer's attention. In posting himself on the subject, some of the principal contributions to the literature which have been made in recent years were obtained from the company library. Four rather outstanding references were found which are to some extent extracted in the Appendix to this discussion.

After studying these, the writer has come to the following conclusions and hypotheses regarding pitting, some of which fit in with previously advanced theories and some are additions thereto, which, so far as he has been able to determine, have not been made before. These are being submitted herewith for critical consideration:

Referring to Fig. 11 of this discussion, based upon the information which the writer has been able to obtain, particularly from reference (1) in the Appendix, it would seem that there are two types of pitting as follows:

1 A large number of usually disconnected small pits as shown at *N* on the dedendum of the gear, that is, below the pitch line. It is understood this is sometimes called "benign" pitting which very frequently occurs on gears in their early stages of operation and which, although somewhat harmful and a source of annoyance to users, may be self-limiting, i.e., it will reach a stable point and not progress further, particularly on soft gears which are not too heavily loaded. It is not clear what is its end point on harder more heavily loaded gears. This is sometimes called "spalling" in the English literature.

2 Pitting, as shown at *O* between the arrows *P-P* directly on the pitch line, is a form of surface failure which, so far as can be ascertained, only occurs on extremely heavily loaded gears. The pits may form connected areas from which the entire surface may be removed, and this form probably leads to ultimate failure.

With respect to type 1 of pitting described, it seems to be the

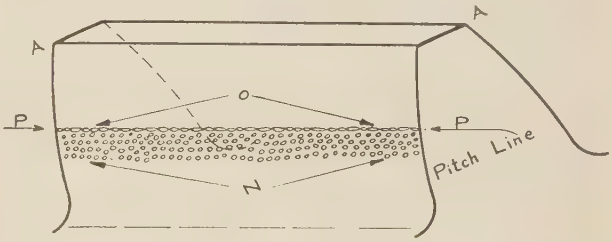


Fig. 11 Gear Pitting

general consensus of opinion: (*a*) This is a difficulty associated with high pressure between two rolling surfaces; (*b*) it takes place only when sliding is in a direction opposite to the rolling; and (*c*) it will only take place in conjunction with and is due to the action of the lubricant.

As a result of the following analysis, and to the evidence of reference (1) the writer has concluded that this type of pitting is much less prevalent on case-hardened gears than on deep-hardened or unhardened gears, and that it might be entirely eliminated by a treatment which would put the surface of the gear tooth under compression, as is done on case-hardened, induction-hardened, or flame-hardened gears which are not ground subsequently.

Let us consider Figs. 12, 13, and 14, of this discussion, respectively, showing the action of two gear teeth along the line of action *a-b-c*, in successive stages of the tooth contact.

In Fig. 12 is shown the first stage of the contact with the motion from left to right and the lower gear driving the upper with the contact at b_1.

The rolling progresses between the two convex surfaces of the teeth upward as the gear turns. Looking at the point of contact in the case of Fig. 12, the tooth on the upper gear slides on the tooth of the lower gear so that the sliding of the upper gear tooth is downward on the lower gear tooth, away from the pitch line, and in such a way as to put the skin of the lower gear tooth, that is, the driver, in tension due to the frictional forces, and the corresponding section of the upper gear tooth, the driven tooth, in compression.

Considering Fig. 13, when the two teeth have progressed so that the contact is at the point where the line of action a_2-b_2-c_2 goes through the pitch line of the two gears, then we have pure rolling contact and theoretically no sliding.

Let us consider Fig. 14, where the contact is on the upper end of the driving tooth just before it leaves contact with the driven tooth at b_3 on the line of action a_3-b_3-c_3. At this point, the gear teeth are being pulled apart by the instantaneous motion, and the driving tooth is sliding downward along the dedendum surface of the driven surface as shown by the arrow *Z*.

[4] Designing Engineer, Turbine Engineering Division, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

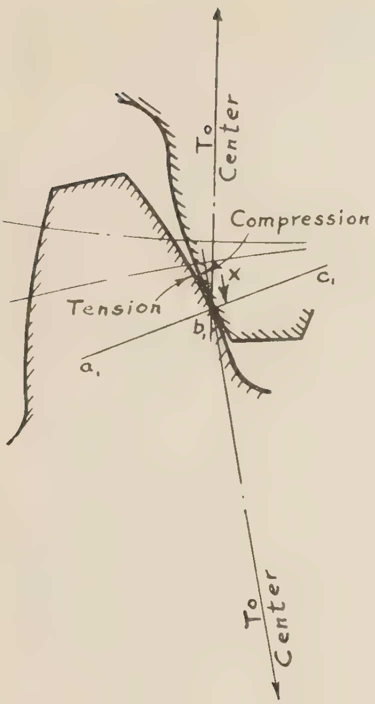


FIG. 12

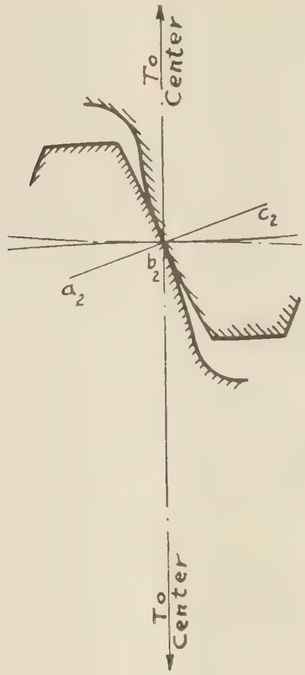


FIG. 13

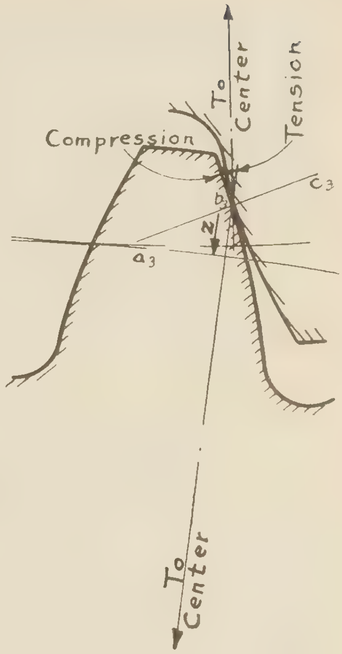


FIG. 14

As stated, the rolling action is progressing upward as the teeth move through the line of action. A careful examination of these three figures will indicate that the addendum, that is, the section of the tooth outside of the pitch line, in each case, is sliding radially on the dedendum portion of the opposite tooth in such a way as to stretch the material of the dedendum ahead of the rolling, and that the oil which has been fed onto the surfaces of the gears is being rolled into this point of contact by the progression of the point of rolling.

This action has been simulated between two rollers, as shown in Fig. 15, by numerous investigators, in which the rollers are

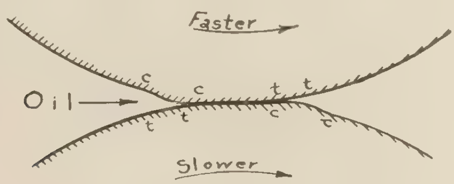


FIG. 15

rolling together in the direction as shown by the arrows but not at the same peripheral speed. If this is done, it can be seen that the traction of the friction parallel to the surface, when the surfaces are forced hard radially together, is such as to produce a tension in the surfaces marked *t-t* and a compression in the portion of the rollers marked *c-c*. The oil is also shown being fed in advance of the rolling point.

Fig. 16 shows an enlarged view of what might be considered as taking place on the surface in which any minute cracks or imperfections on the lower roller would be opened up and correspondingly closed up on the upper roller. The oil would, of course, then be forced into these cracks and raised to very high pressures by its passage under the point of contact, with resulting progressive fatigue failure which would finally lift small pieces of the surface out. This is apparently what happens. Observations show that invariably the pits form on the roller in which the friction of the sliding would produce tension and do not form on the roller in which the sliding would produce compression. This is so marked that even when the upper roller as shown, that is, in this case the higher speed one, is made one half the hardness of the lower roller, the upper roller develops no pits.

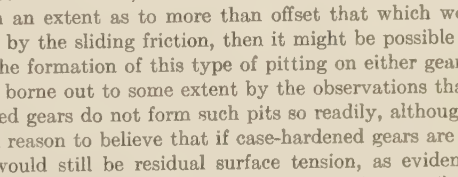


FIG. 16

It would seem that, if the surface could be put in compression to such an extent as to more than offset that which would be formed by the sliding friction, then it might be possible to preclude the formation of this type of pitting on either gear tooth. This is borne out to some extent by the observations that case-hardened gears do not form such pits so readily, although there is good reason to believe that if case-hardened gears are ground there would still be residual surface tension, as evidenced by frequent rejections due to grinding cracks (reference 4). Some tests on induction-hardened rollers which we have made seem to bear this out.

This hypothesis is further evidenced by the fact that on soft gears such pits on the dedendum inside or below the pitch lines, frequently formed during the initial stages of operation, sometimes seem to reach a stable point and do not increase or enlarge from that time on. It would seem as though the formation of a number of pits might permit the escape of the trapped oil, and hence result in a lesser building up of the hydraulic pressure. Also, on soft gears the initial running-in will produce cold work

on the surface and may result in a surface compression stress which would reduce the further formation of these pits.

With respect to type 2 pitting, as shown in Fig. 11 of this discussion, at O along the pitch line, the following hypothesis may explain this formation:

Consider the rolling in Figs. 17 and 18, which takes place at the pitch line of the gearing, as shown in Fig. 13. Theoretically, pure rolling is taking place with no sliding. Actually, up to the pitch line, the sliding had been as shown by the arrows in Fig. 17.

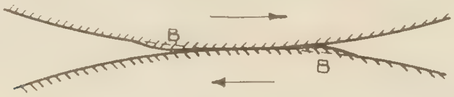


Fig. 17

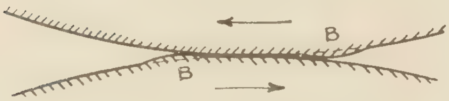


Fig. 18 Gear Pitting

Immediately when the pitch line is reached and passed, the sliding reverses to the direction of the arrows as shown in Fig. 18. It is quite probable that some bulging of the surface, as shown exaggerated in B-B in Figs. 17 and 18, might take place. A more serious situation, however, might exist in the shear forces just below the surface but approximately parallel to the surface due to the tractional or frictional forces existing and to their sudden reversal.

At no other point on the tooth surfaces are these subsurface shear forces reversed through 180 deg in the same material. If the gears are heavy or if the gears and centers are rigid, then this sudden reversal probably takes place along one highly localized line immediately below the surface at the pitch line of both gears. This, it would seem, might easily account for the peculiar types of pits formed at this point, and for the fact that these pits when they are once started seem to progress and run into each other, because, unlike the formation of the other pits, the formation of a few pits would give no relief to the material and would actually increase the pressure on the remaining good surface of the gear.

It is not possible at this time to see any immediate means of remedying this situation.

The Appendix to this discussion contains excerpts from the four references which have been cited and which form the basis for the writer's conclusions.

It is to be hoped that the author or others will extend these roller tests in the near future, including such tests as will prove or disprove the hypotheses presented.

APPENDIX TO DISCUSSION

Reference 1. Excerpts from an article entitled, "Durability of Gears," by H. D. Mansion, which appeared in the *Automobile Engineer*, October, 1941:

Spalling denotes a type of mottled surface failure which appears to consist of a large number of tiny pits extending over a fairly large area, there being no lines in the direction of sliding such as are found in scoring. It has usually been observed on the dedendum of the driven gear teeth and sometimes on the addendum of the driving gear teeth, i.e., mainly during recess in conditions where the oil film is not broken down. Spalling might, in some instances, almost be described as pitting, but the latter term is reserved for the deeper failure to be described.

Pitting is the breaking out from the surface, usually on or near the pitch line, of large flakes, the resultant pits or holes being much deeper than any of the foregoing forms of surface failure. Pitting is generally considered to be a fatigue failure of the surface, though some difference of opinion exists as to the mechanism by which it occurs.

It was now found, in general, that this steel (Ni-Cr-Mo 100-ton oil-hardening steel, type F) would fail by pitting rather than by scuffing. With jet lubrication only, pitting occurred after short periods at loads of 2300 and 2460 lb per in., but with jet and bath lubrication, in a series of 1-hr tests, pitting did not occur until loads of 3830 and 4520 lb per in. were reached.

Of the surface failures which have occurred, two distinct types of pitting have been observed. The direct-hardening steels produce pits which are generally approximately circular and which do not, as a rule, join up with one another. The case-hardening steels, however, appear to become pitted right across the pitch line as though the case had flaked off over that area. It remains to be seen whether this will occur also on gears which are more deeply cased.

Referring in greater detail to the results shown in the tables and dealing first with the 8 D. P. design, it would appear from tables II and III, so far as the irregular nature of these first tentative tests permits of drawing conclusions, that the Ni-Cr-Mo 100-ton oil-hardening steel, when heat-treated in cyanide, in which condition it is virtually case-hardened, has greater resistance both to pitting and to fatigue failure than when simply oil-hardened without cyanide and can consequently carry a higher load without surface failure or breakage.

The incidence of pitting does not appear to be consistently affected by the grinding method used.

Reference 2. Excerpts from an article entitled, "Testing Gear-Wheel Material," which appeared in the *Brown-Boveri Review* for October, 1939:

A new apparatus developed by Brown-Boveri for the testing of gear-wheel material is described in this article. Some results of tests made are given. It is shown that direction of the sliding movement plays a preponderant part in the durability of the material.

Now, in order to reproduce faithfully the conditions pertaining to teeth faces, it is necessary to generate a sliding movement between the rollers. It is not desirable, to this end, simply to cause the shafts to rotate at different speeds. However, it is possible to create a sliding movement even when both shafts have the same number of revolutions, by making the test cylinders of different sizes or by coupling the shafts through eccentric involute gear wheels. The latter is the solution generally adopted in the present tests. It allows of the faithful reproduction on the periphery of one single test cylinder of all the movements occurring between two tooth faces, and thus of determining the behavior of the material by means of a few tests only.

If the sliding and rolling movements of the teeth of a gear wheel are studied, the following results are obtained: The rolling movement on a driving tooth is from root toward tip while on a driven tooth it is from tip toward root. The sliding on a driving tooth is always directed away from the rolling point and on a driven tooth it is directed toward the rolling point. Therefore, on the addendum, the rolling and sliding motions are in the same direction while being opposite on the dedendum.

Contrary to seizing, pitting only appears after a longer testing time. Small particles break out of the roller surface; the appearance of a typical pit is that of a sector of a circle with an angle of 90 to 120 deg. The fracture, being in the center of a sector in formation, penetrates from here at about 15 deg beneath the surface. On the surface, all that is to be seen at first are the two radii as very fine cracks, until finally the crack has advanced so far that the whole wedge now loosened—probably

under the pressure of the lubricating oil forced in—is lifted out of the surface and breaks off in circular shape.

The surface of the fracture is shell-like, and it can be seen that the fracture has advanced progressively as takes place in a fatigue fracture.

The tests show quite clearly that, contrary to seizing, pitting appears exclusively on one of the two test bodies. As is seen in all the illustrations, the pitting only appears on the dedendum arc *a* whereas the addendum arc *b* is always unaffected.

The limitation of pitting to the dedendum arc *a* is very sharply defined indeed. The pitting stops abruptly at rolling point *A*, the limit is sharply defined and coincides with the theoretically determined rolling point, to 0.01 in.

Thus the tests show that the pitting is furthered very much by sliding, but only when the sliding movement is counter to the direction of the rolling movement. The proclivity to pitting is increased to an astonishing degree and the illustrations show some exceptional examples. In one case with a roller of heat-treated chromium-nickel steel having a yield point of about 48 tons per sq in. much pitting appeared on the dedendum arc *a*, while on the other roller of carbon steel with a yield point of about 29 tons per sq in. no pitting occurred at all on the addendum arc *b*. On another roller, the width of the addendum arc *b* was reduced to less than half, and the linear pressure thus more than doubled. Despite this the addendum arc *b* remained intact while the dedendum arc *a* showed marked pitting.

The influence of the viscosity of the lubricating oil is also of interest; the more fluid the lubricating oil the bigger the pit. With transformer oil, big pits of 5 mm diam are formed, while with cylinder oil thep its are hardly bigger than a pinhead. This seems to confirm the view that the lubricating oil forced into the fatigue cracks helps pit formations and finally lifts the loose material from the surface. A heavy cylinder oil will be more likely to detach the material loosened from the lower layer and thus to complete the formation of a pit than a thin transformer oil which can easily escape from the narrowest cracks.

In order to investigate more completely the resistant character of the addendum arc, further tests were carried out with somewhat different apparatus. As said before, it is possible to generate a sliding movement of the two test rollers, one on the other, by imparting uniform running to the shafts but by using test rollers of different sizes. The sliding movement is then the same at every point. With the dimensions chosen here the specific sliding on the smaller test piece was minus 0.29 and on the bigger one plus 0.22. The result of these tests was as follows:

The smaller test pieces were made of carbon steel and of nickel steel with a yield point of 29 and 25 tons per sq in., respectively. The bigger test piece was of chromium-nickel steel with a yield point of 48 tons per sq in. After 3 to 6 hr, pits appeared on the smaller roller. The linear pressure was 1500 lb per in. on this test, the speed being 1000 rpm.

The contrary arrangement, that is, with smaller test pieces of chromium-nickel steel having a yield point of 48 tons per sq in., and the bigger ones of carbon steel and nickel steel having yield points of 29 and 25 tons per sq in., respectively, stood up to 100 running hours without any pitting at all. In order to force deterioration of the addendum arc, the surface of the bigger cylinder was turned down to about half its width and the load increased as much as the safety of the bearings would permit. The linear load was then 3500 lb per in., the stressing, according to the Hertz formula, 70 tons per sq in. After about 100 hr of running, in one case the smaller chromium-nickel roller broke down; in spite of having a yield point of 48 tons per sq in., it could not stand up to the bigger nickel-steel cylinder, having a yield point of only 25 tons per sq in.

We are at a loss, up until today, to furnish a plausible explanation of why the same material should support such exceptionally different stressings according to the direction of the sliding movement. Perhaps subsequent tests will give the answer to this problem.

Reference 3. Excerpts from a paper entitled, "Westinghouse Roller and Gear Pitting Tests," by Dr. Stewart Way, which was presented at the 24th Annual Meeting of the American Gear Manufacturers Association, May 1940:

The investigations being carried on at the Westinghouse Research Laboratories of surface fatigue failures (pitting) are of two types, roller tests and gear tests. Both are more or less indefinitely continuing activities, as new questions about pitting are always arising. Some of the results of the roller tests have been reported previously. (These roller tests were run at the same peripheral speed and were not sliding relatively. —G. B. Warren.)

As a result of these tests there are certain things which can now be put down as fairly definitely known about the formation of surface fatigue cracks on cylindrical surfaces.

We know they originate because of stresses due to surface irregularities rather than because of the theoretical Hertz stress system for smooth cylinders. The theoretical stresses for smooth cylinders do not depend on roughness; on the other hand, pitting strengths depend to a marked degree on roughness. Furthermore, if the pitting cracks were due to the theoretical stress system for smooth cylinders, they would be expected to start at a distance below the surface several times greater than the thickness of the surface layer in which the cracks are observed to start.

We know that it is possible to subject a piece of metal to a range of shearing stresses considerably larger than the shearing endurance range and yet have no failure. Thus the last pair of rollers listed in Table 3 was run 9.4 million cycles at a load of 202,000 psi theoretical compressive stress with no sign of pitting. This loading would give rise to a range of shearing stress plus or minus 50,500 psi on planes parallel to the surface at a depth of 0.0047 in. The shear endurance range of the material was only about plus or minus 30,000 psi, yet when one of these rollers was sectioned there was no evidence of cracks below the surface. The explanation of this is probably that high uniform compression, which exists in the metal below the loaded region, tends to prevent the formation of a fatigue crack at a range of shearing stress that would normally cause a crack.

We know that the growth of surface fatigue cracks to pits is made possible by the action of the oil in the cracks. Pitting cracks which develop in rollers run without oil will not grow to pits. By drilling a small hole (0.006 in. diam) in such a position as to relieve any oil pressure in a growing pitting crack, the growth of the crack can be arrested. The pitting cracks assume a direction that slopes obliquely into the metal; they are roughly in the form of a conical surface so that they meet the surface in the from of a parabola or V, with the vertex of the V being the part that is run over first. It is observed that reversing the direction of rotation after such sloping cracks have formed will stop the growth of the cracks.

When the fatigue cracks form, they may at first slope in various directions. However, there will most likely be some directions more probable than others. All cracks that have been observed in an early stage, both when viewed on the surface and in sections, are oriented in such a way that a perpendicular to the plane of the crack drawn in the outward direction leans in the direction of motion of the contact area. This means that any oil which enters such a crack will tend to be trapped as the crack passes under the loaded region, and a high pressure may be built up in it. A high oil pressure in a crack will produce high tensile stresses at the end of the crack, unless such tension is

canceled by the stresses due to external loads. It is quite possible that a crack, filled with oil, may come under the loaded area so that a high pressure is developed in the oil before the external loads can cause compression stress at the end of the crack to cancel the tension developed by the oil pressure. The crack might then easily be driven further into the metal. If this theory is accepted as a hypothetical picture of what goes on after a surface fatigue crack is formed, it accounts for the observations that oil is necessary for crack growth, that changing the direction of rotation stops crack growth, and that relief of oil pressure in the crack stops its growth.

To study the pitting resistance of gear teeth as influenced by various factors, and to determine what relations exist, if any, between the pitting limits of gear and pinion materials, as determined by roller tests and the pitting strengths of these materials when built into gear teeth, a gear-testing program has been undertaken.

We are here concerned only with the surface compressive stresses, not with the stresses due to tooth bending.

A synopsis of several gear tests shows the following:

The pitting of the gear was confined to the region between the pitch line and the end of one tooth contact.

It is significant that no pitting occurred above the pitch line, although the compressive stresses were slightly higher there.

There was no pitting above the pitch line on the pinion, and nine out of the ten pits were below the pitch line in the region of one-tooth contact.

In reviewing the Brown-Boveri results (2), Dr. Way goes on to say: It was found that negative specific sliding, as occurs in the dedendum region, diminishes pitting resistance, and positive specific sliding increases it (that is the resistance). A carbon steel that pitted at 92,000 psi compressive stress with negative specific sliding carried 140,000 psi without pitting, with positive specific sliding. This finding agrees well with the observation in our gear tests that pitting practically never occurred in the addendum region. It also indicates that negative specific sliding decreases the pitting resistance as compared to that for pure rolling contact.

Future rolling tests with rollers of different ratios and with various amounts of sliding will throw more light on the expected performance of gear teeth.

In a discussion of his article Dr. Way states the following:

On the sketch this is the pitch point and in this region the sliding is in the direction toward the pitch line. It was actually observed in gear tests that pitting was more or less entirely distributed over that whole region below the pitch line and in the region of one-tooth contact, rather than there being any particular tendency for pitting to occur at the pitch line. And since pitting occurred in the region of one-tooth contact where the contact pressure was even lower than it was just above the pitch line, it seems that the sliding, when it is in the direction opposite to the rolling, has the effect of promoting the pitting rather than retarding it, as might be expected, if this view of the oil-film formation is accepted.

You will observe in this region, just above the pitch line, where one pair of teeth was in contact that there was no pitting, while below the pitch line, even though the pitch line where the stresses were much lower than this, there was pitting.

Reference 4. Excerpt from an article entitled, "Peened Surfaces Improve Endurance of Machine Parts," by J. O. Almen, which appeared in *Metal Progress* for February, 1943:

When carburized parts such as bearing races, wrist pins, and gear teeth are ground, we may expect the surface to be stressed in tension.

The residual compressive stress in the carburized layer may be a hazard for members stressed in tension, because the tension stress in the core is equal to the working stress plus the tension stress due to the compressive preload of the case. For members stressed in bending and in torsion the internal compressive stress in the carburized case improves the fatigue strength of the part, except for the thin surface layer which, especially after grinding, is severely stressed in tension. It is, however, a simple matter to convert this thin tension-stressed layer into stress in compression by a suitable peening or rolling operation.

Author's Closure

The Research Committee accepts with thanks Mr. Warren's discussion as a valuable addition to the report. The only comment is in relation to his statement in his fifth paragraph: ".... it would seem that there are two types of pitting as follows:"

This could well be amended to read ".... there appears to be at least two types of *surface* pitting as follows:"

The breakdown of the surface is a very complex process; many different influences are acting and many types of pitting may be present. The author's presentation has been primarily of that type of pitting which results from the subsurface shear stresses. The many surface manifestations of distress deserve much more study, and it is hoped that continued tests, particularly with the harder materials, will shed more light on this phase of the problem.

Simplified Method of Analysis of Reactions Developed by Expansion in a Three-Anchor Piping System

By BORIS LOCHAK,[1] NEW YORK, N. Y.

Existing methods of stress analysis in a three-dimensional piping system fully restrained at three points involve the predetermination of 78 coefficients, and the solution of 12 simultaneous equations. The author proposes a method which reduces the complicated three-anchor problem virtually to a simple two-anchor problem, involving only 21 constant coefficients and the determination of 6 equations. Actually, only routine tabulated computations are required by this method.

The determination of the reactions set up by expansion in a three-dimensional piping system fully restrained in three points, as illustrated in Fig. 1, is regarded as a complicated problem of pipe stress analysis. The total number of component reactions developed in such a system amounts to 18, as follows:

3 Component moments, and 3 component forces at point A
3 Component moments, and 3 component forces at point B
3 Component moments, and 3 component forces at point C

The 6 reactions at point C being equal, with opposite signs, to the sums of the respective reactions at the two other points A and B, 12 unknowns only remain to be found:

6 unknowns at point A
6 unknowns at point B

The determination of the 12 unknown reactions requires 12 simultaneous equations which are obtained from the following conditions of equilibrium:

1 There is no rotation in either direction at point A; i.e., the summation of the moment areas between the points A and C alongside the branches I and III is equal to zero. This condition supplies 3 equations.

2 There is no translatory movement of point A in either direction, i.e., the summation of the moments about the point A of the moment-areas between the points A and C is equal to zero. This condition supplies 3 more equations.

3 No rotation at point B: 3 equations analogous to 1.

4 No movement of point B: 3 equations analogous to 2.

The four groups, of 3 equations each, constitute the system of 12 simultaneous equations solving the problem.

Existing Methods. The existing methods of stress analysis, whether it is the slope-deflection method, the moment-area method, or column analogy, all lead to the formulation of 12 equations, and various procedures proposed for the solution of three-anchor problems actually use 12 equations in numerical applications. So does the method published by the M. W. Kellogg Company,[2] in which the constant coefficients appearing in the equations have been given definite, although dissimilar, mathematical expressions.

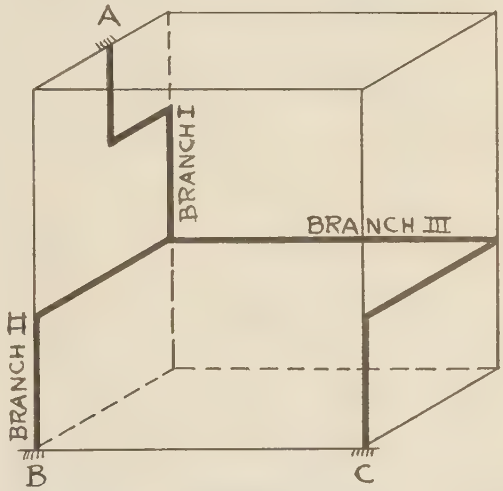


Fig. 1

All these methods involve the predetermination of 78 coefficients, which is exacting work, requiring thorough mathematical training and special skill. They require also the solution of 12 simultaneous equations.

Proposed Method

The proposed method to be described goes two steps further:

1 The 12 equations have been developed first by the moment-area method, and then reduced to 6 only by means of a quite extensive mathematical transformation which has to be omitted from this paper. In the 6 equations, the moments have been eliminated, and only the 6 unknown component forces appear. The number of constant coefficients has therefore been reduced from 78 to 21. The determinant of the 6 equations is symmetrical about the diagonal, offering therefore a simple solution by using the Kellogg method.[2]

2 By selection of the co-ordinate system, the 21 constant coefficients are given very simple mechanical meanings. These coefficients are either moments of inertia, or products of inertia, elementary notions with which every designer is quite familiar. No mathematics whatsoever is required for the understanding and the use of the method, only routine tabulated computations being necessary.

With these two transformations, the solution of the complicated three-anchor problem becomes practically as simple as the solution of the two-anchor problem by S. W. Spielvogel's well-known method.[3]

[1] Structural Engineer, Gibbs & Hill, Inc.

[2] "Design of Piping Systems," published by the M. W. Kellogg Company, New York, N. Y., 1941.

Contributed by the Power Division of The American Society of Mechanical Engineers.

[3] "Piping Stress Calculations Simplified," by S. W. Spielvogel, McGraw-Hill Book Company, Inc., New York, N. Y., 1943.

Procedure. (*a*) Refer the piping system to a system of axis of co-ordinates centered at point O, junction point of the three branches, Fig. 2. Designate the branch OA, branch I; branch OB, branch II; branch OC, branch III. Assume positive directions of co-ordinates, as shown by the arrows, Fig. 2.

(*b*) Project each branch of the piping system on the three co-ordinate planes X-Y, X-Z, Y-Z, Fig. 3.

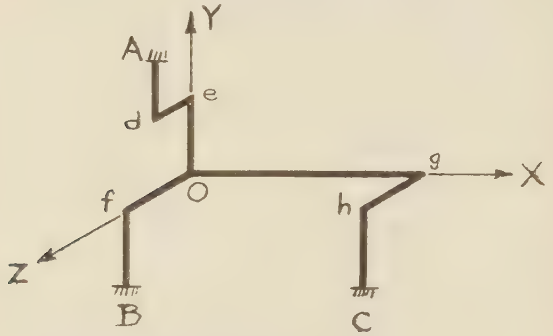


Fig. 2

their algebraical expressions in the general table of notations, Chart 1.)

(*e*) Compute in each projected plane the K constant[5]

$$K = \frac{1}{\frac{1}{L_1} + \frac{1}{L_2} + \frac{1}{L_3}}$$

where L_1, L_2, L_3 are the developed elastic lengths, respectively, of the branches I, II, and III.

(*f*) Compute the moments and the products of inertia of each branch about its own elastic centroid.[4]

The foregoing computations have to be carried out algebraically with signs as they come out with reference to the axis under consideration.

Setting Up the Equations. The six simultaneous equations are given in Chart 2 in the shape of a determinant. Each coefficient is an algebraical sum of the constants previously computed.

Example 1: The coefficient of X_1 in the first equation is equal to

$$+ [Ix_1 + Ix_3 + K(b^2 + c^2)]$$

(Refer to Chart 1, general table of notations.)

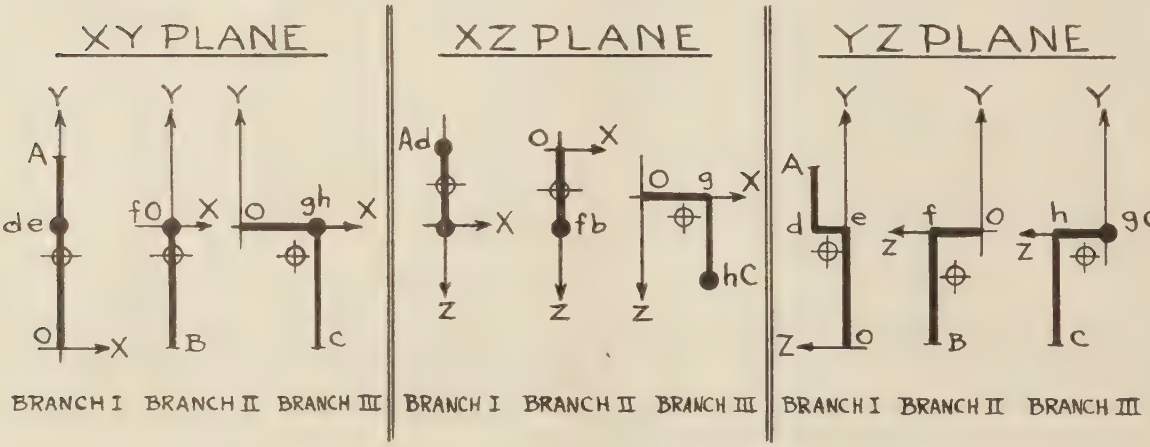


Fig. 3

(*c*) Compute in each projection the elastic centroid of each branch.[4] In doing this (as well as in all the computations hereafter), attribute a stiffness factor equal to 1 to each member projected in a line, and a stiffness factor equal to 1.3 to each branch projected in a point (ratio of EI in bending and GI_p in torsion). The elastic length of every member having a different size, or in general, a different value of EI, shall be corrected in the reversed ratio of the EI values; the stiffer, the shorter. Locate the computed centroids in each projection with reference to the origin (point O).

Designate by $(x_1y_1z_1)$ the co-ordinates of the centroid of branch I
Designate by $(x_2y_2z_2)$ the co-ordinates of the centroid of branch II
Designate by $(x_3y_3z_3)$ the co-ordinates of the centroid of branch III

(*d*) Designate also:

By (a, b, c), respectively, the (x, y, z) distances between the centroids of the branches I and III
By (d, e, f), respectively, the (x, y, z) distances between the centroids of the branches II and III

These distances shall be used with their signs as they come out, with reference to the co-ordinate axis centered at O. (Refer to

The moments of inertia Ix_1 and Ix_3 are computed in the X-Y and X-Z planes, i.e., about the OX direction.

The term Kb^2, which is also a moment of inertia, is computed in the X-Y plane. *Note:* y term and X axis.

The term Kc^2, also a moment of inertia, is computed in the X-Z plane. *Note:* z term and X axis.

Example 2: The coefficient of Y_1 in the first equation is equal to

$$-[Ixy_1 + Ixy_3 + Kab]$$

The three terms are products of inertia computed in the X-Y plane. *Note:* $a = x$ term, $b = y$ term

Following the numerical example given hereafter, the entire procedure will be clarified.

Computation of Expansion Constants. Release the anchors of branch I and branch II, and let the piping system, anchored at the point C, expand freely. Compute the expansion constants as usual with their signs as they come out with reference to the positive direction of the co-ordinate axis.

Computation of Moments. When the 6 equations have been

[4] This can be done in accordance with the basic rules of column analogy, or as explained by S. W. Spielvogel.[3]

[5] The K quantity is an equivalent length. Note the analogy with the expression of the equivalent resistance in electrical network.

solved and the 6 reactions obtained, the moments are easily computed by using the formulas given in Chart 2.

Single-Plane Three-Anchor Problems

Cancel all the z terms, z constants, and z equations. The determinant will be reduced to 4 equations only, and the computations to a surprising simplicity.

Charts 1 and 2 summarize the entire method of analysis of the three-anchor problem.

Numerical Example

The numerical example given in Charts 3, 4, 5, and 6 is taken from M. W. Kellogg's[2] work.

The reactions obtained by Kellogg's method involving the solution of twelve simultaneous equations are (in pounds):

$$X_1 = -4939,\ Y_1 = -6418,\ Z_1 = 2579,\ X_2 = -5515,\ Y_2 = 389,\ Z_2 = 1769$$

The reactions obtained by the method presented involving the solution of six simultaneous equations only are:

$$X_1 = -4937,\ Y_1 = -6468,\ Z_1 = 2600,\ X_2 = -5500,\ Y_2 = 429,\ Z_2 = 1755$$

Acknowledgment

Appreciation is expressed here to Mr. J. Rabinovitz, structural designer, Gibbs & Hill, Inc., for his valuable assistance in the preparation of this paper, especially in working out the charts and the numerical example.

THREE ANCHOR PROBLEM— NOTATIONS

$X_1 Y_1 Z_1$ Reactions applying to branch I

$X_2 Y_2 Z_2$ Reactions applying to branch II

$I_{x1} I_{y1} I_{z1}$ Moments of Inertia of the branch I about its centroid.

$I_{x2} I_{y2} I_{z2}$ " " " " " " II " " "

$I_{x3} I_{y3} I_{z3}$ " " " " " " III " " "

$I_{xy1} I_{xz1} I_{yz1}$ Products " " " " " I " " "

$I_{xy2} I_{xz2} I_{yz2}$ " " " " " " II " " "

$I_{xy3} I_{xz3} I_{yz3}$ " " " " " " III " " "

$\Delta x_1 \Delta y_1 \Delta z_1$ Component expansions of point "a" when released.

$\Delta x_2 \Delta y_2 \Delta z_2$ " " " " "b" " "

E I Modulus of elasticity and sectional moment of inertia of the pipe.

$S_{x1} S_{y1} S_{z1}$ Static moments of branch I about the origin of coordinates.

$S_{x2} S_{y2} S_{z2}$ " " " " II " " " " "

$S_{x3} S_{y3} S_{z3}$ " " " " III " " " " "

L_1 Developed elastic length of branch I.

L_2 " " " " " II.

L_3 " " " " " III.

$$a = \frac{S_{x1}}{L_1} - \frac{S_{x3}}{L_3} \qquad b = \frac{S_{y1}}{L_1} - \frac{S_{y3}}{L_3} \qquad c = \frac{S_{z1}}{L_1} - \frac{S_{z3}}{L_3}$$

$$d = \frac{S_{x2}}{L_2} - \frac{S_{x3}}{L_3} \qquad e = \frac{S_{y2}}{L_2} - \frac{S_{y3}}{L_3} \qquad f = \frac{S_{z2}}{L_2} - \frac{S_{z3}}{L_3}$$

$$K = \frac{1}{\frac{1}{L_1} + \frac{1}{L_2} + \frac{1}{L_3}}$$

$$\left.\begin{matrix} x_1 = \frac{S_{x1}}{L_1} & x_2 = \frac{S_{x2}}{L_2} & x_3 = \frac{S_{x3}}{L_3} \\ y_1 = \frac{S_{y1}}{L_1} & y_2 = \frac{S_{y2}}{L_2} & y_3 = \frac{S_{y3}}{L_3} \\ z_1 = \frac{S_{z1}}{L_1} & z_2 = \frac{S_{z2}}{L_2} & z_3 = \frac{S_{z3}}{L_3} \end{matrix}\right\} \text{Coordinates of the centroids of the branches}$$

$M_{ox1}\ M_{oy1}\ M_{oz1}$ Moments at the origin of coordinates applying to branch I.

$M_{ox2}\ M_{oy2}\ M_{oz2}$ " " " " " " " " " II.

$M_{ox3}\ M_{oy3}\ M_{oz3}$ " " " " " " " " " III.

$M_{ax}\ M_{ay}\ M_{az}$ " " any point "a" of the system.

Chart 1

THREE ANCHOR PROBLEM

Determinant of the six (6) simultaneous equations solving the Reactions $X_1\ Y_1\ Z_1$ and $X_2\ Y_2\ Z_2$.

CONSTANTS	X_1	Y_1	Z_1	X_2	Y_2	Z_2
$\Delta_{x_1} EI$	$+\begin{cases} I_{x1} \\ I_{x3} \\ K(b^2+c^2) \end{cases}$	$-\begin{cases} I_{xy1} \\ I_{xy3} \\ Kab \end{cases}$	$-\begin{cases} I_{xz1} \\ I_{xz3} \\ Kac \end{cases}$	$+\begin{cases} I_{x3} \\ K(be+cf) \end{cases}$	$-\begin{cases} I_{xy3} \\ Kbd \end{cases}$	$-\begin{cases} I_{xz3} \\ Kcd \end{cases}$
$\Delta_{y_1} EI$	$-\begin{cases} I_{xy1} \\ I_{xy3} \\ Kab \end{cases}$	$+\begin{cases} I_{y1} \\ I_{y3} \\ K(a^2+c^2) \end{cases}$	$-\begin{cases} I_{yz1} \\ I_{yz3} \\ Kbc \end{cases}$	$-\begin{cases} I_{xy3} \\ Kae \end{cases}$	$+\begin{cases} I_{y3} \\ K(ad+cf) \end{cases}$	$-\begin{cases} I_{yz3} \\ Kce \end{cases}$
$\Delta_{z_1} EI$	$-\begin{cases} I_{xz1} \\ I_{xz3} \\ Kac \end{cases}$	$-\begin{cases} I_{yz1} \\ I_{yz3} \\ Kbc \end{cases}$	$+\begin{cases} I_{z1} \\ I_{z3} \\ K(a^2+b^2) \end{cases}$	$-\begin{cases} I_{xz3} \\ Kaf \end{cases}$	$-\begin{cases} I_{yz3} \\ Kbf \end{cases}$	$+\begin{cases} I_{z3} \\ K(ad+be) \end{cases}$
$\Delta_{x_2} EI$	$+\begin{cases} I_{x3} \\ K(be+cf) \end{cases}$	$-\begin{cases} I_{xy3} \\ Kae \end{cases}$	$-\begin{cases} I_{xz3} \\ Kaf \end{cases}$	$+\begin{cases} I_{x2} \\ I_{x3} \\ K(e^2+f^2) \end{cases}$	$-\begin{cases} I_{xy2} \\ I_{xy3} \\ Kde \end{cases}$	$-\begin{cases} I_{xz2} \\ I_{xz3} \\ Kdf \end{cases}$
$\Delta_{y_2} EI$	$-\begin{cases} I_{xy3} \\ Kbd \end{cases}$	$+\begin{cases} I_{y3} \\ K(ad+cf) \end{cases}$	$-\begin{cases} I_{yz3} \\ Kbf \end{cases}$	$-\begin{cases} I_{xy2} \\ I_{xy3} \\ Kde \end{cases}$	$+\begin{cases} I_{y2} \\ I_{y3} \\ K(d^2+f^2) \end{cases}$	$-\begin{cases} I_{yz2} \\ I_{yz3} \\ Kef \end{cases}$
$\Delta_{z_2} EI$	$-\begin{cases} I_{xz3} \\ Kcd \end{cases}$	$-\begin{cases} I_{yz3} \\ Kce \end{cases}$	$+\begin{cases} I_{z3} \\ K(ad+be) \end{cases}$	$-\begin{cases} I_{xz2} \\ I_{xz3} \\ Kdf \end{cases}$	$-\begin{cases} I_{yz2} \\ I_{yz3} \\ Kef \end{cases}$	$+\begin{cases} I_{z2} \\ I_{z3} \\ K(d^2+e^2) \end{cases}$

Moments at the origin of Coordinates

$$M_{ox_1} = K\left[Z_1\left(\frac{Y_1}{L_2} + \frac{Y_1}{L_3} + \frac{Y_3}{L_1}\right) - Y_1\left(\frac{Z_1}{L_2} + \frac{Z_1}{L_3} + \frac{Z_3}{L_1}\right) - \frac{1}{L_1}(Z_2 e - Y_2 f)\right] \quad \text{(YZ Plane)}$$

$$M_{oy_1} = K\left[X_1\left(\frac{Z_1}{L_2} + \frac{Z_1}{L_3} + \frac{Z_3}{L_1}\right) - Z_1\left(\frac{X_1}{L_2} + \frac{X_1}{L_3} + \frac{X_3}{L_1}\right) - \frac{1}{L_1}(X_2 f - Z_2 d)\right] \quad \text{(XZ Plane)}$$

$$M_{oz_1} = K\left[Y_1\left(\frac{X_1}{L_2} + \frac{X_1}{L_3} + \frac{X_3}{L_1}\right) - X_1\left(\frac{Y_1}{L_2} + \frac{Y_1}{L_3} + \frac{Y_3}{L_1}\right) - \frac{1}{L_1}(Y_2 d - X_2 e)\right] \quad \text{(XY Plane)}$$

$$M_{ox_2} = K\left[Z_2\left(\frac{Y_2}{L_1} + \frac{Y_2}{L_3} + \frac{Y_3}{L_2}\right) - Y_2\left(\frac{Z_2}{L_1} + \frac{Z_2}{L_3} + \frac{Z_3}{L_2}\right) - \frac{1}{L_2}(Z_1 b - Y_1 c)\right] \quad \text{(YZ Plane)}$$

$$M_{oy_2} = K\left[X_2\left(\frac{Z_2}{L_1} + \frac{Z_2}{L_3} + \frac{Z_3}{L_2}\right) - Z_2\left(\frac{X_2}{L_1} + \frac{X_2}{L_3} + \frac{X_3}{L_2}\right) - \frac{1}{L_2}(X_1 c - Z_1 a)\right] \quad \text{(XZ Plane)}$$

$$M_{oz_2} = K\left[Y_2\left(\frac{X_2}{L_1} + \frac{X_2}{L_3} + \frac{X_3}{L_2}\right) - X_2\left(\frac{Y_2}{L_1} + \frac{Y_2}{L_3} + \frac{Y_3}{L_2}\right) - \frac{1}{L_2}(Y_1 a - X_1 b)\right] \quad \text{(XY Plane)}$$

$$M_{ox_3} = +(M_{ox_1} + M_{ox_2})$$
$$M_{oy_3} = +(M_{oy_1} + M_{oy_2})$$
$$M_{oz_3} = +(M_{oz_1} + M_{oz_2})$$

Moments at any point of the system

$$\text{Branch I} \begin{cases} M'_x = M_{ox_1} - (yZ_1 - zY_1) \\ M'_y = M_{oy_1} - (zX_1 - xZ_1) \\ M'_z = M_{oz_1} - (xY_1 - yX_1) \end{cases}$$

$$\text{Branch II} \begin{cases} M''_x = M_{ox_2} - (yZ_2 - zY_2) \\ M''_y = M_{oy_2} - (zX_2 - xZ_2) \\ M''_z = M_{oz_2} - (xY_2 - yX_2) \end{cases}$$

$$\text{Branch III} \begin{cases} M'''_x = M_{ox_3} - (yZ_3 - zY_3) \\ M'''_y = M_{oy_3} - (zX_3 - xZ_3) \\ M'''_z = M_{oz_3} - (xY_3 - yX_3) \end{cases}$$

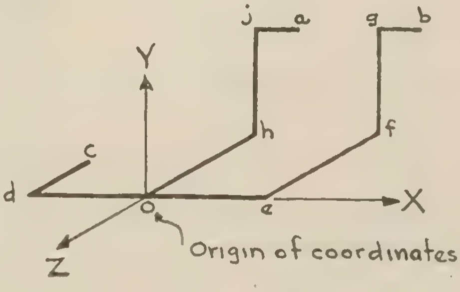
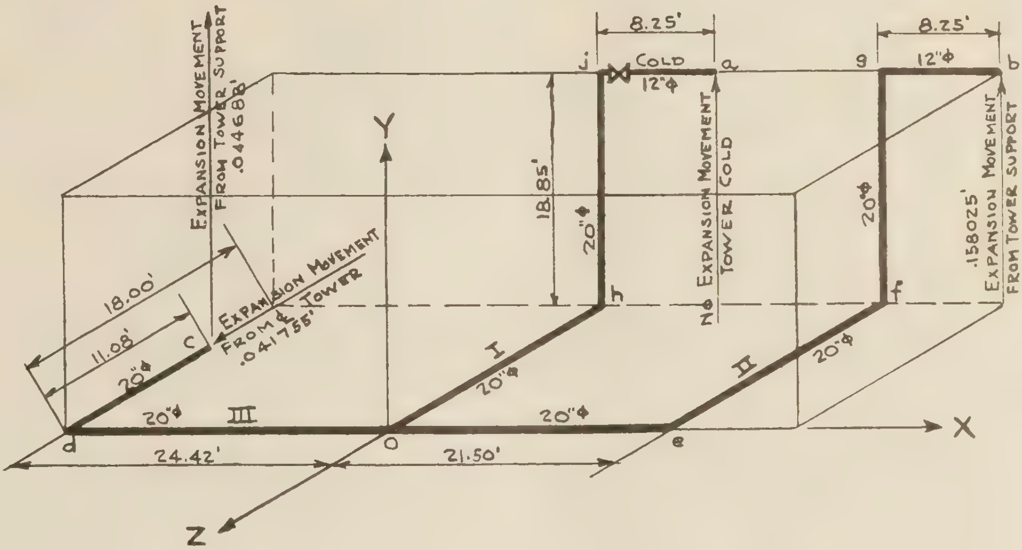


CHART 2

THREE ANCHOR PROBLEM



DATA:—

Carbon Steel
Line Temperature 825°F.
Room Temperature 70°F.
Temperature Diff. 755°F.
Expansion per ft. per 1° = 7.9×10^{-6}
Expansion per ft. per 755° = .005965 Ft.

$E = 29.0 \times 10^6$ for member (a j) only
$E = 23.1 \times 10^6$ for all other members
$I = 1457.0''^4$ for 20" Φ
$I = 361.5''^4$ for 12" Φ
$EI = 233{,}727\,000$ #ft²
$EI = 57{,}991\,000$ #ft² (bg) only
$EI = 72{,}802\,000$ #ft² (aj) only

COMPUTATION OF CONSTANTS

$\Delta x_1 EI =$	233 727 000	[(.005965 × 24.42)]	= + 34 046 000
$\Delta y_1 EI =$	do	[(do × 18.85) + .044688]	= + 36 725 000
$\Delta z_1 EI =$	do	[(do × 6.92) − .041755]	= + 111 000
$\Delta x_2 EI =$	do	[(do × 54.17)]	= + 75 522 000
$\Delta y_2 EI =$	do	[(do × 18.85) + .044688 − .158025]	= − 210 000
$\Delta z_2 EI =$	do	[(do × 6.92) − .041755]	= + 111 000

STIFFNESS CORRECTIONS

Member (a j) $= \dfrac{233\,727}{72\,802} = 3.21$

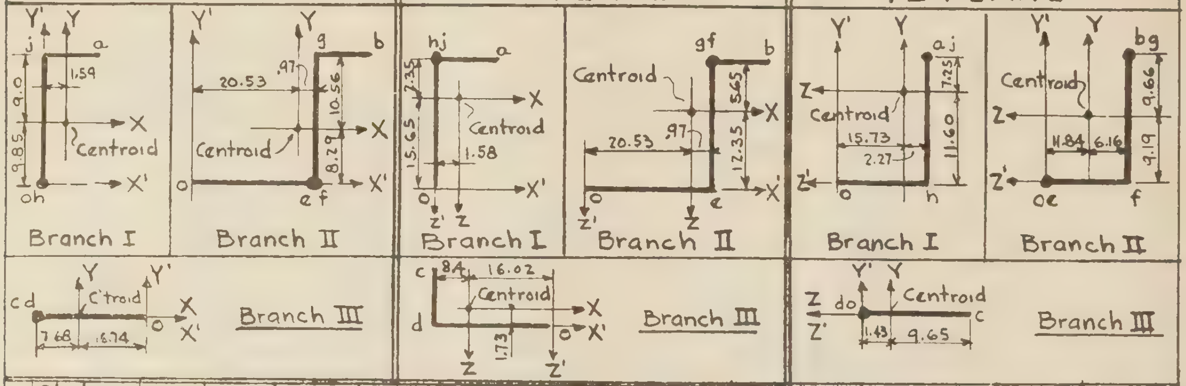
Member (b g) $= \dfrac{233\,727}{59991} = 4.03$

All other Members $= 1.00$

CHART 3

PROJECTION IN

XY PLANE	XZ PLANE	YZ PLANE



Plane	Branch		L	X'	Y'	lX'	lY'	X	Y	lXY	lX²	lY²	I own X	I own Y	IY	IX
XY PLANE	I	aj	26.49	+4.125	+18.85	+109.27	+499.34	+2.535	+9.00	+604.37	170.23	2145.69	150.22	—	320.45	2145.69
		jh	18.85	0.0	+9.425	0.0	+177.66	-1.59	-.425	+12.74	47.65	3.40	—	585.15	47.65	561.55
		ho	23.40	0.0	0.0	0.0	0.0	-1.59	-9.85	+366.48	59.16	2270.33	—	—	59.16	2270.33
		Σ	68.74	+1.59	+9.85	+109.27	+677.00			+983.59					427.26	4977.57
			L1	X1	Y1	Sx1	Sy1			Ixy1					Iy1	Ix1
	II	bg	33.25	+25.625	+18.85	+852.03	+626.76	+5.095	+10.56	+1788.96	863.14	3707.83	188.59		1051.73	3707.83
		gf	18.85	+21.50	+9.425	+405.28	+177.66	+0.97	+1.135	+20.75	17.74	24.28	—	558.15	17.74	582.43
		fe	23.40	+21.50	0.0	+503.10	0.0	+0.97	-8.29	-188.17	22.02	1608.14	—	—	22.02	1608.14
		eo	21.50	+10.75	0.0	+231.13	0.0	-9.78	-8.29	+1743.14	2056.44	1477.57	728.20	—	2884.64	1477.57
		Σ	97.00	+20.53	+8.29	+1991.54	+804.42			+3364.68					3976.13	7375.97
			L2	X2	Y2	Sx2	Sy2			Ixy2					Iy2	Ix2
	III	cd	24.42	-12.21	0.0	-298.17	0.0	+4.53	0.0	0.0	501.12	0.0	1213.54	—	1714.66	0.0
		dc	14.40	-24.42	0.0	-351.65	0.0	-7.68	0.0	0.0	849.35	0.0	—	—	849.35	0.0
		Σ	38.82	-16.74	+0.0	-649.82	0.0			0.0					2564.01	0.0
			L3	X3	Y3	Sx3	Sy3			Ixy3					Iy3	Ix3

Plane	Branch		L	X'	Z'	lX'	lZ'	X	Z	lXZ	lX²	lZ²	I own X	I own Z	IZ	IX
XZ PLANE	I	aj	26.49	4.125	-18.00	+109.27	-476.82	+2.545	-2.35	-158.43	171.58	146.29	150.22	—	321.80	146.29
		jh	24.505	0.0	-18.00	0.0	-441.09	-1.58	-2.35	+90.99	61.17	135.33	—	—	61.17	135.33
		ho	18.00	0.0	-9.00	0.0	-162.00	-1.58	+6.65	-189.13	44.94	796.01	—	486.0	44.94	1282.01
		Σ	69.00	+1.58	-15.65	+109.27	-1079.91			-256.57					427.91	1563.63
			L1	X1	Z1	Sx1	Sz1			Ixz1					Iz1	Ix1
	II	bg	33.25	+25.625	-18.00	+852.03	-598.30	+5.095	-5.65	-957.16	863.14	1061.42	188.59	—	1051.73	1061.42
		gf	24.505	+21.50	-18.00	+526.86	-441.09	+0.97	-5.65	-134.30	23.06	782.26	—	—	23.06	782.26
		fe	18.00	+21.50	-9.00	+387.00	-162.00	+0.97	+3.35	+58.49	16.94	202.01	—	486.00	16.94	688.01
		eo	21.50	+10.75	0.0	+231.13	0.0	-9.78	+12.35	-2596.83	2056.44	3279.23	728.20	—	2784.64	3279.23
		Σ	97.26	+20.53	-12.35	+1997.02	-1201.59			-3679.80					3876.37	5810.92
			L2	X2	Z2	Sx2	Sz2			Ixz2					Iz2	Ix2
	III	cd	24.42	-12.21	0.0	-298.17	0.0	+3.81	+1.73	+160.96	+354.48	73.09	1213.54		1568.02	73.09
		dc	11.08	-24.42	-5.54	-270.57	-61.38	-8.40	-3.81	+354.60	781.80	160.84		113.35	781.80	274.19
		Σ	35.50	-16.02	-1.73	-568.74	-61.38			+515.56					2349.82	347.28
			L3	X3	Z3	Sx3	Sz3			Ixz3					Iz3	Ix3

Plane	Branch		L	Y'	Z'	lY'	lZ'	Y	Z	lYZ	lY²	lZ²	I own Y	I own Z	IZ	IY
YZ PLANE	I	aj	34.43	+18.85	-18.00	+649.01	-619.74	+7.25	-2.27	-566.63	1809.73	177.41	—	—	1809.73	177.41
		jh	18.85	+9.425	-18.00	+177.66	-339.30	-2.175	-2.27	+93.07	89.17	97.13	558.15		647.32	97.13
		ho	18.00	0.0	-9.00	0.0	-162.00	-11.60	+6.73	-1405.22	2422.08	815.27	—	486.00	2422.08	1301.27
		Σ	71.28	+11.60	-15.73	+826.67	-1121.04			-1878.78					4879.13	1575.81
			L1	Y1	Z1	Sy1	Sz1			Iyz1					Iz1	Iy1
	II	bg	43.23	+18.85	-18.00	+814.89	-778.14	+9.66	-6.16	-2572.43	4043.03	1640.39	—	—	4043.03	1640.39
		gf	18.85	+9.425	-18.00	+177.66	-339.30	+.235	-6.16	-27.29	1.04	715.27	558.15		559.19	715.27
		fe	18.00	0.0	-9.00	0.0	-162.00	-9.19	+2.84	-469.79	1520.21	145.18	—	486.00	1520.21	631.18
		eo	27.95	0.0	0.0	0.0	0.0	-9.19	+11.84	-3041.23	2360.55	3918.19	—	—	2360.55	3918.19
		Σ	108.03	+9.19	-11.84	+992.55	-1279.44			-6110.74					8473.98	6905.03
			L2	Y2	Z2	Sy2	Sz2			Iyz2					Iz2	Iy2
	III	od	31.75	0.0	0.0	0.0	0.0	0.0	+1.43	0.0	0.0	64.93	—	—	0.0	64.93
		dc	11.08	0.0	-5.54	0.0	-61.38	0.0	-4.11	0.0	0.0	187.16	—	113.35	0.0	300.51
		Σ	42.83	0.0	-1.43	0.0	-61.38			0.0					0.0	365.44
			L3	Y3	Z3	Sy3	Sz3			Iyz3					Iz3	Iy3

CHART 4

SUMMARY OF CONSTANTS FROM PRECEDING CALCULATIONS

I_{X_1} = 4977.57 + 1563.63 = 6541.20 I_{X_2} = 7375.97 + 5810.92 = 13186.89 I_{X_3} = 0.0 + 347.28 = 347.28	I_{Y_1} = 427.26 + 1575.81 = 2003.07 I_{Y_2} = 3976.13 + 6905.03 = 10881.26 I_{Y_3} = 2564.01 + 365.44 = 2929.45	I_{Z_1} = 427.91 + 4879.13 = 5307.04 I_{Z_2} = 3876.37 + 8473.98 = 12350.35 I_{Z_3} = 2349.82 + 0.0 = 2349.82
I_{XY_1} = + 983.59 I_{XY_2} = + 3364.68 I_{XY_3} = + 0.00	I_{XZ_1} = − 256.57 I_{XZ_2} = − 3629.80 I_{XZ_3} = + 515.56	I_{YZ_1} = − 1878.78 I_{YZ_2} = − 6110.74 I_{YZ_3} = + 0.00

ADDITIONAL CONSTANTS

XY PLANE		XZ PLANE		YZ PLANE	
K	$\frac{1}{\frac{1}{L_1}+\frac{1}{L_2}+\frac{1}{L_3}} = \frac{1}{\frac{1}{68.47}+\frac{1}{97.0}+\frac{1}{38.82}}$ = 19.755	K	$\frac{1}{\frac{1}{L_1}+\frac{1}{L_2}+\frac{1}{L_3}} = \frac{1}{\frac{1}{69.0}+\frac{1}{97.26}+\frac{1}{35.5}}$ = 18.889	K	$\frac{1}{\frac{1}{L_1}+\frac{1}{L_2}+\frac{1}{L_3}} = \frac{1}{\frac{1}{71.28}+\frac{1}{108.03}+\frac{1}{42.83}}$ = 21.441
a	$\frac{S_{X_1}}{L_1} - \frac{S_{X_3}}{L_3}$ = 1.59 + 16.74 = + 18.33	a	$\frac{S_{X_1}}{L_1} - \frac{S_{X_3}}{L_3}$ = 1.58 + 16.02 = + 17.60	b	$\frac{S_{Y_1}}{L_1} - \frac{S_{Y_3}}{L_3}$ = 11.60 − 0.0 = + 11.60
b	$\frac{S_{Y_1}}{L_1} - \frac{S_{Y_3}}{L_3}$ = 9.85 − 0.0 = + 9.85	c	$\frac{S_{Z_1}}{L_1} - \frac{S_{Z_3}}{L_3}$ = − 15.65 + 1.73 = − 13.92	c	$\frac{S_{Z_1}}{L_1} - \frac{S_{Z_3}}{L_3}$ = − 15.73 + 1.43 = − 14.30
d	$\frac{S_{X_2}}{L_2} - \frac{S_{X_3}}{L_3}$ = 20.53 + 16.74 = + 37.27	d	$\frac{S_{X_2}}{L_2} - \frac{S_{X_3}}{L_3}$ = 20.53 + 16.02 = + 36.55	e	$\frac{S_{Y_2}}{L_2} - \frac{S_{Y_3}}{L_3}$ = 9.19 − 0.0 = + 9.19
e	$\frac{S_{Y_2}}{L_2} - \frac{S_{Y_3}}{L_3}$ = 8.29 + 0.0 = + 8.29	f	$\frac{S_{Z_2}}{L_2} - \frac{S_{Z_3}}{L_3}$ = − 12.35 + 1.73 = − 10.62	f	$\frac{S_{Z_2}}{L_2} - \frac{S_{Z_3}}{L_3}$ = − 11.84 + 1.43 = − 10.41

XY PLANE	K	a	b	d	e	a^2	b^2	d^2	e^2	ab	ad	ae	bd	be	de
	19.755	+18.83	+9.85	+37.27	+8.29	335.99	97.02	1389.05	68.72	180.55	683.16	151.96	367.11	81.66	308.97
	Multiplied by K →					6637.48	1916.63	27440.68	1357.56	3566.77	13495.83	3001.97	7252.26	1613.19	6103.70

XZ PLANE	K	a	c	d	f	a^2	c^2	d^2	f^2	ac	ad	af	cd	cf	df
	18.889	+17.60	-13.92	+36.55	-10.62	309.76	193.77	1335.90	112.78	$\bar{2}44.99$	643.28	$\bar{1}86.91$	$\bar{5}08.78$	147.83	$\bar{3}88.16$
	Multiplied by K →					5851.06	3660.12	25233.82	2130.30	$\bar{4}627.62$	12150.92	$\bar{3}530.54$	$\bar{9}610.35$	2792.36	$\bar{7}331.95$

YZ PLANE	K	b	c	e	f	b^2	c^2	e^2	f^2	bc	be	bf	ce	cf	ef
	21.441	+11.60	-14.30	+9.19	-10.41	134.56	204.49	84.46	108.37	$\bar{1}65.88$	106.60	$\bar{1}20.76$	$\bar{1}31.42$	148.86	$\bar{9}5.67$
	Multiplied by K →					2885.10	4384.47	1810.90	2323.56	$\bar{3}556.63$	2285.61	$\bar{2}589.22$	$\bar{2}817.78$	3191.71	$\bar{2}051.26$

Substituting the above constants into the Determinant we obtain the coefficients for the 6 equations

X_1	Y_1	Z_1	X_2	Y_2	Z_2
I_{X_1} = 6541.20 I_{X_3} = 347.28 Kb^2 = 1916.63 Kc^2 = 3660.12 + +12465.23	I_{XY_1} = 983.59 I_{XY_3} = 0.00 Kab = 3566.77 − +4550.36	I_{XZ_1} = − 256.57 I_{XZ_3} = + 515.56 Kac = −4627.62 − −4368.63	I_{X_3} = 347.28 Kbe = 1613.19 Kcf = 2792.36 + +4752.83	I_{XY_3} = 0.00 Kbd = 7252.26 − +7252.26	I_{XZ_3} = +515.56 Kcd = −9610.35 − −9094.79
	I_{Y_1} = 2003.07 I_{Y_3} = 2929.45 Ka^2 = 6637.48 Kc^2 = 4384.47 + +15954.47	I_{YZ_1} = −1878.78 I_{YZ_3} = − 00.00 Kbc = − 3556.63 − −5435.41	I_{XY_3} = + 0.00 Kae = + 3001.97 − +3001.97	I_{Y_3} = 2929.45 Kad = 13495.83 Kcf = 3191.71 + 19616.99	I_{YZ_3} = 0.00 Kce = −2817.78 − −2817.78
		I_{Z_1} = 5307.04 I_{Z_3} = 2349.82 Ka^2 5851.06 Kb^2 2885.10 + +16393.12	I_{XZ_3} = + 515.56 Kaf = − 3530.54 − −3014.98	I_{YZ_3} = 0.00 Kbf = −2589.22 − −2589.22	I_{Z_3} = 2349.82 Kad = 12150.92 Kbe = 2285.61 + +16786.35
Symmetrical about diagonal			I_{X_2} = 13186.89 I_{X_3} = 347.28 Ke^2 1357.56 Kf^2 2130.30 + +17022.03	I_{XY_2} 3364.68 I_{XY_3} 0.00 Kde 6103.70 − +9468.38	I_{XZ_2} = −3629.80 I_{XZ_3} + 515.56 Kdf −7331.95 − −10446.19
				I_{Y_2} = 10881.26 I_{Y_3} = 2929.45 Kd^2 = 27440.68 Kf^2 = 2323.56 + +43574.95	I_{YZ_2} = −6110.74 I_{YZ_3} = 0.00 Kef = −2051.26 − −8162.00
					I_{Z_2} = 12350.35 I_{Z_3} = 2349.82 Kd^2 = 25233.82 Ke^2 = 1810.90 + +41744.89

CHART 5

The 6 equations are as follows:

$$+12465X_1 - 4550Y_1 + 4369Z_1 + 4753X_2 - 7252Y_2 + 9095Z_2 + 34046000 = 0$$
$$-4550X_1 + 15954Y_1 + 5435Z_1 - 3002X_2 + 19617Y_2 + 2818Z_2 + 36725000 = 0$$
$$+4369X_1 + 5435Y_1 + 16393Z_1 + 3015X_2 + 2589Y_2 + 16786Z_2 + 111000 = 0$$
$$+4753X_1 - 3002Y_1 + 3015Z_1 + 17022X_2 - 9468Y_2 + 10446Z_2 + 75522000 = 0$$
$$-7252X_1 + 19617Y_1 + 2589Z_1 - 9468X_2 + 43575Y_2 + 8162Z_2 - 210000 = 0$$
$$+9095X_1 + 2818Y_1 + 16786Z_1 + 10446X_2 + 8162Y_2 + 41775Z_2 + 111000 = 0$$

Solving these equations:

$X_1 = -4937.29$; $Y_1 = -6468.72$; $Z_1 = +2599.87$

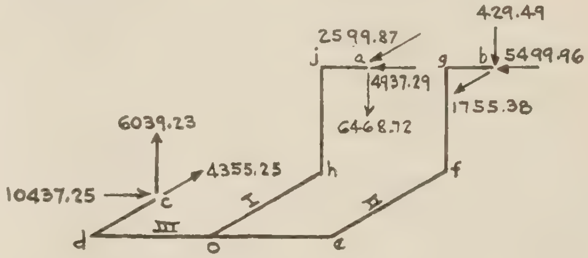
$X_2 = -5499.96$; $Y_2 = +429.49$; $Z_2 = +1755.38$

$X_3 = -(X_1 + X_2) = +10437.25$

$Y_3 = -(Y_1 + Y_2) = +6039.23$

$Z_3 = -(Z_1 + Z_2) = -4355.25$

ALL FORCES GIVEN IN POUNDS



COMPUTATION OF THE MOMENTS AT THE ORIGIN OF COORDINATES

$$M_{OX_1} = 21.441\left[2599.87\left(\frac{11.60}{108.03} + \frac{11.60}{42.83} + \frac{0.00}{71.28}\right) + 6468.72\left(\frac{-15.73}{108.03} + \frac{-15.73}{42.83} + \frac{-1.43}{71.28}\right) - \frac{1}{71.28}(1755.38 \times 9.19 - 429.49 \times -10.41)\right] = -59040'^{\#}$$

$$M_{OY_1} = 18.889\left[-4937.29\left(\frac{-15.63}{97.26} + \frac{-15.65}{35.50} + \frac{-1.73}{69.0}\right) - 2599.87\left(\frac{1.58}{97.26} + \frac{1.58}{35.50} + \frac{-16.02}{69.0}\right) - \frac{1}{69.0}(-5499.96 \times -10.62 - 1755.33 \times 36.55)\right] = 68456'^{\#}$$

$$M_{OZ_1} = 19.755\left[-6468.72\left(\frac{1.59}{97.0} + \frac{1.59}{38.82} + \frac{-16.74}{68.74}\right) + 4937.29\left(\frac{9.85}{97.0} + \frac{9.85}{38.82} + \frac{0.0}{68.74}\right) - \frac{1}{68.74}(429.49 \times 37.27 - 5499.96 \times 8.29)\right] = 40870'^{\#}$$

$$M_{OX_2} = 21.441\left[1755.38\left(\frac{9.19}{71.28} + \frac{9.19}{42.83} + \frac{0.00}{108.03}\right) - 429.49\left(\frac{-11.84}{71.28} + \frac{-11.84}{42.83} + \frac{-1.43}{108.03}\right) - \frac{1}{108.03}(2599.87 \times 11.60 + 6468.72 \times -14.30)\right] = 29627'^{\#}$$

$$M_{OY_2} = 18.889\left[-5499.96\left(\frac{-12.35}{69.0} + \frac{-12.35}{35.50} + \frac{-1.73}{97.26}\right) - 1755.38\left(\frac{20.53}{69.0} + \frac{20.33}{35.50} + \frac{-16.02}{97.26}\right) - \frac{1}{97.26}(4937.29 \times -13.92 - 2599.87 \times 17.60)\right] = 28550'^{\#}$$

$$M_{OZ_2} = 19.755\left[429.49\left(\frac{20.53}{68.74} + \frac{20.66}{38.82} + \frac{-16.74}{97.0}\right) + 5499.96\left(\frac{8.29}{68.74} + \frac{8.29}{38.82} + \frac{0.00}{97.0}\right) - \frac{1}{97.0}(-6468.72 \times 18.33 + 4937.29 \times 9.85)\right] = 56114'^{\#}$$

$$M_{OX_3} = +(-59040 + 29627) = -29413'^{\#}$$
$$M_{OY_3} = +(+68456 + 28550) = +97006'^{\#}$$
$$M_{OZ_3} = +(+40870 + 56114) = +96984'^{\#}$$

$$\left.\begin{aligned} M_{aX} &= -59040 - (18.85 \times 2599.87) + (-18.0 \times -6468.72) = +8389'^{\#} \\ M_{aY} &= +68456 - (-18.0 \times -4937.29) + (8.25 \times 2599.87) = -1034'^{\#} \\ M_{aZ} &= +40870 - (8.25 \times -6468.72) + (18.85 \times -4937.29) = +1169'^{\#} \end{aligned}\right\} \text{Moments at point "a"}$$

$$\left.\begin{aligned} M_{bX} &= +29627 - (18.85 \times 1755.38) + (-18.0 \times 429.49) = -11193'^{\#} \\ M_{bY} &= +28550 - (-18.0 \times -5499.96) + (29.75 \times 1755.38) = -18226'^{\#} \\ M_{bZ} &= +56114 - (29.75 \times 429.49) + (18.85 \times -5499.96) = -60337'^{\#} \end{aligned}\right\} \text{Moments at point "b"}$$

CHART 6

Behavior of Synthetic Phenolic-Resin Adhesives in Plywood Under Alternating Stresses

By A. G. H. DIETZ[1] and HENRY GRINSFELDER[2]

Because plywood is being used in ever-increasing quantities in the construction of aircraft and boats of many types, it is essential to know the effect upon this material of vibrating loads. Hence whether failure occurs in the wood or in the adhesive under repeated stress, what allowable stresses may be applied, and what the fatigue limit may be, are questions which the engineer must be prepared to answer. The present paper gives the results of a research program covering the behavior of plywood and laminated wood bonded with synthetic phenolic-resin adhesives, undergoing alternating stress, in the unweathered and weathered conditions, and at high and low temperatures. This is a continuation of a previous report,[3] which presented some findings with respect to fatigue behavior of plywood and laminated wood under alternating stresses.

Introduction

PLYWOOD is being employed in increasing quantities in aircraft and seacraft, in both of which it is subjected to vibrating loads. It is important for engineers to know whether or not the wood or the adhesive is likely to fail under repeated stress, what behavior should be expected of both wood and adhesive, what allowable stresses can be employed, and what the fatigue limit may be. Should the bond show a tendency to weaken, it is desirable to know the extent of the loss in strength. It should be known whether or not there is any loss of stiffness under alternating stress. In this paper are contained the results of a program of research undertaken to determine

(*a*) The fatigue behavior of plywood and laminated wood under alternating stress.

(*b*) The behavior of the synthetic-resin adhesives used in plywood and laminated wood, under alternating stress.

(*c*) The effect, under alternating stresses, of weathering on the properties of the adhesives used in plywood.

(*d*) The effect, under alternating stresses, of high and low temperatures on the properties of the adhesives used in plywood.

A previous report[3] presented some of the findings with respect to the fatigue behavior of plywood and laminated wood under alternating stresses. In that paper, only item (*a*), that is, the fatigue behavior of plywood and laminated wood as a whole, was reported, but no attempt was made to analyze the effect of alternating shear stresses upon the strength of the glue line. In this paper, items (*b*), (*c*), and (*d*) are reported, that is, the results of investigations into the behavior of the adhesives in plywood under alternating shear stresses. For the sake of completeness, the earlier results are included in this report.

Previous Work

Of the fatigue behavior of plain wood, comparatively little is known, and still less is known about that of plywood. Stanton (1)[4] in 1916 concluded that the endurance limit of spruce was about 27 per cent of the tensile strength, which, in the material he employed, was low (6800 psi). Kommers (2) in 1927 reported tests made at the Forest Products Laboratory, on the basis of which values ranging from 26 to 30 per cent of the static bending strength were assigned Southern white oak, Sitka spruce, and Douglas fir. Schlyter (3) found that Sitka spruce, European spruce, and common European pine had endurance limits 22 to 25 per cent of the static bending strength. Kraemer (4), working with hardwoods (walnut and ash) and softwoods (pine and spruce), found endurance limits ranging from 25 to 33 per cent of the bending strength. He discovered, moreover, that the endurance limit of the heavier woods could in some instances be determined by as few as 20,000 cycles whereas the lighter specimens required several million cycles. Kollmann (5) evaluating the foregoing results, points out that although the endurance limits for wood are relatively low, the ratio of endurance limit to specific gravity may be relatively high compared to the metals used in aircraft. Kraemer (6) investigating plywood bonded with synthetic phenolic-resin film and with casein, found that if the grain of face plies was parallel to the axis of the test piece, the endurance limit was 26 per cent of the tensile strength for resin-bonded and 25 per cent for casein-bonded plywood. If the grain of the face plies was placed at an angle of 45 deg to the axis, the endurance limit for resin-bonded material rose to 52 per cent; for casein-bonded it remained practically the same, or 26 per cent. Failure of the casein-bonded plywood in this instance was caused by exfoliation of the veneers.

TESTS ON UNWEATHERED SPECIMENS

Materials

Aircraft-quality birch veneer was selected because of its strength and toughness and because it is moderately difficult to bond. The combination of strong wood and somewhat difficult bonding ability consequently was expected to reveal any tendency of the bond to fail under repeated stress.

Adhesives were as follows:

1 Thermosetting phenol-formaldehyde film.

2 Aqueous solution of thermosetting phenolformaldehyde.

3 The same as item 2, but extended with ground walnut shells (Glufil).

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[2] Senior Chemist, The Resinous Products and Chemical Company, Philadelphia, Pa.

[3] "Behavior of Plywood Under Repeated Stresses," by A. G. H. Dietz and H. Grinsfelder, Trans. A.S.M.E., vol. 65, 1943, pp. 187–191.

Contributed by the Wood Industries Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

4 Aqueous solution of cold-setting urea-formaldehyde.

These were representative of the principal types of synthetic-resin adhesives commonly employed.

Veneers, $^1/_{16}$ in. thick, were laid up and bonded as follows:

1 Phenolic-resin film: Two-ply laminated; three-ply laminated; three-ply plywood.

2 Phenolic-resin, plain aqueous solution: Three-ply plywood.

3 Phenolic-resin, extended aqueous solution: Three-ply plywood.

4 Cold-setting urea resin, plain aqueous solution: Three-ply plywood.

Both laminated and regular plywood construction were investigated in the film-bonded material because differing stress conditions are set up in laminated wood and in plywood. Laminated wood behaves much the same as solid wood, but in plywood with the face grain parallel to the specimen axis, the bending stresses are almost entirely concentrated in the face plies and the shear-stress intensity practically reaches its maximum at the glue line. If the bond were weakened by repeated stress, the behavior of laminated wood and plywood might be expected to differ. Only three-ply plywood was investigated with the other three adhesives because experience with the film-bonded materials indicated that glue-line failure was more apt to occur, if at all, in plywood than in laminated wood.

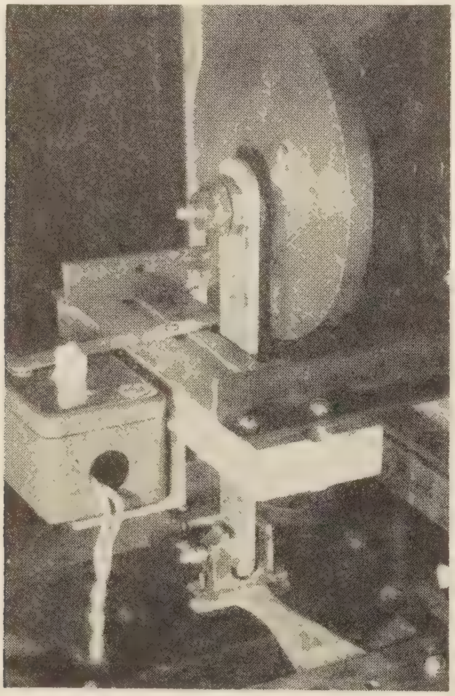


Fig. 1 Cantilever-Type Machine Employed With Fatigue Specimens

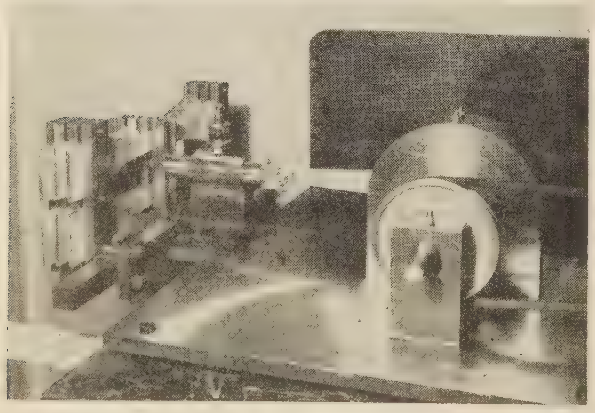
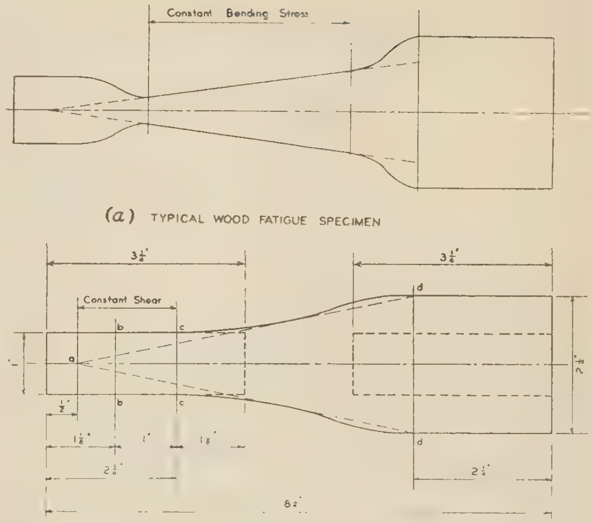


Fig. 2 Cantilever-Type Machine Equipped to Handle Several Specimens Simultaneously



(a) TYPICAL WOOD FATIGUE SPECIMEN

(b) TYPICAL SHEAR FATIGUE SPECIMEN

Fig. 3 Types of Specimens Tested

Machines

Cantilever-type machines were employed because these were more adaptable to flat stock than rotating-beam machines, and because both bending and shear stresses are induced in cantilever specimens. In the earlier work, the machines, as shown in Fig. 1, handled one specimen at a time. Because of the large number of specimens involved, however, and because of the length of time required for some of the tests, a machine was constructed which could handle as many as twelve specimens at one time, as illustrated in Fig. 2.

In this apparatus the specimens, in groups of two or four, are clamped between upright pairs of cold-rolled steel plates which are in turn bolted to a heavy horizontal steel bar fastened to the steel base plate supporting the entire apparatus. The free ends of the specimens are forced back and forth by a pusher designed to hold the specimen firmly but to allow the end to deflect freely. The pusher is attached to a driving rod pinned to a connecting rod in turn mounted on an eccentric which forms part of a heavy steel driving wheel. The eccentric can be adjusted to give a throw ranging from zero to $1^1/_2$ in. The driving wheel is mounted in a heavy bronze bearing and is belt-driven by a 1-hp motor. The machine can be run at various speeds, but was generally operated at 1100 to 1300 rpm.

Types of Specimens

Alternating Stress. Two types of specimens were employed as shown in Fig. 3. To study the fatigue behavior of laminated wood and plywood, the shape shown in Fig. 3(a) was selected and found generally satisfactory. It provided a considerable

length over which the bending stresses were constant, and still allowed enough taper to prevent splitting in the outer veneers. Tests were run at uniform amplitude, and length was varied in accordance with the stresses desired. A specimen of this kind is shown in Fig. 1. It has failed in the tapered portion.

A different specimen was required to study the behavior of the adhesives under alternating shear stresses. In this instance, the specimen was also designed for flexing as a cantilever and was of such a shape as to provide a constant shear-stress area 1 in. sq at the loaded end while the strength at the supported end was sufficient to avoid overstressing the outermost fibers in bending.

The specimen is illustrated in Fig. 3(*b*). Load was applied at *a*. The specimen was 1 in. wide to point *c* where it was flared as shown to line *d-d*. From *d-d* to the end, the specimen was held in the grips during test. The distance *b-c* was 1 in. and the area *bbcc* was the 1-in-sq area over which uniform alternating shear was induced by the flexing of the specimen. After being run through a given cycle of stress, a glue-line shear test specimen was cut from this portion of the specimen as shown. The far end of the specimen was made long enough to provide sufficient area for rigid clamping and to provide a second glue-line shear test specimen with an unstressed central portion for comparison with the stress specimen.

Five alternating-stress test specimens were cut from each panel, as shown in Fig. 4. From the end of each specimen a third glue-line shear test specimen was cut to provide a second comparative test for the stressed sample. Each panel consequently yielded five "fatigue" specimens, five vibrated glue-line shear specimens, and ten unvibrated glue-line shear specimens.

Glue-Line Shear Test Specimens. As shown in Fig. 3(*a*), glue-line shear test specimens were the standard type commonly employed in the testing of plywood (7). They were tested in a briquette-testing machine equipped with a set of standard glue-line shear-testing jaws. Specimens were loaded at the rate of 600 to 800 lb per min.

Static Controls. To provide standards of comparison, simple static bending specimens 1 in. wide were cut from the panels and tested as center-loaded end-supported beams on spans 20 times the depth of the specimen. From these, modulus-of-rupture values were obtained. In addition, tension test specimens were cut from the same panels as the film-bonded-plywood fatigue specimens, but experience indicated that bending tests provided a better basis of comparison than did tension inasmuch as the alternating-stress tests were essentially bending tests.

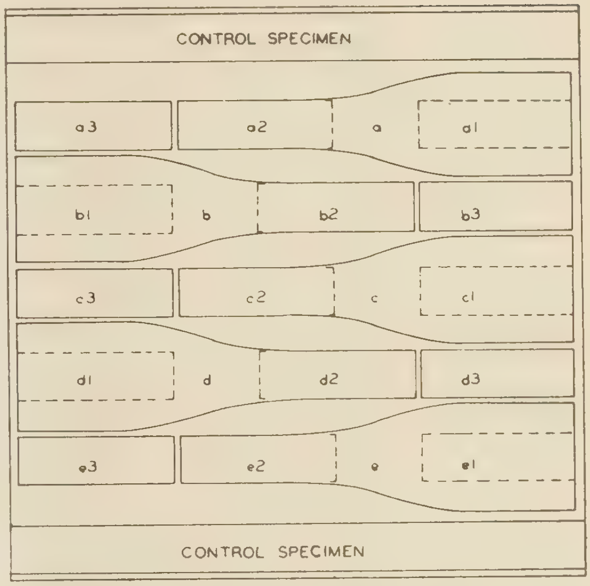


Fig. 4 Cutting Schedule for Alternating and Static Shear Specimens

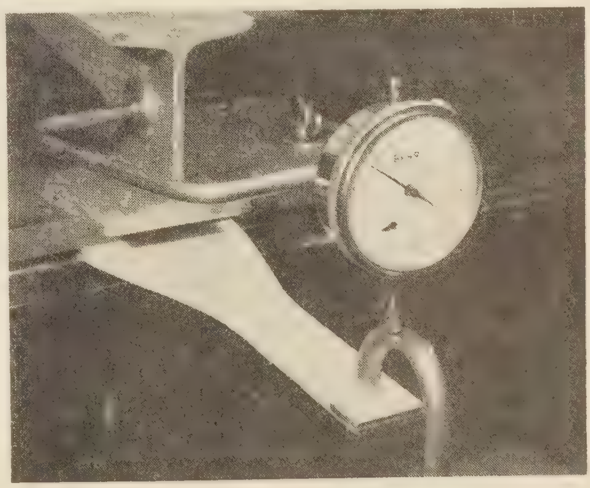


Fig. 5 Calibrating Specimens by Loading With Dead Weights

Testing Procedure

Fatigue Tests. Each fatigue specimen was first calibrated by clamping the fixed end and loading the free end with dead weights to obtain the load-deflection curve. From this the stress for a given deflection could be calculated. The same procedure was employed for calibrating the alternating-shear test specimens and is illustrated in Fig. 5.

The specimen was clamped rigidly to a heavy table, a hook and pan were hung from the free end of the cantilever, and deflections were read to the nearest thousandth inch on the dial, whose stem rested on a small horizontal plane on top of the hook. The hook hung from the loading point of the specimen and dead weights were supported on the pan directly below the loading point.

After calibration, the fatigue specimen was subjected to alternating stress. In the preliminary runs the motor was stopped at rather frequent intervals and the specimen inspected for extent and type of incipient rupture. After a satisfactory shape and size of test piece had been found, specimens were in general permitted to remain in the machine until failure had occurred.

It was necessary to determine some criterion of failure because a piece which had cracked and was carrying no load might hang on for a considerable number of cycles before parting. Consequently, when a crack had formed completely across the width of the piece so that the cantilever to all intents and purposes was merely hinged at the line of the crack, failure was considered to have occurred. In general, it was found that failure, so defined, followed soon after the first appearance of a fatigue crack.

Fatigue stresses ranged from 90 to 20 per cent of the static moduli of rupture of the controls. Because of the variability of the material, four to six pieces of each length were tested and six to eight lengths were employed.

Alternating-Shear Tests. Alternating-shear test specimens were given the same preliminary calibration as the fatigue test specimens. Since these specimens were not vibrated to destruction, it was possible to recalibrate them after vibration to determine if repeated stress had any effect upon the load-deflection curve,

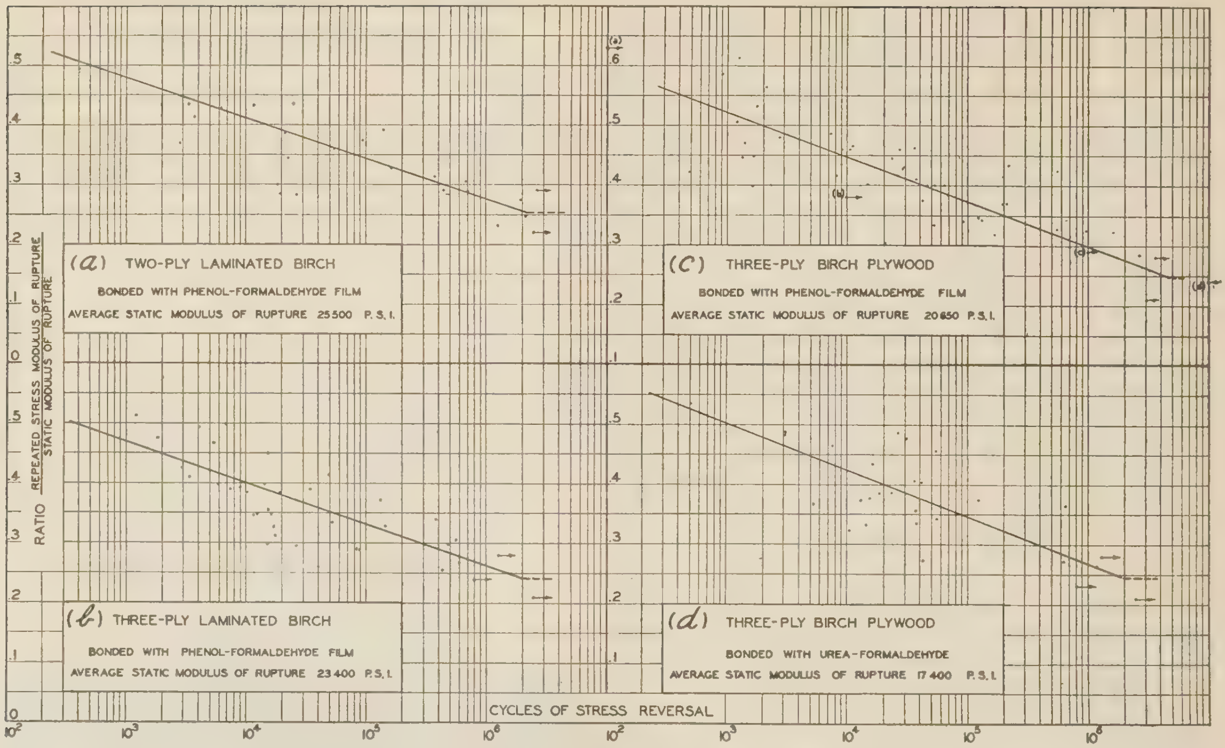


Fig. 6 Summary of Results of Fatigue Tests

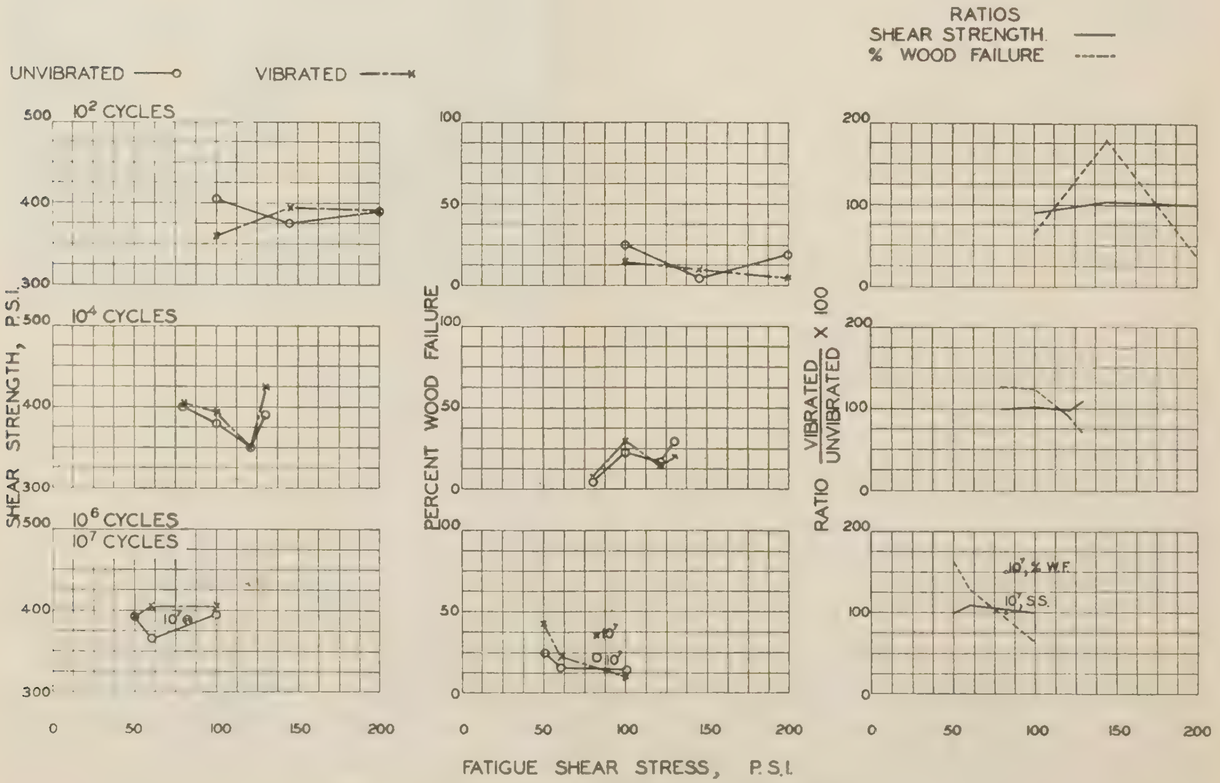


Fig. 7 Results of Tests on Phenol-Formaldehyde Film

TABLE 1 CYCLES AND AVERAGE ALTERNATING SHEAR STRESS IN UNWEATHERED SPECIMENS

Number of cycles	Shear stress, psi: Phenol-formaldehyde film	Phenol-formaldehyde solution: Plain	Phenol-formaldehyde solution: Extended
100	200	200	207
	144	160	153
	101	127	124
		94	95
10000	131	126	126
	123	94	90
	101	79	78
	80	...	...
1,000,000	101	99	92
	62	80	64
	50	67	...
10,000,000	86	59	68

that is, if there had been any change in the modulus of elasticity of the plywood.

Tests were run at 100 cycles, 10,000 cycles, 1,000,000 cycles, and 10,000,000 cycles. At each of these cycles, loads were regulated to induce three or four different shear stresses in the glue lines, in order to ascertain if there was any progressive tendency to weaken the adhesive as the shear stress was increased. The highest load applied at any given number of cycles was the highest that could be employed without causing the specimen to fail in bending fatigue. Specimens were made as short as feasible in order to obtain the highest possible shear stresses. Five to fifteen specimens were tested at each combination of cycles, and shear stress to obtain a dependable average. Cycles and stresses are given in Table 1.

Test Results

Fatigue Tests. Fatigue results are summarized in Fig. 6 (*a* to *d*). Fatigue stresses are plotted as ratios of the static moduli of rupture, and numbers of cycles are plotted on a logarithmic scale to bring out the trend. As is to be expected, a considerable amount of scatter is found in spite of the care with which the material was selected and cut. The general trend is apparent, however, and the averages of the plotted points fall fairly close to the indicated straight lines.

The horizontal dotted lines, at approximately 25 per cent of modulus of rupture, represent averages of specimens which had not failed at 1,000,000 to 3,000,000 cycles of stress reversal. In Fig. 6(*c*), the points marked *a*, *b*, *c*, and *d* represent the averages of the most heavily loaded film-bonded specimens employed in the second phase of this research, that is, the alternating-shear specimens. Each of these points represents the average of five to fifteen tests, none of which resulted in failure. Points *c* and *d*, in particular, fall close to the straight line derived from the fatigue tests.

Periodic examination of specimens during the test runs revealed little tendency for the bond to fail or the veneers to separate. After the outer plies had cracked through to the glue line, that is, after failure as previously defined had occurred, delamination did in some instances take place. This tendency was somewhat more prevalent in the plywood than in the laminated material.

TABLE 2 RESULTS OF ALTERNATING-SHEAR TESTS ON UNWEATHERED SPECIMENS

Cycles of Stress Reversal	Static Controls: Glue-Line Shear Strength p.s.i.	Static Controls: Percent Wood Failure	Vibrated Specimens: Alternating Shear Stress p.s.i.	Vibrated Specimens: Glue-Line Shear Strength p.s.i.	Vibrated Specimens: Percent Wood Failure	Ratios Vibrated to Unvibrated: Shear Strength Percent	Ratios Vibrated to Unvibrated: Percent Wood Failure	Change in Modulus of Elasticity Percent
Phenol-Formaldehyde Film								
10^2	393	20	200	392	7	100	35	-3
	375	5	144	396	9	106	180	-2
	403	25	101	362	16	90	64	0
10^4	392	30	131	428	20	109	67	-2
	353	17	123	353	16	100	94	-2
	381	24	101	393	30	103	125	-5
	400	7	80	403	9	101	128	-1
10^6	397	14	101	405	9	102	64	-
	367	17	62	406	22	111	129	4
	397	26	50	397	43	100	165	0
10^7	380	23	86	385	35	101	152	-5
Phenol-Formaldehyde Solution, Plain								
10^2	380	14	200	365	18	96	129	0
	404	31	160	369	25	91	81	+1
	396	39	127	401	38	101	98	0
	431	25	94	333	2	77	8	0
10^4	398	39	126	344	35	86	90	0
	386	33	94	417	34	108	103	0
	441	32	79	407	7	92	22	0
10^6	412	14	99	368	10	89	71	0
	395	27	80	378	15	96	56	-1
	429	25	67	375	25	87	100	0
10^7	345	31	59	403	45	117	76	-1
Phenol-Formaldehyde Solution, Extended								
10^2	400	74	207	428	49	107	66	0
	447	85	153	450	76	101	89	+2
	446	76	124	436	60	98	79	+1
	461	83	95	414	69	90	84	1
10^4	500	60	126	465	50	93	83	0
	458	63	90	453	72	99	114	+2
	459	75	78	425	74	93	99	+2
10^6	461	82	92	425	69	92	84	0
	473	91	64	445	79	94	97	+2
10^7	500	68	79	422	67	84	98	+1

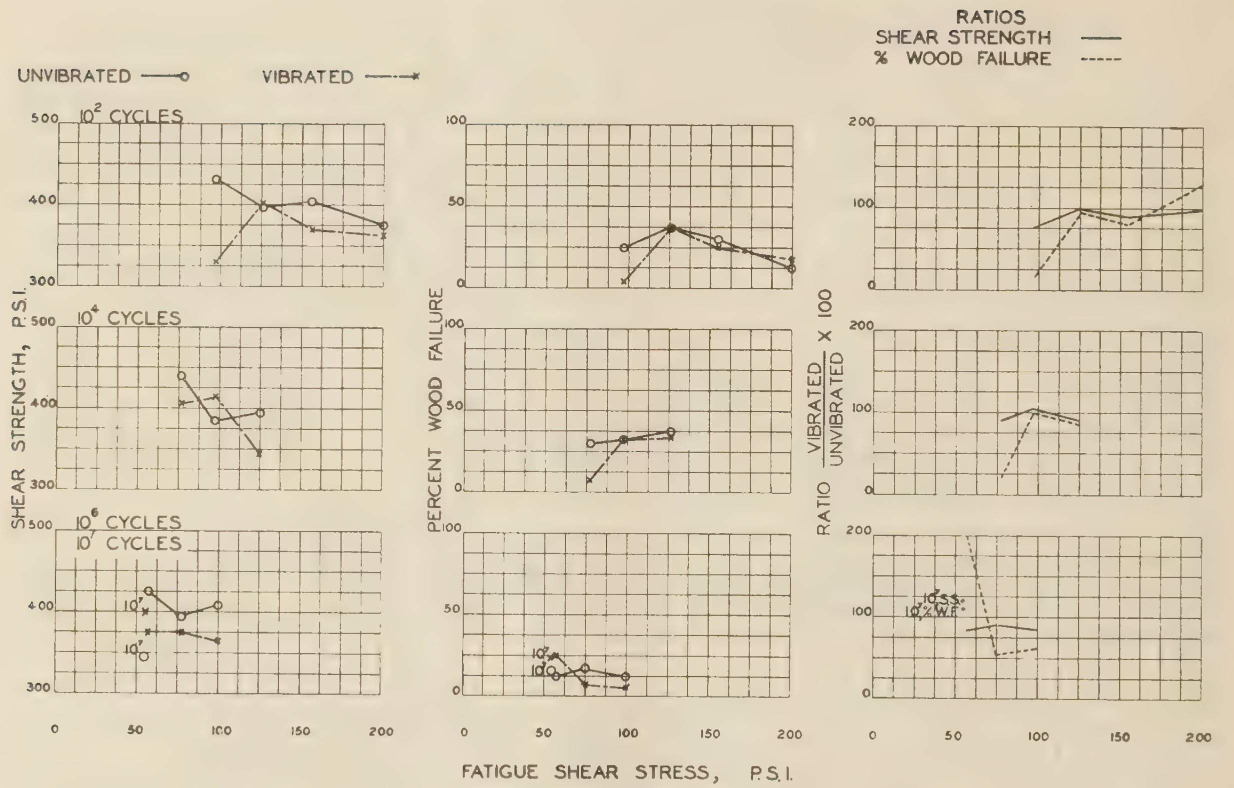


Fig. 8 Phenol-Formaldehyde Solution, Plain

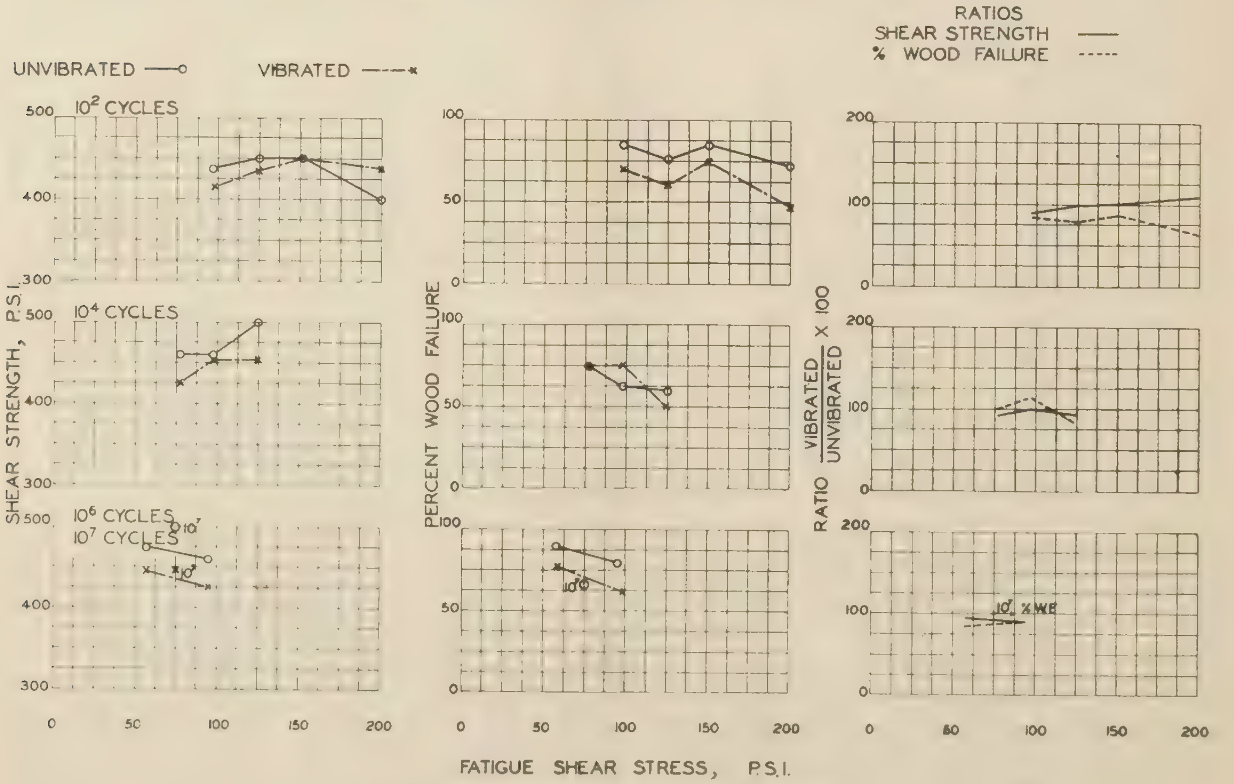


Fig. 9 Phenol-Formaldehyde Solution, Extended

Alternating-Shear Tests. Results are summarized in Table 2 and in Figs. 7 to 9, inclusive. For each type of adhesive and for each number of cycles of stress, values are given for the average glue-line shear strengths and percentages wood failure of the static controls, the average alternating shear stresses at the glue lines, the average glue-line shear strengths and percentages wood failure after vibration, and the percentage changes in modulus of elasticity.

1 Phenol-formaldehyde film (Table 2, Fig. 7): The most striking feature of the results is that there is no loss in glue-line shear strength after stressing in alternating shear. This is best shown by the ratios of glue-line strength of stressed to unstressed specimens in Fig. 7. The lines lie very close to the 100 per cent mark, in some instances even rising slightly above it. The ratios are strikingly constant in spite of the variation in the actual glue-line strengths of the individual panels. Equally striking is the manner in which the material holds its stiffness. The secondary calibrations show very little change in modulus of elasticity, the largest average for any panel being 6 per cent. A few recalibrations seem to indicate a small increase in stiffness but this is undoubtedly more apparent than real. For all practical purposes, it can be said that there is no significant change in modulus of elasticity under the conditions of alternating stress employed in this research.

No definite trend in percentages wood failure is discernible. At 10,000 cycles, the more heavily vibrated specimens exhibit lower percentages of wood failure relative to the unvibrated controls than do the more lightly vibrated. This is true of the specimens at 1,000,000 cycles. In both instances, however, the lightly vibrated pieces exhibit higher percentages wood failure than do the unvibrated. It does not seem probable that light stressing in alternating shear would actually increase the shear strength, and the trends seemingly indicated by these two sets of specimens are probably more apparent than real. Furthermore, at 100 cycles the material stressed at 146 psi exhibits much higher percentages of wood failure than do the unvibrated controls while that stressed at both 101 and 200 psi is lower, and by nearly the same amount. At 10,000,000 cycles, the vibrated material shows higher percentages wood failure than does the unvibrated. Finally, the variation in percentages wood failure from piece to piece is considerably greater than the variations in the averages from stress to stress or cycle to cycle.

2 Phenol-formaldehyde solution, plain and extended (Table 2, Figs. 8 and 9): In general, the results are similar to those for the film series, in that no great losses in glue-line strength are observed, as shown by the strength ratios of vibrated to unvibrated specimens. This is particularly true at 100 and 10,000 cycles, where the strength-ratio lines hug the 100 per cent line quite closely.

At 1,000,000 cycles both the plain and the extended materials show strength ratios for the vibrated specimens averaging consistently 7 to 9 per cent less than the strength of the unvibrated material. At 10,000,000 cycles, on the other hand, the "plain" vibrated material runs 17 per cent higher than the unvibrated, whereas the "extended" material runs 16 per cent lower. The unvibrated "extended" controls, however, are considerably stronger than the unvibrated "plain" controls.

Table 2 shows that approximately 75 to 80 per cent of the vibrated specimens run lower in glue-line strength than the unvibrated controls, but the diminutions in strength in no instance exceed 16 per cent, and in most instances are less than 10 per cent. This is less than the variation found within a single panel.

Like the strength ratios, the per cent wood-failure ratios are in 75 to 80 per cent of all instances less than 100 per cent, but here the scatter is more marked, as is to be expected. Generally speaking, however, the drops are small and are less than the variation to be found within a single panel.

Quite striking is the consistent behavior of the "extended" adhesive. In not one instance does the glue-line shear strength, either before or after "fatigue" stressing, fall below 400 psi. Percentages wood failure are correspondingly high. Consequently, although the vibrated specimens generally show a diminution of strength properties, the absolute strength values are higher.

Changes in modulus of elasticity are so small as to be insignificant. The majority of the specimens show no change in stiffness after being subjected to vibration, some show an apparent

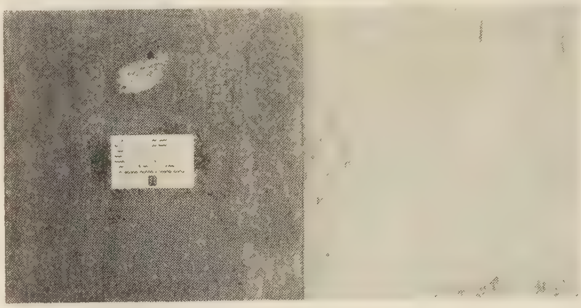


FIG. 10 WEATHERED AND UNWEATHERED PLYWOOD PANELS
(Weathered material has turned color and shows surface checks but gives no indication of disintegration at the glue line.)

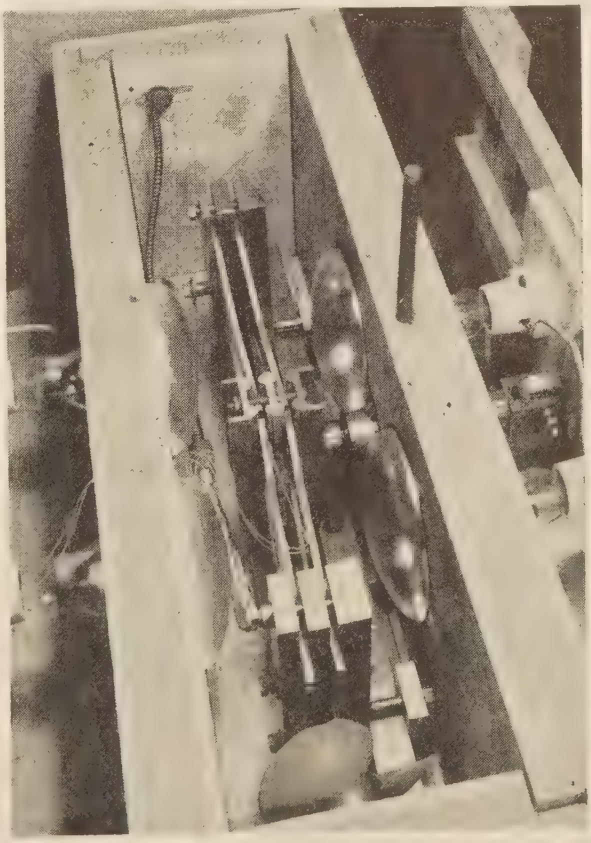


FIG. 11 VIBRATION TEST, HIGH TEMPERATURE
(Four 250-w infrared lamps supply power. Heat controlled in chamber by thermostat, and temperature checked inside specimens by thermocouple.)

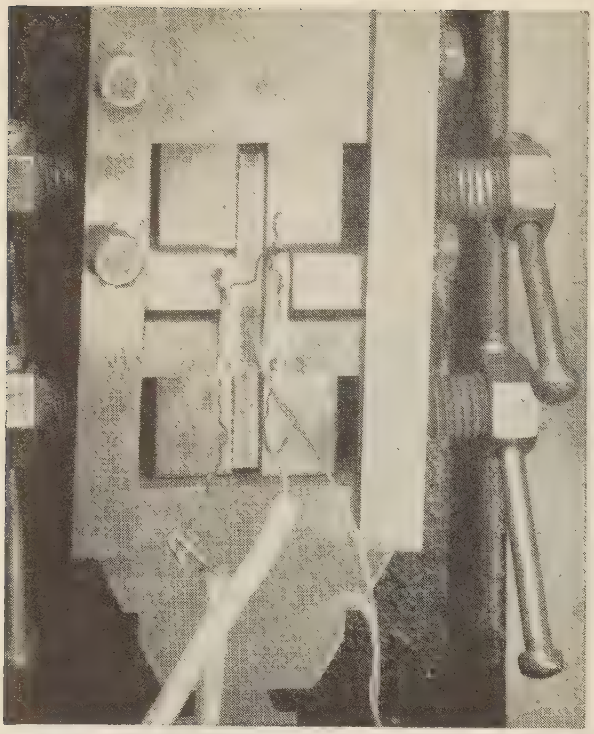


Fig. 12 Plywood Shear Test, High Temperature; Heating Pads and Thermocouple in Place

TABLE 3 AVERAGE ALTERNATING-SHEAR STRESS IN WEATHERED PLYWOOD

Temperature, deg F	Adhesive	Shear stress, psi — Number of cycles: 10,000	1,000,000	10,000,000
— 70	Film	127	83	57
	Solution:			
	Plain	125	81	65
	Extended	121	77	65
+ 70	Film	134	84	66
	Solution:			
	Plain	130	77	64
	Exended	133	78	65
+180	Film	140	100	58
	Solution:			
	Plain	128	95	58
	Extended	125	94	60

increase of 1 or 2 per cent, some a correspondingly small decrease. As a matter of fact, the "extended" adhesive shows more increases than decreases.

TESTS ON WEATHERED SPECIMENS

Materials

Nine plywood panels, similar to those previously described, were exposed to the weather for 6 months, February–July, 1943, on the Florida coast. Three panels were bonded with phenolic-resin film, three with plain phenol-formaldehyde solution, and three with extended phenol-formaldehyde solution. These were fabricated into specimens and tested in alternating shear in the same manner as the unweathered plywood, except that tests were run at the following three temperatures: —70 F, + 70 F (room temperature), + 180 F. This series was designed to ascertain

TABLE 4 RESULTS OF TESTS ON WEATHERED PLYWOOD; 3-PLY, 3/16-IN., ROTARY-CUT BIRCH

Cycles of Stress Reversal Temperature	Static Controls: Glue-Line Shear Strength p.s.i.	Static Controls: Percent Wood Failure	Vibrated Specimens: Alternating Shear Stress p.s.i.	Vibrated Specimens: Glue-Line Shear Strength p.s.i.	Vibrated Specimens: Percent Wood Failure	Ratios Vibrated to Unvibrated: Shear Strength Percent	Ratios Vibrated to Unvibrated: Percent Wood Failure	Change in Modulus of Elasticity Percent
Phenol-Formaldehyde Film								
10^4 -70°F	297	50	127	288	65	97	130	-
+70	467	20	134	471	35	101	175	-5
+180	365	35	140	428	10	117	28	-1
10^6 -70	331	55	83	382	75	115	136	-5
+70	457	15	84	553	5	121	33	-2
+180	421	15	100	372	50	93	330	-
10^7 -70	301	5	57	322	15	107	300	-7
+70	447	25	66	533	25	117	100	+2
+180	404	15	58	452	5	112	33	-
Phenol-Formaldehyde Solution, Plain								
10^4 -70	361	65	125	347	45	96	69	-
+70	476	35	130	484	25	102	72	-2
+180	385	30	128	435	30	113	100	+1
10^6 -70	384	65	81	291	5	76	8	-3
+70	425	25	77	435	20	102	82	-1
+180	396	30	95	458	10	116	33	-
10^7 -70	338	45	65	345	25	102	56	-6
+70	429	45	64	411	15	96	33	-2
+180	448	15	58	465	10	104	67	-4
Phenol-Formaldehyde Solution, Extended								
10^4 -70	409	90	121	404	100	99	111	-
+70	560	70	133	528	80	94	114	-6
+180	431	75	125	450	90	104	120	-3
10^6 -70	442	85	77	365	70	83	83	0
+70	550	85	78	526	85	96	100	0
+180	514	75	94	547	65	106	87	-2
10^7 -70	379	85	65	367	95	97	112	-4
+70	524	85	65	552	70	105	83	-5
+180	487	95	60	552	95	113	100	-5

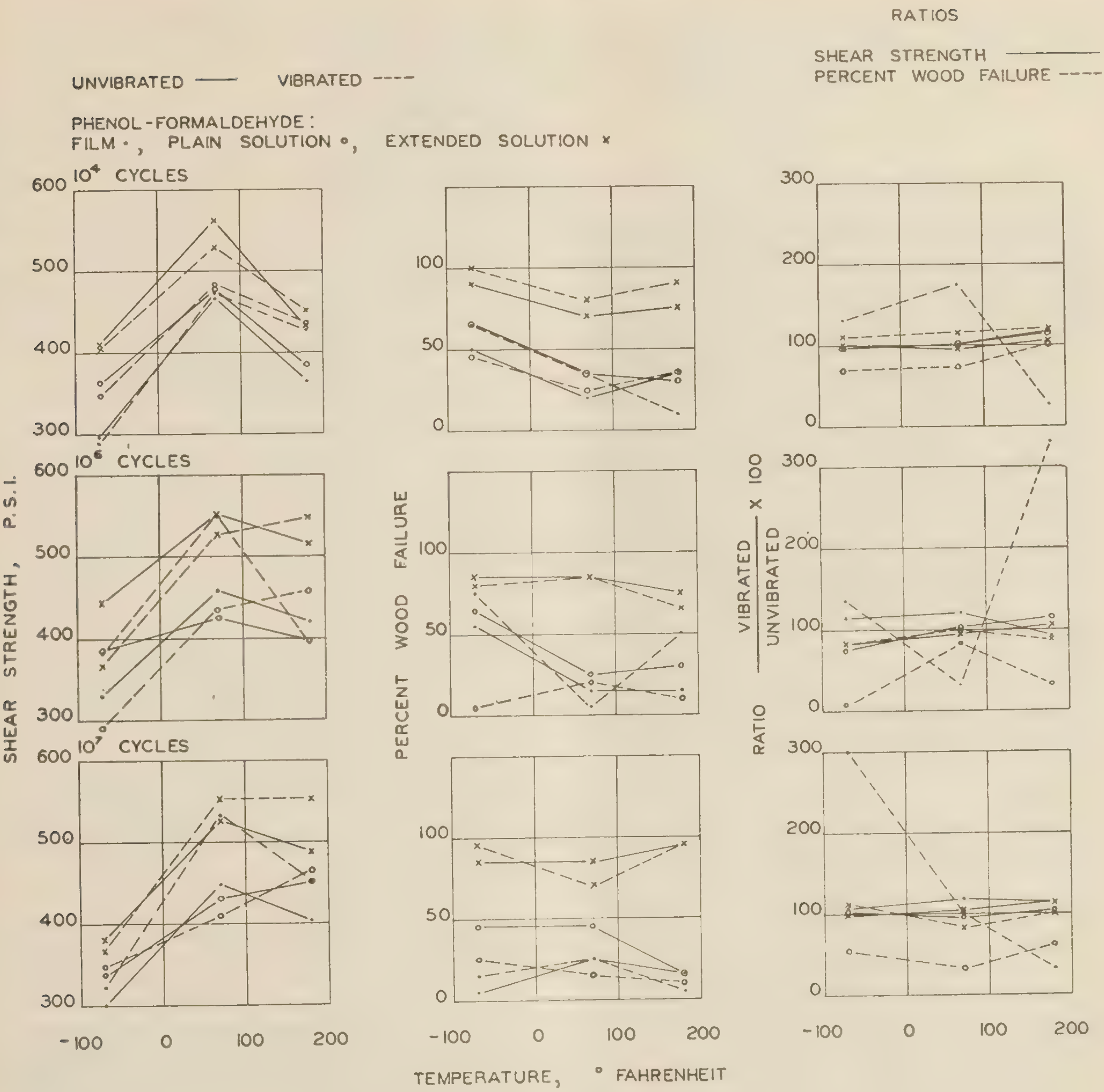


Fig. 13 Results of Tests on Weathered Plywood

if weathering, high temperatures, and low temperatures have any effect upon the strength properties of phenolic-resin-bonded plywood. Typical weathered and unweathered panels are shown in Fig. 10.

High-Temperature Tests

Vibrating. The pusher and specimen holders of the fatigue machine were surrounded with an insulated cabinet, as shown in Fig. 11, equipped with four 250-w infrared lamps. Lamps were arranged in two parallel-series pairs, and connected to a 110-v a-c main. Power supplied was controlled by two rheostats, one permanently in the line, and another, in parallel with the first, automatically switched in and out by an aluminum-foil-coated thermostat, set to maintain 180 F. Temperature inside the specimens was checked by thermocouples, and the machine was not started until this temperature had risen to +180 F. A small fan maintained air circulation.

Plywood Shear Tests. Small electrical-resistance heating pads, as shown in Fig. 12, were made to be slipped into the standard plywood shear jaws adjacent to the center of the specimen. Pads were designed to heat and to hold the specimen at 180 F. Temperatures were checked by a thermocouple inserted into a small hole drilled in the center ply. Power was controlled by Variac.

Low-Temperature Tests

Vibrating. The fatigue machine was placed in a room maintained at —10 F to —20 F, and the insulated cabinet around the specimens was packed with dry ice. It was found by thermocouple readings that this arrangement permitted average temperatures of —70 F to be maintained in the specimens.

Plywood Shear Tests. Specimens were packed in dry ice, and the shear apparatus was placed in the —10 to —20 F room. Experiment showed that specimens removed from dry ice and placed in the shear tester rose to —70 F during the short time needed to make the tests.

Test Results

Because of the limited material available, tests were run at 10,000 cycles, 1,000,000 cycles, and 10,000,000 cycles only, and at only one shear stress for each number of cycles. Shear stresses were somewhat lower than the maximum values employed with the unweathered material, because weathering had worn away some of the surface plies and thereby reduced the maximum load which the plywood could sustain. Even with the reduced stresses, a few specimens showed incipient wood fatigue failures at the completion of the alternating stress cycles.

Cycles, stresses, and temperatures are summarized in Table 3. Results of all weathered plywood tests are presented in Table 4 and in Fig. 13.

Effect of Weathering. Comparison of Tables 2 and 4 reveals that weathering causes no glue-line strength to be lost at room temperatures. As a matter of fact, all three types of adhesives exhibit higher room-temperature strengths in the weathered than in the unweathered material. This is true of both vibrated and unvibrated specimens. Percentages wood failure in weathered and unweathered, vibrated and unvibrated, material are comparable.

Effect of Temperature. All three adhesives drop off in strength, as a rule, at both low and high temperatures, with the greater diminution occurring at —70 F. Again, this is true of both vibrated and unvibrated material. There is no definite trend in percentages wood failure; individual deviations are much more marked than any average trends.

At —70 F, the plywood shear strength of unvibrated phenolic film drops approximately one third, plain phenolic solution approximately one sixth, and extended phenolic solution approximately one tenth. At +180 F, all three drop approximately one tenth. Vibrated film and extended-resin solution both drop about one third at —70 F, and plain resin solution about one fifth. At +180 F, the vibrated film shows a drop of about one fifth, but the solutions maintain their strength at all but 10^4 cycles (heavy stress, large amplitude).

Effect of Vibration. As in the unweathered material, vibration does not affect the glue-line strength at any of the three temperatures. Generally speaking, the ratios of strengths do not deviate greatly from 100 per cent, with the vibrated material testing stronger, if anything, than the unvibrated. Variations in individual percentage wood failures are much larger than any average trends, so that no general conclusion can be drawn. Although vibration appears to cause a small decrease in modulus of elasticity, the apparent difference is too small to be of practical significance.

Conclusions

Fatigue Specimens. The fatigue tests indicate that, in birch plywood and laminated wood, bonded with thermosetting phenol-formaldehyde film, and in plywood bonded with an aqueous solution of cold-setting urea-formaldehyde resin, at room temperatures (70 to 80 F):

1 Fatigue failures are primarily wood failures.

2 Delamination of the veneers occurs very seldom before the outer plies have given way.

3 Laminated material may be expected to withstand at least 2 000,000 and film-bonded plywood at least 10,000,000 stress reversals without failing, when stressed to 25 per cent or less of the static modulus of rupture.

Alternating Shear Stress. It may be concluded that at room temperature (70 to 80 F):

1 Alternating bending stresses inducing alternating shear stresses up to 200 psi at 100 cycles, up to 125 psi at 10,000 cycles, up to 100 psi at 1,000,000 cycles, and up to 85 psi at 10,000,000 cycles may be expected to decrease the glue-line shear strength of birch plywood not more than an average of 7 to 9 per cent, and the percentages of wood failure by approximately the same amount.

2 Absolute values of strength of phenol-formaldehyde film-bonded plywood and plywood bonded with plain and extended aqueous solutions of phenol formaldehyde may be expected to run within approximately 10 per cent of each other.

3 Vibrations inducing the stresses just given have no discernible effect upon the stiffness (modulus of elasticity) of the material.

Weathering. Weathering has no adverse effect upon the glue-line strength of phenolic-resin-bonded plywood. This is true of percentage wood failure as well, and of both vibrated and unvibrated material.

Temperature. The following may be concluded concerning the effect of temperature:

1 Temperatures as low as —70 F cause a loss in glue-line shear strength in materials bonded with all three phenolic resins. Percentage wood failures do not indicate clearly whether the loss in strength is caused by the effect of cold upon wood or upon resin. Film-bonded material loses approximately one third of its strength; plain-phenolic-solution-bonded material approximately one tenth to one sixth and extended-phenolic-solution-bonded material approximately one tenth to one third in unvibrated and vibrated material, respectively.

2 Temperatures as high as +180 F cause losses in glue-line strength but these are less marked than at low temperatures. All three, unvibrated, drop approximately one tenth. Vibrated film-bonded drops about one fifth, the others show little or no loss in strength.

Acknowledgments

The authors acknowledge their indebtedness to the individuals who assisted in this program, particularly to Mr. Harry Majors, Jr., Instructor in the Department of Mechanical Engineering at the Massachusetts Institute of Technology, and Mr. John D. Barry, Research Assistant, who carried the brunt of the experimental work. Dr. Wm. F. Murray of the Department of Mechanical Engineering was instrumental in devising the testing equipment. Messrs. S. N. Tu, M. Becker, and F. Carroll assisted in the testing.

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Maintenance of Hydroelectric Generating Units

By GEORGE H. BRAGG,[1] SAN FRANCISCO, CALIF.

This paper covers the complete overhaul of a 40,000-hp vertical Francis-type hydraulic turbine, operating under 454 ft head to drive a 35,000-kva generator at 257 rpm. Some of the work done was to repair normal wear due to operation over a 20-year period. Other work was of a corrective nature modifying the original design to include improvements found desirable after prolonged operation. Interesting problems were met and solved in some of the work, which involved the handling of heavy parts, and in the use of materials not originally included in the design of the unit.

MODERN hydroelectric generating units usually perform with such reliability that high praise should be given to the manufacturers' engineers who have progressively incorporated improvements in their design and produced machines which often operate continuously year after year without service interruption.

A typical case history usually records only minor annual maintenance work such as repairing the cavitated areas and cleaning the generator windings, etc., all of which may be done in a comparatively short time and without disassembling the unit. But of course there comes a time when a general overhauling is imperative, in which the parts are completely dismantled for careful examination. Then, unusual conditions may be discovered, some of which are due to "wear and tear" while others to "shortcomings" in manufacture.

A good illustration of this is a general overhaul job on a 40,000-hp Francis-turbine-driven unit which had been in continuous operation for almost 20 years. Of course minor repairs had been made on it from time to time but it had not been previously disassembled.

The items of work involving problems of interest were as follows:

(1) Pitting of runner due to cavitation; (2) damaged shaft bushing; (3) shaft packing; (4) worn guide vanes; (5) guide-vane lubrication; (6) runner tip; (7) cavitated liner plates; (8) shaft journals; (9) cavitated relief valve; (10) generator-stator winding.

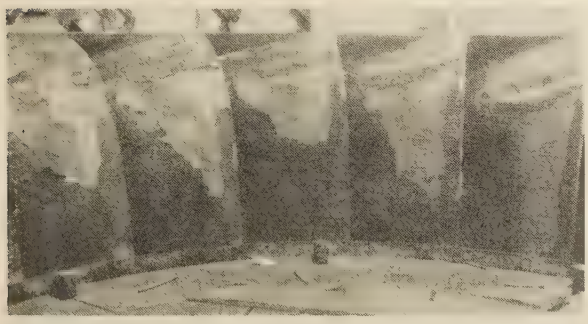

FIG. 1 PITTED AREAS HAVE BEEN ARC-WELDED BUT NOT FINISHED

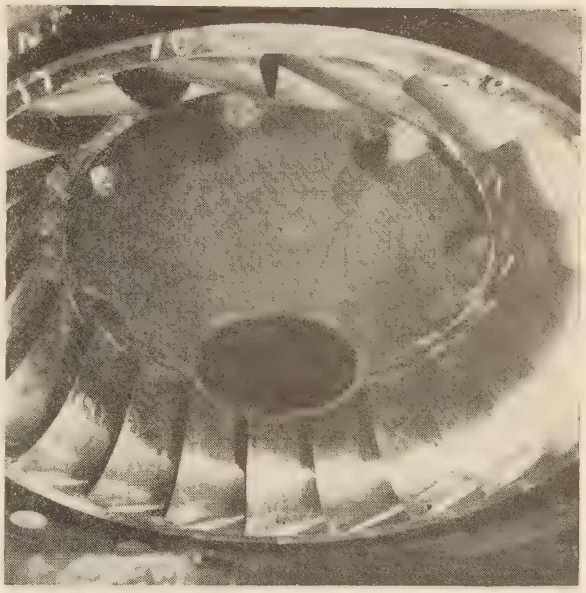

FIG. 2 RUNNER IN PLACE AFTER ALL REPAIRS HAVE BEEN COMPLETED

Pitting of Runner Due to Cavitation. It is rarely possible for a hydraulic-turbine runner to operate over an extended period without pitting of certain areas due to cavitation. The extent and length of time required for pitting to become serious varies greatly but, in general, it is possible to keep it under control by restoring the affected areas by welding at intervals of sufficient frequency to keep the amount of welding at a minimum.

The repair of the pitted areas on the runner vanes required no unusual technique, aside from the precaution to warm the entire casting to 100 C before applying the arc. As no normalizing oven was available to relieve internal strains resulting from localized excessive temperature, the welding was not done continuously in one place but was carried on by skipping from one part to another doing small patches at a time.

In preparing the surface for welding, the spongelike metal is chipped out of the pitted area. The chipping may be as much as $^1/_2$ in. deep in some spots and feather out to the edges of an area which may be 20 or 30 sq in. in extent. With a $^5/_{32}$-in. welding rod a single bead of metal is deposited on a section about $^1/_2$ in. $\times$ 2 in., after which the welding is shifted to another blade while the first section cools. When the entire area has been covered with a single bead, it is peened and any slag is chipped off, after which the second layer is added and so on until the required contour is built up. Rotary files applied to the welded surface remove the excess material faster than emery-wheel grinders.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

Fig. 3 Shaft Bushing Worn by Packing

Fig. 5 Stainless-Steel Band Welded in Place

However, the surface was finished by emery-wheel grinding, Figs. 1 and 2.

Shaft Bushing. As originally constructed, the turbine shaft was provided with a bushing at the point where the shaft passes through the stuffing box in the turbine head cover. This bushing was about 15 in. long and made from carbon steel $1^{1}/_{2}$ in. thick. On account of the flanges on either end of the turbine shaft, it was necessary to make the bushing in halves and join them together by means of special keys having an H-section. Because of the fact that the bushing was made of carbon steel operating submerged in water, deep grooves were found worn into the surface of the metal against which the segmental lignum-vitae packing bore, Figs. 3 to 6, inclusive.

After considering an entirely new bushing made of noncorrosive metal, the cost of which would have been in excess of \$800, it was decided not to disturb the worn bushing but to repair it in place.

Accordingly, the turbine shaft, with bushing in place, and weighing approximately 10 tons, was supported horizontally in its own guide bearings on the floor of the powerhouse. A motor-operated worm wheel was bolted to a flange, which revolved the shaft at a proper lathe cutting speed. A compound slide rest with tool post holder was then set in position on T-rail cribbing and a rectangular recess was turned in the bushing, removing all the irregular worn material. The dimensions of the recess are width 4 in., depth $^{1}/_{2}$ in.

The ring to fill the recess was made of bar nickel steel which measured 4 in. $\times$ $^{5}/_{8}$ in. It was rolled into a circular form of proper diameter, cut into halves, fitted into the recess and secured by welding it circumferentially to the bushing. Both longitudinal joints were then welded thus making a band inserted into the recess.

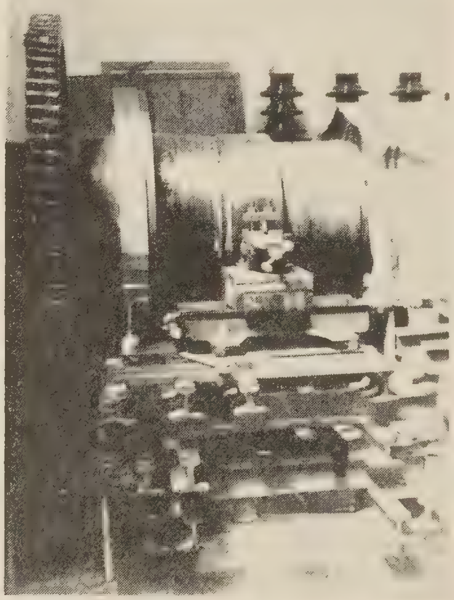

Fig. 4 Motor-Operated Worm Drive Attached to Flanged End of Shaft, Also Compound Slide Rest Supported on T-Rail Crib

Fig. 6 Stainless-Steel Surface Was Given a Mirrorlike Finish

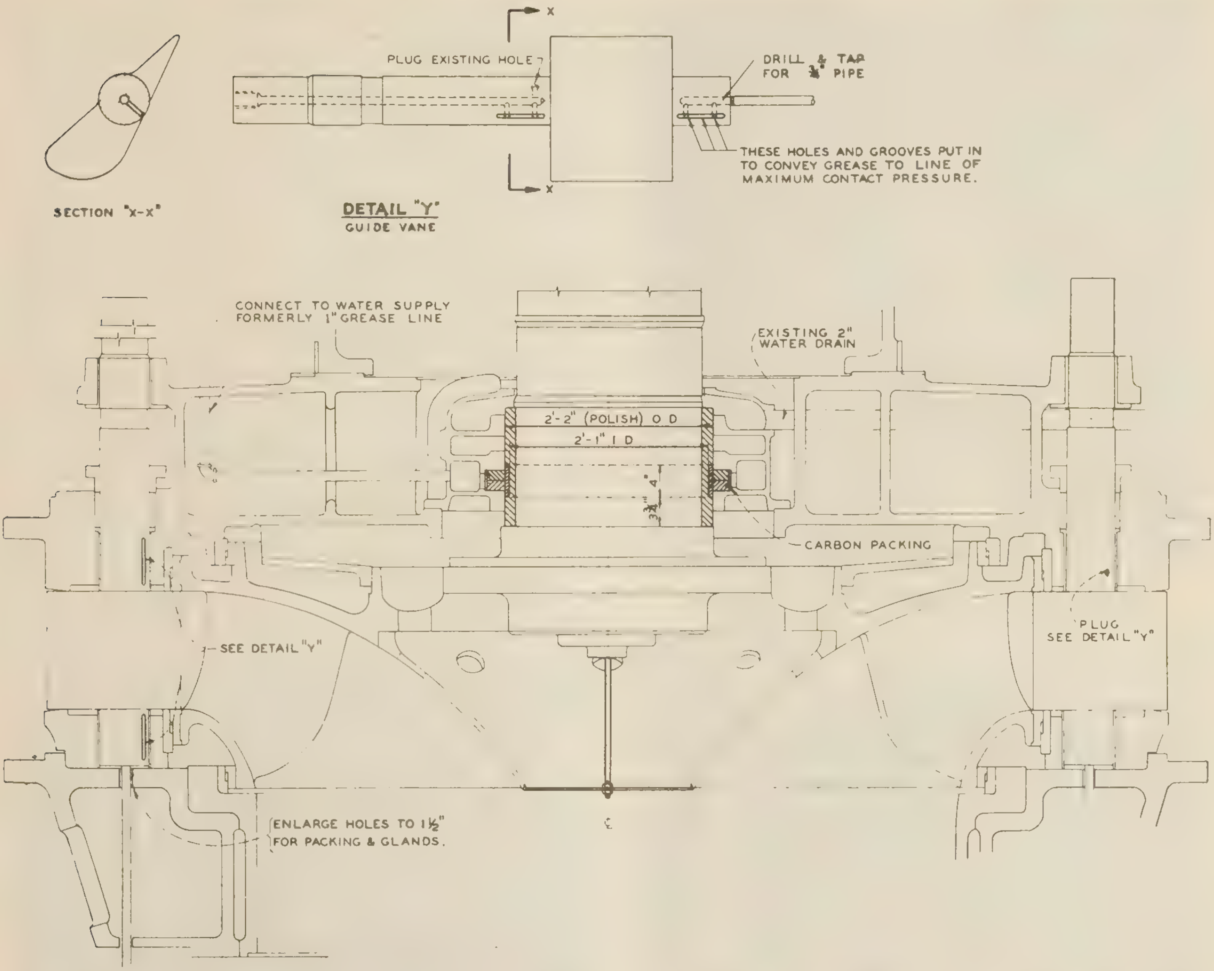

Fig. 7 Repair and Reconstruction of 40,000-Hp Francis Turbine

Fig. 8 Guide Vane Repaired by Arc-Welding Steel to Worn Edges

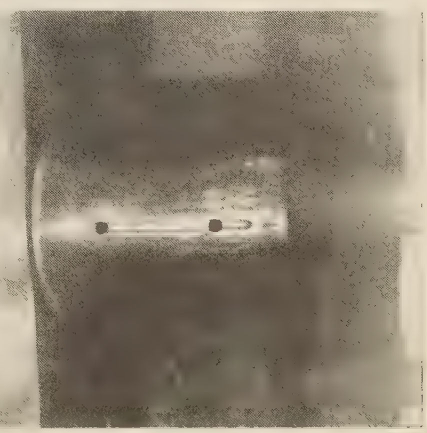

Fig. 9 Guide-Vane Stem Showing Grease Groove to Lubricate Along Line of Maximum Pressure

The nickel-steel band was then turned, trued, ground, and polished to a mirrorlike finish to minimize the friction and wear on the carbon packing which was installed to replace the lignum-vitae packing.

Shaft Packing. Originally, the shaft packing, serving as a water seal in the turbine head cover, was made of two segmental rings of lignum vitae held in position by means of circular retaining springs. Lignum vitae has certain disadvantages but up to a

Fig. 10 Shaft Journal Marred by Shaft Current

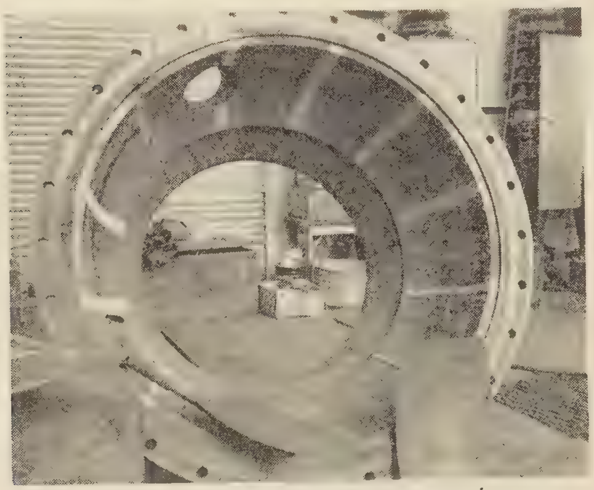

Fig. 12 Steel Plates $^1/_4$ In. Thick Preformed and Fitted to Cover Pitted Surface

few years ago was the best available material for such installations. In recent years, micarta and carbon have become available and offer many advantages over the lignum vitae. In this case, it was decided to use carbon as being the most durable material of the three and the least severe on the metal against which it is pressed, Fig. 7.

The carbon rings were each made of six segments of special carbon having a crushing strength of 20,000 psi and a Brinell hardness of 180 ±. The segments were purchased from the manufacturer in strict accordance with the dimensional drawings submitted. No cutting or fitting was necessary on the job.

Guide Vanes. Ordinarily the most wear on the guide vanes occurs on the edges which are in contact with the throat liner plates, but if some sand is carried by the water, erosion takes place on the streamlined surfaces. In either case, the missing metal may be replaced by arc-welding and the original shape restored by resorting to the manufacturer's methods in fabricating new ones from castings or forgings. Such work can be expertly done by the plant mechanics if a lathe of adequate size to swing the guide vanes is available in the shop, Fig. 8.

Guide-Vane Lubrication. In detailing the guide-vane lubricating system, designers sometimes overlook the fact that the place to exude the grease is between the bearing surfaces which are most intimately in contact for otherwise the lubricant flows in the path of least resistance where it does not serve its purpose.

When upon dismantling the guide vanes the stems are found to be dry and worn where the pressure has been the greatest, the greasing system should be reconstructed by providing a recess on the surface of the stem at top and bottom into which the grease may flow from the central duct to the contact surface between the stem and the bushed bearing. Then if the vanes are moved while the grease is being pumped, the surface will be "wiped" and lubricated without waste, Figs. 7 and 9.

Runner Tip. The runner tip is a separate hollow conical casting attached to the discharge side of the runner, forming a continuation of the upper shroud. The vent holes in the center of the runner were ported within the conical tip and the leakage water from the top side of the runner escaped to the draft tube from the center of the hollow cone.

Notwithstanding close running clearances and a minimum quantity of seal-ring water, a back pressure of 10 psi was built up on top of the runner at full load which subjected the thrust bearing to an additional load of about 75,000 lb.

After analyzing these facts, it was concluded that the rotating conical tip functioned as a centrifugal-pump runner and actually caused the back pressure which retarded the free discharge of the seal-ring water into the draft tube.

Although the thrust bearing had operated at normal tempera-

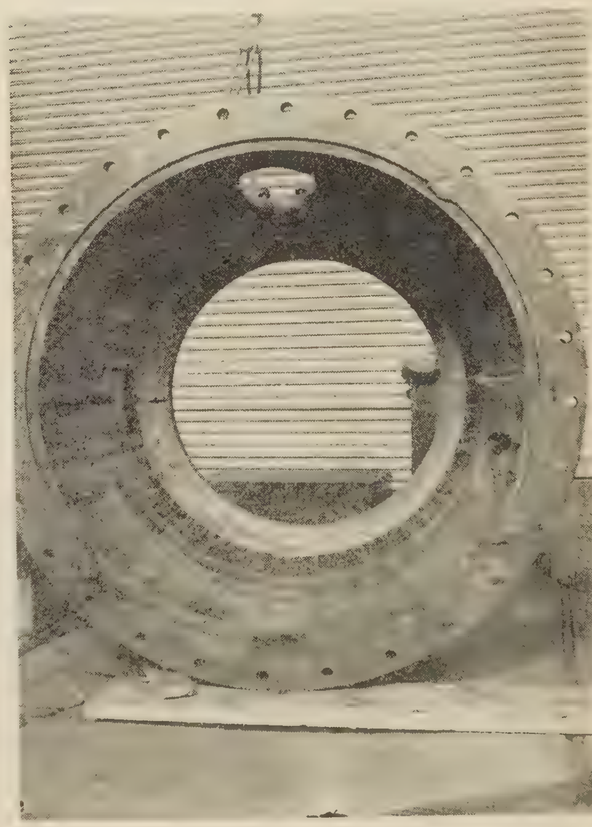

Fig. 11 Pitted Body of Relief Valve of 40,000-Hp Francis Turbine

ture at all times, it was deemed advisable to relieve it of the extra load by providing holes in the tip opposite those in the runner and a plate which closed the center opening. After doing this, the seal-ring water escaped to the draft tube through the drilled holes in the conical tip, and thereafter the pressure above the runner was reduced from 10 psi at full load to a negative value at all loads, Fig. 7.

Liner Plates. Although, as previously stated, the turbine had been in continuous operation for almost 20 years, the seal-ring clearances, contrary to usual experience, had not enlarged appreciably.

When first installed, both upper and lower clearances were 0.030 in., and, when measured prior to taking the unit out of service, the upper one was 0.032 in. and the lower one 0.035 in. This is exceptional and may be explained by the fact that the water carries no sand. The seal rings therefore were not replaced, nor was it necessary to remove the throat liner plates. The slight cavitations in them were arc-welded and the surfaces were refinished by rotary files and emery wheels.

Shaft Journals. It is evident by the appearance of the shaft journal that shaft currents may flow without being detected by the operator. The upper guide bearing was insulated but probably had become short-circuited in some unaccountable manner. Such a condition in a vertical unit is not necessarily serious for the bearing temperature in this instance had not been abnormal at any time. The surface was polished by rubbing it with abrasive

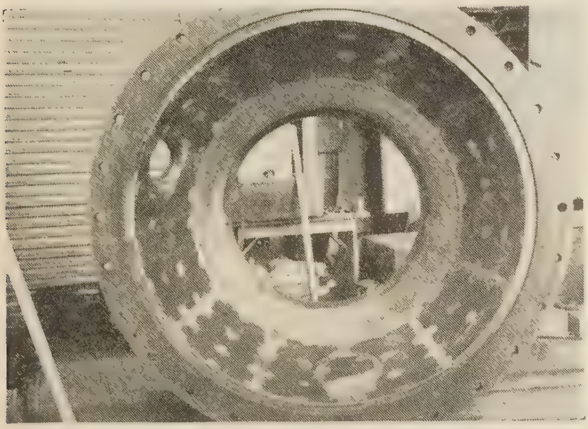

Fig. 13 Steel Plates Completely Line Inner Surface
(Plates were welded along the joints and fastened with patch bolts.)

cloth which lined wooden blocks shaped to fit the journal. The blocks in turn were rotated manually by radial levers. This improved the surface, but of course did not remove the pits, Fig. 10.

Relief Valve. The turbine relief valve invariably pits near the seat rings and, if not allowed to progress too far, effective repairs may be made which will compare favorably with a substitution of new parts for the damaged ones. Irregularities in the bronze seat rings are filled by arc-depositing alloy from a rod having approximately the same composition as that of the parent metal. The contours of the surfaces are restored by filing and grinding according to a template.

The cast-iron body is lined inside by steel plates $^{1}/_{4}$ in. thick preformed to fit the curvature of the casting and attached by means of patch bolts. The joints between the plates are welded after all the plates have been attached, and when the surface has been made smooth by grinding, the completed steel surface will be more resistant to pitting than the original cast iron.

Upon reassembling the valve, the seat-ring surfaces are coated with grinding compound and the valve is pressed against the body

Fig. 14 Strongbacks Attached to Screw Jacks Forced Coils Into Slots

seat as if it were in the closed position. It is then oscillated and rotated manually by radial levers, simulating the technique of grinding in automobile-engine valves. The contact eventually becomes perfect and there will be absolutely no leakage, Figs. 11 to 13, inclusive.

Generator Winding. While the turbine was being overhauled by one crew, all the generator-stator coils were removed and replaced with new coils by another crew. As the coils were too heavy to be handled and manipulated safely, mechanical devices for various operations were adopted. Of these a description of the screw jacks which forced the coils into the slots may be of interest. A vertical pipe was rigidly positioned in the center of the stator core. Four screw jacks were then fabricated into radial arms each of which was attached to the center pipe by a ring which allowed the arms to be rotated about the pipe center. A wooden strongback was then attached to the outer extremities of each pair of arms. As the coils were set in place in front of their slots, the strongback was rotated and the jacks adjusted so that pressure throughout the entire length of the straight portion of the coil was uniformly applied until the coil was finally seated in its slot. This device, Fig. 14, definitely avoided any possibility of applying a strain to the mica insulation which might produce a crack that later would break down when the generator was in service.

By restoring each worn part carefully and making improvements where necessary at stated intervals, generating units should maintain their records of continuity of service indefinitely and give reasonable assurance that failures will not occur at inopportune times.

Discussion

P. M. Hess.[2] This paper, dealing with the complete overhaul of a 40,000-hp vertical Francis-type hydraulic turbine operating

[2] Station Superintendent, Safe Harbor Water Power Corporation, Safe Harbor, Pa.

under a head of 454 ft, should be of interest to all companies having hydraulic generating stations. This is particularly true for stations which were built many years ago. During the last 25 years, however, much improvement of design features has been made, and the use of better materials has to a great extent eliminated the need for much of the repair work mentioned in this paper.

Pitting Repairs. Repairs to pitted areas due to cavitation, as described in this paper, are treated along the lines followed by most companies. However, it is of interest to note that rotary files have been used for the rough-grinding, thereby greatly reducing the grinding time which constitutes an appreciable item of expense.

Shaft Repairs. The work of repairing the worn part of the shaft at the packing gland is interesting, but it is believed that this job could have been done satisfactorily by the use of sprayed stainless steel. Recently, information has become available on the use of carbon packing to eliminate most of the wear at this point.

Wicket Gates. Leakage through the guide vanes or wicket gates is of concern to most companies, especially where corrosive or erosive waters are encountered. On the latest designs at Safe Harbor, the contact surfaces are prewelded with stainless steel which thus far is proving well worth the additional investment. On the extremely large units, where complete dismantling of the unit is a major job, it is desirable to weld these corroded contact areas of the gate in place. This job certainly will be a difficult one without warping the surface. It is believed, however, that some attempts should be made to coat these areas with a noncorrosive hard stainless material in place. There is a possibility of applying sprayed stainless steel to the heel part of the gate, but it is doubtful whether or not a sprayed material will adhere properly to the tip.

Shaft Currents. Much discussion is heard concerning damage to bearings from shaft currents, but it is believed that in vertical hydraulic units there are very few cases on record of guide-bearing failures from this cause. It is desirable to insulate at two points in order to provide additional protection and permit testing.

Correcting Pitting From Cavitation. This paper stresses the necessity of complete overhaul after a period of many years' operation but further states that minor repairs in place have been made from time to time. We are of the opinion that, when dealing with propeller-type units, where much damage can be done to the blades in a period of a year or less, very frequent inspections should be made during this early operating period. Local cavitation, which is due to improper shaping of the blades or poor surface conditions, will show its early presence, if not by actual pitting, by the disappearance of the paint or protective coating on the blades. These local conditions can be corrected by grinding ahead of the pitted section to relieve the high spot and reshaping if due to improper shape.

Fig. 15 of this discussion shows a condition of severe pitting resulting from improper shape of the entrance edge of a Kaplan blade after only 6 months of operation. This pitted area could have been corrected by welding with an 18-8 stainless material, but this would not have corrected cavitation. The cavitation or the cause of the pitting was eliminated by reshaping the entrance edge to eliminate the jump of the water. No welding was resorted to, and after 10 years of operation no more pitting has been discovered.

A similar condition is shown in Fig. 16, where the trailing edge was rounded. This rounded surface set up cavitation close enough to the metal so that pitting occurred and continued progressively. The pitting was corrected by making provision for a rectangular trailing edge so that the cavitation which did result existed in the water sufficiently far away from the metal to prevent attack.

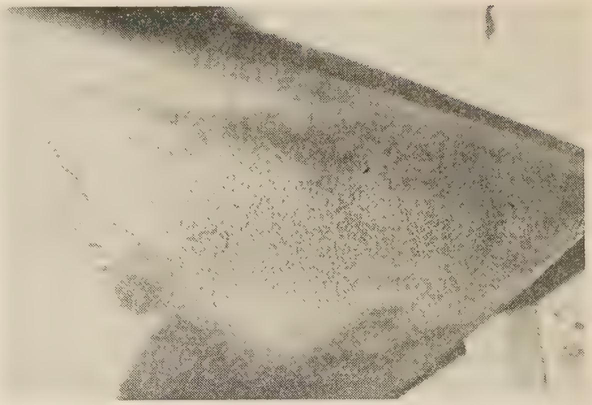

Fig. 15 Pitting on Bottom of Blade

Fig. 16 Pitting on Trailing Edge of Blade

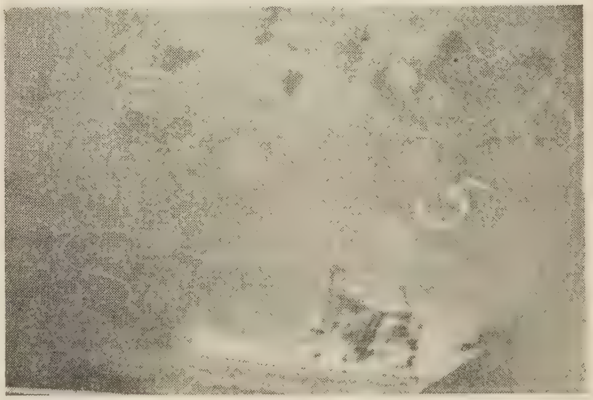

Fig. 17 Pitted Area on Bottom Trailing Side

Fig. 17 shows a condition of pitting at various points due to local cavitation. These areas were corrected by grinding the more or less high section ahead of the pitted area. Here again no weld was required to prevent further pitting.

Recently, one of our newer units showed considerable pitting on the pressure and suction side of the leading edge. Some of this area was prewelded with 18-8 stainless and withstood the attack, but the mild steel pitted rapidly. Investigation showed that a comparatively blunt section of the entrance edge existed,

FIG. 18 PITTING ON UNDER SIDE AND LEADING EDGE OF TURBINE BLADE

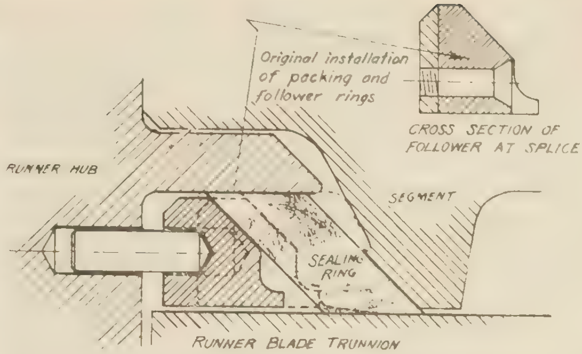

FIG. 19 ORIGINAL RUNNER-BLADE PACKING INSTALLATION
(The dotted lines show how neoprene packing is pressed out of shape and lifted from the trunnion with the original-type installation.)

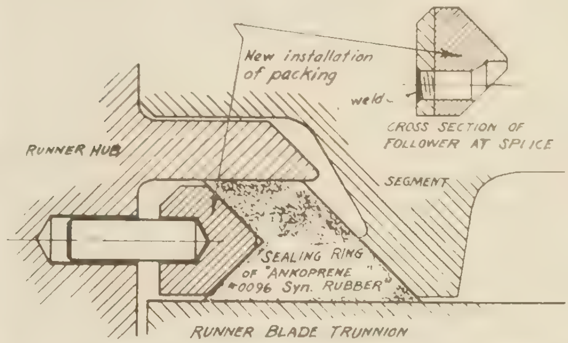

FIG. 20 NEW INSTALLATION OF RUNNER-BLADE PACKING
(Sketch shows how original follower rings were machined.)

causing a tendency for the water to break away from the blade surface. Fig. 18 shows this condition on the suction side of a blade. Here again it would have been possible to combat this pitting by extending the stainless weld over the pitted area; instead, due to the scarcity of stainless welding rod at this time, it was decided to reshape the entrance edge properly. In order to do this, it would have been necessary to grind away practically all of the prewelded stainless steel. Nevertheless, this was done and has almost entirely eliminated the pitting. This is of interest, because pitting, which is shown in Fig. 18, occurred after approximately 1 year of operation, and, after more than a year's additional operation, the reshaped surfaces are still good.

It is therefore suggested that pitting resulting from cavitation should be corrected, if possible, by eliminating the cause, namely, cavitation.

Blade Packing. When the Kaplan units were installed in the Safe Harbor plant, they were many times larger in capacity and dimensions than any units previously built or installed in this country. One problem encountered in the early days was the sealing of the area around the blade journals or shanks to keep the lubricating oil in the hub from leaking outward and the water from leaking inward. A special design, Fig. 19 of this discussion, utilizing a leather packing, was provided for this area, but it never proved entirely satisfactory, growing progressively worse to the point that oil wastage was a considerable factor. Furthermore, it was necessary from time to time to shut the unit down to drain water, which entered at certain loads, from the hub. On the newer designs the water-removal problem has been simplified by the installation of a special pipe within the shaft; also the packing has been redesigned. However, in these units it was extremely difficult to redesign the packing area, and so it was necessary to experiment with different materials and shapes.

After much experimentation, it was found that a neoprene rubber reshaped to the section, as shown in Fig. 20, has practically eliminated any leakage of oil out or water in. It was necessary to secure a neoprene of the proper hardness, because this material has a tendency to become quite hard at freezing temperature and, if made too soft, it will have a tendency to stretch, creep, or swell too much under oil pressure. The neoprene rings shaped similarly to the leather cross section, shown in Fig. 19, were more satisfactory than the leather but did not entirely eliminate all the leakage.

Some of the units have considerable pitting just outside of the sealing ring which caused increased leakage when using the leather. This pitted condition is not as serious when using the neoprene or more pliable material for packing. It has, however, been necessary in several cases to chip, weld, and grind these pitted areas.

The more frequent presentation of similar papers by operating men who are in contact with the equipment in the field is highly desirable, since such information, based upon actual operating experience, has much more value than that to be found in many of the papers which give only opinions without experience.

J. F. ROBERTS.[3] The writer has been greatly interested in the turbines dealt with in this paper, since he spent several months at the plant when it was first started up in 1923. Early in the operation of these units, it was found that there was an unusual vibration which caused a loud humming noise and considerable movement in the exposed parts of the penstocks. The turbines

[3] Manager, Hydraulic Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

themselves operated quite satisfactorily and there appeared to be no unusual vibrations around them.

The penstocks are housed in concrete-lined tunnels underneath the switch structure on the upstream side of the powerhouse, and while they are supported at intervals of 18 to 20 ft, the actual movement of the penstocks in this location and the noise which they produced were quite disturbing.

We feel sure that, in the ensuing 20 years, most other manufacturers have had similar occurrences, and we are not disclosing any professional secrets when we reveal that the cause of these disturbances was in the water wheel itself. We found that the space between the inlet edge of the runner blades and the guide vanes was too small so that, in effect, as each runner blade passed the guide vane, it tended to shut off the flow of water from that opening. This sent a small pressure wave back through the guide-vane opening into the spiral casing, and these pressure waves added together and were transmitted up the penstock. There were 19 runner blades operating at 257 rpm and the pressure waves corresponded exactly to this product, or 4880 vibrations per min. The actual pressure variation was in the neighborhood of $1^1/_2$ psi, or about 0.75 per cent variation on 195 lb static pressure. The difficulty was first eliminated by cutting down the diameter of the runner 5 in. by burning off the inlet edges of the blades. Subsequently, new runners were installed having a smaller inlet diameter and a different number of runner blades, and the results were very satisfactory.

We would be interested in knowing if the author has any records of the change in efficiency of this unit after about 20 years of operation. The original units, as tested by Professor Allen in April, 1925, showed about 90 per cent turbine efficiency. At that time the power company had some venturi connections in the penstock, and they expected to keep a continuous record of the performance of the units. Such records would be interesting to the profession, particularly in view of the rather unusual capacity of these units and the date on which they were installed.

The author mentions the matter of wear on the guide-vane bodies themselves but does not give any figures as to the wear on the guide-vane stems. Apparently the water in the Pit River is extremely clean, as he points out that the runner clearances had not increased appreciably in the 20 years of operation, but some figure on the amount of wear on the guide-vane stems and the bushings would be of interest. The writer has had considerable experience with mountain streams containing silt where the guide-vane stems were worn so badly that they had to be built up by welding and then turned to restore them to the original diameter.

The Pit River units are equipped with 9-ft-diam butterfly valves operating under 450 ft head, the valves having snap rings or piston rings on the disk, in order to give them a tight seal and take up any deflection which occurs in the valve housing under pressure. Originally the leakage on these valves was relatively small, possibly 5 to 10 gpm. It would be interesting if the author could supply information as to how these valves maintained their original tightness.

Illustrations are shown in the paper of the welding on the bottom side of the runner blades which was done to fill in the pitted areas. At the time of the installation of this plant, the question of suction head was not as thoroughly understood as at present, and to our knowledge cavitation tests on model runners had not then been conducted.

The throats of the Pit River No. 1 runners are set about $11^1/_2$ ft above normal tail water. Present practice would require that these units be set down within 4 to 5 ft of the tail water, in order to reduce the sigma value to a figure which is now considered good practice for units of this type.

This powerhouse is located at an altitude of nearly 2900 ft above sea level, and at this altitude perfect vacuum occurs at slightly under 30 ft of water vacuum. With the relatively high draft head which exists, it is not surprising that some pitting has taken place, but the amounts indicated in the paper do not appear excessive in view of the long life of these machines and the operating conditions.

The author's method of correcting the wear in the main-shaft sleeve is interesting and instructive. The use of a high-chrome steel with from12 to 15 per cent chrome is now common practice and appears to give the best results. It stands up considerably better than a nickel-chrome steel. The change in the method of draining the pressure from the upper side of the runner is interesting, and we hope that the results will be available at a later date.

F. H. Rogers.[4] The high-head turbine described in this paper had an excellent record of 20 years of continuous operation before major repairs were required.

The pitting of the runner, as described, occurred on the back of the runner vanes beyond the discharge orifice, which is the usual location for pitting on high-head Francis runners, as it is difficult to guide the water so that it will remain in contact with the back of the vane after the water leaves the orifice formed by the preceding vane.

In recent years, prewelding of the runner vanes has often been done during manufacture to protect the areas where pitting may occur, as determined by past experience. This is now quite general practice in the case of propeller-type runners and has been done in a few cases on high-head Francis runners. The surfaces to be prewelded are cast as depressions about $^1/_8$ in. below the final contour. The initial cost of this work is rather high, and its advisability may be questioned in the case of Francis runners as the actual pitting may not always occur at the protected surfaces.

The cause of pitting in a particular installation may be due to faulty operation as well as to faulty design. The runner is designed for operation under given conditions of horsepower output, total head, and minimum tail-water elevation, as specified by the purchaser. The designer allows some margin over the rated horsepower required to allow for speed regulation, ordinarily from 5 to 10 per cent.

To prevent cavitation, it is necessary that a reasonable absolute pressure exist at all points on the runner vanes. The absolute pressure available is dependent upon the water barometer pressure at the plant site (from 30 to 33 ft, depending upon elevation above sea level). This absolute pressure is reduced by (1) elevation of runner above tail water, (2) regained velocity head of the flowing water, and (3) local pressure drop due to curvature of the vanes, or to the pressure difference required to develop power. All these factors are taken into account by the designer, so that the actual remaining absolute pressure will be from 5 to 10 ft at all points on the vanes.

In actual operation, the turbine may develop a full-load capacity of from 5 to 10 per cent in excess of its rated capacity; the efficiency, however, at this load being somewhat reduced. If the unit is operated at this overload, the discharge will be considerably higher than at the rated output. Under such conditions, the absolute pressure on the runner vanes will be reduced below the value established by the designer and may fall so low as to result in a break in the continuity of the flow causing cavitation and pitting.

This condition is illustrated by the curves in Fig. 21 of this discussion, produced from the actual design of a 60,000-hp turbine operating under a head of 390 ft at a speed of 225 rpm. At the

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normal rating, the discharge is 1500 cfs. At the elevation of this plant above sea level, the atmospheric pressure minus the water-vapor pressure at 80 F is 31.8 ft. This total absolute pressure at the runner discharge is reduced by the three factors previously mentioned, the values of which as shown are as follows:

			Feet
1	Elevation of tail water (static draft head)	=	5.2
2	Average velocity head regained.........	=	11.1
3	Local pressure drop....................	=	9.4
	Total pressure reduction................	=	25.7
	Margin in pressure....................	=	6.1
	Total available pressure................	=	31.8

If this unit is operated at full gate, developing 64,500 hp, or $7^1/_2$ per cent excess power, the efficiency falls to 86 per cent, and

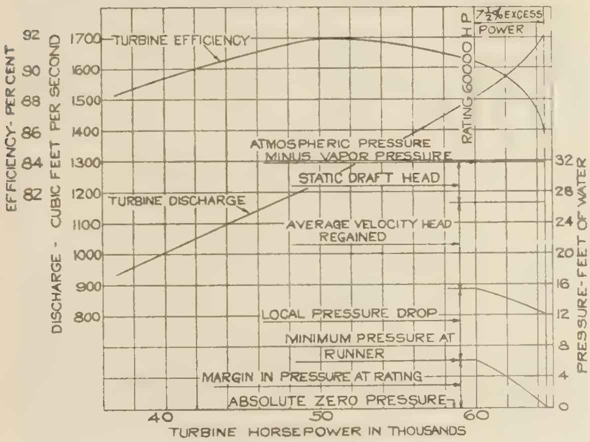

FIG. 21 CAVITATION EFFECT DUE TO OVERLOADS

the discharge increases to 1700 cfs. As both the velocity head regained and the local pressure drop vary as the velocity squared or the discharge squared, it will be noted that these two curves drop rapidly, so that at this output the pressure reduction is made up of the following:

			Feet
1	Elevation above tail water............	=	5.2
2	Average velocity head regained.......	=	14.4
3	Local pressure drop..................	=	12.2
	Total pressure reduction.............	=	31.8

Under these conditions all the design margin is lost and only the water-vapor pressure occurs at the vanes, resulting in cavitation and pitting. To avoid this danger it is good practice to furnish the plant operators with charts showing the maximum allowable outputs at various heads and tail-water elevations, based on data furnished by the manufacturer. Under emergency conditions a unit may be operated beyond these limits for a short period of time, recognizing the fact that pitting may occur, the extent of which would be dependent upon the period of operation. The value of this additional output versus the cost of repairs and outage time should be studied carefully by the operating company before permitting such overloading of the unit.

The repair work on the guide vanes, described in the paper, is frequently found in high-head units where abrasive materials are carried by the water. It is now usual practice to preweld the guide vanes at the manufacturer's plant at certain locations for high-head plants where it is known that the water carries silt or other abrasive materials. This is done by welding with stainless steel on the vertical contact lines between the guide vanes, as this hard metal not only preserves the shape of the vanes at the point of highest velocity but also maintains the leakage at the original low value when the guide vanes are closed. This is of considerable value when shutting down a unit and applying the brakes, as it is frequently found that after the guide vanes are worn the leakage in the closed position is so great that the unit cannot be stopped with the brakes and it is necessary to close the main intake valve. Wear at the top and bottom of the guide vanes also increases the leakage at the distributor plates, resulting in loss in efficiency as well as difficulty in shutting down the unit. This portion of the vanes is sometimes protected by welding stainless-steel plates to the top and bottom of the vanes before final machining. This protective work can be done at lower cost in the manufacturer's shop than after wear has occurred in the plant.

Another method of reducing leakage above and below the guide vanes is the use of the so-called "disk guide vanes." In this type, disks are cast with the guide vanes, machined for a close fit in holes bored in the upper and lower distributor plates. These disks reduce the leakage area above and below the guide vanes by the difference between the diameter of the disk and the diameter of the vane shanks.

T. C. STABLEY.[5] The complete overhauling of the 40,000-hp turbine, as outlined in this paper, is more or less typical of Francis-type turbines. Generally, the most common maintenance problems are pitting of runner by cavitation, lack of grease lubrication at crucial points, water-lubricated bearings, wear of shaft or shaft sleeves at bearings and packing glands, and corrosion.

Another more serious condition, which has appeared in recent years and which is confined mainly to cast-iron runners, is that of cracking of blades at the runner crown, beginning at the outflow edge and progressing toward the inflow edge. On this condition very little has been published.

The California Unit. Repairs and modifications to the 40,000-hp turbine were carried out evidently under favorable conditions, in that it was possible to dismantle the entire unit completely. This method is undoubtedly the most satisfactory and should be used where time and costs permit, as it facilitates and insures a more dependable job, especially so when the runner can be positioned for flat welding instead of the usual difficult overhead method.

Considering the capacity and the extraordinarily high head, the runner evidently is made of cast steel, which lends itself readily to the production of a first-class and enduring welding job of the pitted areas with stainless steel.

The most difficult repair method of welding runners is in the overhead position when it is not convenient to dismantle, especially so with cast-iron runners which are the most common type. Great care must be exercised in these welding repairs to re-establish the original contours by grinding. Experience has proved that if only the general curvature of buckets is followed high spots remain, causing severe pitting to appear well beyond the welded section.

Stainless-Steel Welding. We pioneered with welding stainless steel (18-8 and 18 Cr) on cast-iron runners in the overhead position in 1929 and 1930. Since then we have welded ten large runners with excellent results, so that none of the welds indicates any trace of pitting. In some of our early jobs, insufficient attention had been given to contours, resulting in further expansion of pitting beyond the welds.

Shaft Repairs. Like runner repairs, work on shafts can be more readily carried out when dismantled and set up in a lathe, or in some improvised turning rig as described by the author. In recent years, several interesting articles have been published on

[5] Station Superintendent, Pennsylvania Water & Power Company, Holtwood, Pa.

truing or machining vertical shafts in position, because of corroded or worn sections. We have found it necessary to machine 24-in-diam shafts on two vertical 20,000-hp single-runner, 60-cycle units at the water-lubricated, lignum-vitae, guide-bearing section of the main shaft, where split bronze sleeves 62 in. long are attached. Water entered between the shaft and the sleeve, causing considerable corrosion and loss of metal, allowing some clearance between the shaft and the sleeve. With a locally fabricated tool-holding rig, mounted within the bearing barrel, the shaft was trued to clean metal at 13 rpm. In each case this was accomplished by driving the 60-cycle unit as a synchronous motor with a 13,500-hp 25-cycle unit as the generator, on hand control, running at approximately 38 rpm. The two generators were directly connected through temporary cables between the corresponding oil circuit breakers. To compensate for the slightly reduced shaft diameter, the contact ribs of the bronze sleeve were slightly enlarged in the bore, bronze rings fitted and bored to the reduced shaft diameter.

Inhibiting Shaft Corrosion. On previous fittings of shaft and sleeve, the former was liberally coated with thick white lead, which in time was dissolved by the entrance of water. A "No-Ox-Id" preparation was applied on the last fitting and this is expected to be more durable than white lead.

Restoration of Worn Runner Seals. The author states that the runner seals required no attention in 20 years of service, which is an excellent record. The unit surely must have been operating under very favorable conditions, such as the absence of sand, silt, or other abrasives common to rivers, during freshets and floods. We have seven vertical double-runner units (13,500 hp), and three vertical single-runner units (20,000 hp) in the Holtwood plant with from 19 to 33 years of service, in all of which the seals have become considerably enlarged. These enlargements are caused locally by operating conditions that generally do not apply to all hydrostations.

During the winter months, when frazil ice troubles occur, we are obliged to raise the intake screens which allows all trash, timbers, tree stumps, railroad ties, etc., to enter wheel pits for the duration of the ice run, which may last for several days at a time. Many of these obstructions do not pass through runners but become lodged between the buckets causing unbalancing and excessive shaft oscillations. The units frequently operate under these conditions for extended periods before unwatering and removal of obstructions, hence the seals become progressively larger yearly, owing to the runner band rubbing on the distributor ring.

Efficiency losses are a factor to be considered when a number of units are involved; accordingly this year we began reducing these losses on a 13,500-hp unit, by electrically welding a bronze seal band 1 in. wide on the inside of the lower cast-iron distributor ring, thereby restoring the original clearance of $^1/_{16}$ in.

Determining Normal Position of Runner. The inside diameter of the lower distributor of this unit was found to be worn eccentrically. To locate the normal running position of the runner, the unit was rotated as a motor and measurements taken at the seal of the varying clearances around the distributor. It was surprising to learn that the runner band rotated nearly concentric with the shaft. Owing to the great amount of work entailed in dismantling the entire unit, the job was simplified by removal of vanes, unbolting the distributor and raising it 8 in. for the welding job. After welding, the excess metal was ground off and a true circle established by means of a chart showing all dimensions from a circle drawn on the distributor plate through the fulcrum-pin holes.

The following unusual turbine troubles are cited:

Cracked Blades. Several years ago, we were confronted with a major repair problem in that cracks had developed close to the crown at the base of the fillet in two 20,000-hp (53-ft head) cast-iron runners. (Over-all dimensions: Throat diameter 14 ft, height 9 ft.) Fortunately the cracks were discovered in time to prevent disastrous results. Only a brief summary of the job will be given.

All fractures started at the outflow edges of the blades and progressed toward the inflow, varying in length from a minimum of 4 to a maximum of 32 in. and extending through the blades from the pressure to the suction side. The line of juncture at blade and crown is 57 in. long. A survey of the situation revealed that similar troubles had been experienced and corrected elsewhere by welding. After a thorough study of several methods of restoring the runner, it was decided to make repairs by electric welding without dismantling the unit. Briefly, the cast iron was chipped to a width of 3 in. on either side of the cracks, $^3/_8$ in. deep, and generously studded with $^5/_8$- and $^3/_4$-in. steel studs, carefully welded and ground to smooth contours. All upper sections of the outflow edges were chipped off. Steel inserts 16 in. long and 4 in. wide at the top, and tapering to zero at the lower end, were fitted between blades at crown and runner hub and securely welded. The hub, crown, and blades were liberally studded to make a secure bond so as to reinforce the blade and to reduce the likelihood of cracks starting at the outflow edges again.

Elaborate studies were made by means of seven strain gages fixed to the outflow edge of a blade and used in conjunction with an oscilloscope and oscillograph with the unit operating under varying loads, unloading and loading, to determine conditions which might overstress the blades. Since these analyses have involved pioneering work in the measurement field, as well as in the exploration of stresses in a blade under operating conditions, there is naturally some uncertainty in the results of the studies which remain to some extent in an inconclusive stage. Several theories have been advanced as to conditions overstressing blades, i.e., that of the transition period of the rising water column in the draft tube when changing from condenser operation to power generation, due to impact of water column striking rotating blades; another, fluttering of outflow edges of blades, owing to the long cantilever span between crown and shroud ring causing vibration fatigue of metal. Still another theory advanced is the "out-of-step" condition which occurred a number of times on these units and may have produced severe shocks and irregular rotation. It is thought possible that the thrust on the turbine could increase to such an extent as to cause failure between blades and crown.

Runners of the Francis type made of cast steel would be a marked improvement over cast iron owing to added strength, far less likelihood of failure, such as the development of fractures and broken out section of blades, and would have the added advantage of allowing welding repairs to be made readily on the pitted areas by the application of stainless steel on the vulnerable spots.

Runner-Blade Replacement. A year ago we were obliged to perform another unusual piece of mechanical surgery on a 33-year old 13,500-hp vertical double-runner turbine in which a blade had completely broken out of the lower runner between the crown fillet and the shroud ring. The over-all dimensions of the cast-iron runner are throat-ring diameter 10 ft 7 in., height 6 ft. The blade had been previously welded at what was considered a fatigue crack extending approximately one third of the way across the face of the blade slightly below the crown fillet. Fortunately, the broken-out blade was found intact directly outside the draft-tube discharge and was used as a pattern for casting a new steel blade. Forged-steel anchor plates were fitted to crown and shroud ring, and the new blade welded to these plates, making a secure job and restoring the capacity of unit to normal. In order to compensate for the added weight of the new anchor plates, counterweights were secured to the crown and band

on the opposite side to balance the runner. The fitted blade was 60 in. long, 27 in. to 32 in. wide, and weighed 580 lb.

In 1925, the writer presented a similar paper[6] outlining maintenance problems and procedure which mainly concerned early (1907) low-head types of vertical double-runner units, early welding procedure with high-carbon steel, rubber bearings, etc. The general overhauling periods of these old units have been extended by scheduled inspection periods and minor repairs from every 7 years to an average of 12 years. This was accomplished by providing better lubrication, grease retainers, fitting bronze bushings to unbushed moving mechanisms, and the substitution of stainless steel for ordinary steels to eliminate corrosion in the submerged operating mechanism. Reference was made in the present paper to the first use of rubber for bearing linings in vertical hydraulic turbines. The trial was made in the Holtwood Station in 1924, and since that date eight of the ten main, and two exciter, units have been fitted with rubber bearings; numerous similar installations have been made throughout the country. These bearings have given excellent results and long life; in fact, we have found them far superior to lignum vitae and other modern substitute materials.

In 1925, a rubber liner was fitted to a bearing housing for a 16,500-hp vertical unit and has been in continuous service to date. The only modification made was to enlarge the bore slightly to accommodate a new stainless-steel sleeve fitted on the shaft to replace a worn bronze sleeve. We have likewise found that stainless-steel sleeves on the main shaft at bearing locations are far more durable than bronze owing to its resistance to the abrasive actions of sand and silt in the river.

Vertical Thrust Bearings. It may be of interest to mention that Kingsbury thrust bearings for large vertical hydraulic units had their first trial and development in 1912 in the Holtwood Station, and since have had wide application.

E. B. Strowger.[7] The author has described an ingenious method of installing a split bushing on the shaft of a 40,000-hp Francis turbine after it had been in operation for almost 20 years. This record of performance seems to be very good, and the methods used show ingenuity in handling maintenance problems. The method now used in maintaining the shafts of the 5500-hp vertical turbines of The Niagara Falls Power Company, some of which are 40 years old, might be of interest for the record. The 11-in. nickel-steel shafts have worn in the guide bearings and to a greater extent in the stuffing boxes. No provision in the original design was made for a sleeve to take up this wear. Until recently, the shafts were repaired by truing them up at the guide-bearing section and at the packing-gland section by machining to the depth of the grooves worn by the packing and then fitting a split sleeve or band in the recess and welding it together and fastening it in place. The shaft was then finished to the proper dimensions. With this method, as repairs were made, the various shafts kept decreasing in diameter at the bearing and eventually no two shafts had exactly the same diameter at this point.

To overcome the difficulties resulting from this nonuniformity it was decided to try metallizing. In 1939, one of these shafts was built up by metal-spraying over a length of about 4 ft 4 in., which included the section at the lower packing gland, the bearing section, and the section at the upper packing gland. Fig. 22 of this discussion shows the spraying in progress. The diameter was increased from $^1/_4$ in. to about 1 in. above the worn size using stainless steel. The shaft was then machined and placed in service and is still running. An impregnated flax packing was used in the packing gland. Fig. 23 shows the completed job.

Fig. 22 Metal-Spraying Turbine Shaft

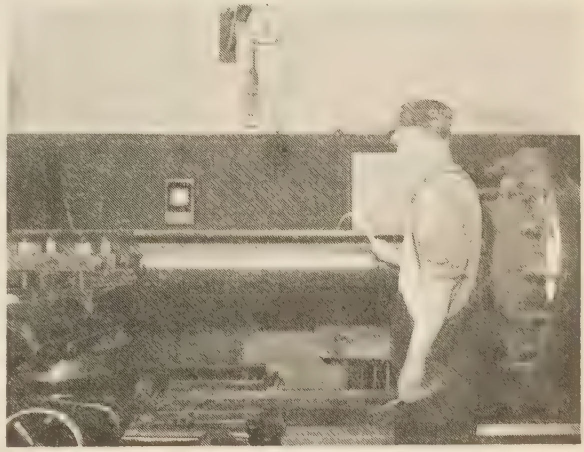

Fig. 23 Completed Shaft Built Up by Metal-Spraying Stainless Steel

On this work it was found that while the flax packing was very satisfactory, some of the original U-shaped leather packing rings could not be used because particles of stainless steel would adhere to the leather and the resulting abrasive surface of the metal would quickly wear the leather away. It is therefore necessary in such repair work to make sure that the combination of materials chosen is suitable for the job.

I. M. White.[8] The maintenance time required in connection with the author's modification to the shaft sleeve and the relief-valve body clearly indicates the desirability of taking considerable care in initial purchasing in order to hold turbine outage time to a minimum. For instance, in modern turbine installations a spare split-shaft sleeve is normally ordered at the time of original manufacture. In such cases, the spare sleeve can be completely finished in the manufacturer's shop so as to eliminate any need of special fixtures that might be required should the spare sleeve only be ordered at a later date. Such spare sleeves can be held together with either H-section keys as noted by the author or by special machine screws. Normally, a small notch is

[6] "Hydraulic Maintenance at Holtwood Plant," by T. C. Stabley, *Mechanical Engineering*, vol. 48, 1926, pp. 341–349.

[7] Hydraulic Engineer, The Niagara Falls Power Company, Niagara Falls, N. Y. Mem. A.S.M.E.

[8] Chief Engineer, The Pelton Water Wheel Company, San Francisco, Calif. Mem. A.S.M.E.

provided at the two joints to permit welding or brazing of the notch at final field assembly. After the welding, the metal can be ground and finished by hand to form a very smooth surface contour. In present-day designs, such sleeves are normally made of stainless steel in order to provide maximum wearing life.

The author has indicated that considerable work was required in the lining of the relief valve in order to overcome cavitation. This excessive repair work and consequent outage of the turbine could have been almost eliminated had the relief valve been made of cast steel at original manufacture. It is probable that the relatively small additional cost of providing initially a cast-steel body would have been offset by the outage time required for the extensive repair.

The change in head cover pressure by modification of the runner tip as discussed by the author has been found to exist on other installations. Several experiments have been conducted in order to determine the best place for venting the upper head cover through the runner. The results of these tests indicate that the author has selected a very desirable location.

Modern practice in connection with the replaceable seats of relief valves indicates a preference for the utilization of 18-8 layer weld on mild steel in place of cast bronze. Such seats cannot be satisfactorily ground as noted by the author because of the galling action between stainless-steel surfaces. However, an excellent seat can be obtained with proper cutting tools. Frequently spare seats are ordered at the same time the original unit is purchased. With such an arrangement, it is possible to install the spare seats and then to recondition the original seats in the machine shop so that they may be returned for use as spares when the need arises.

The author does not specifically mention the type of rod used in connection with the repair of the runner vanes and repair to other cast-steel surfaces. Experience indicates that an 18-8 stainless-steel rod is very satisfactory. This rod provides a surface which has proved of high quality in resistance to cavitation. Another factor that greatly influences the cavitation of the runner

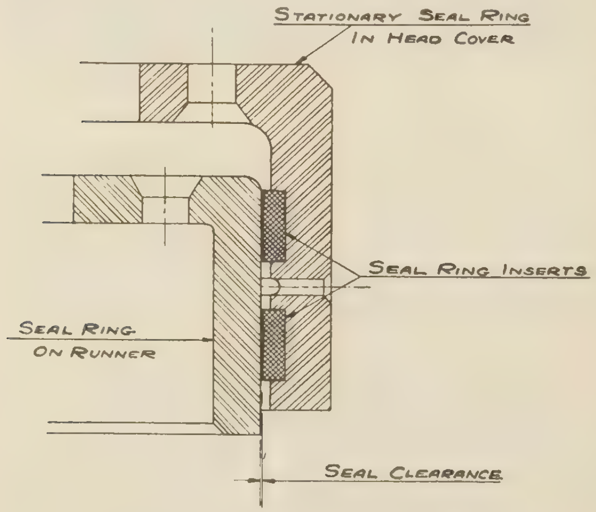

Fig. 24 Typical Seal-Ring Construction

is the actual contour of the runner surfaces upstream from the area experiencing cavitation. It has been found that a very slight high spot upstream will have a tendency to start cavitation, which accelerates as the cavitation becomes more and more severe. It is advisable to take considerable care in inspecting the area upstream of a repaired cavitated area in order to insure that no high points exist which would make the stream of water leave the surface and thereby create cavitation.

The author has stated that the seal rings on the subject turbine showed practically no wear after the 20 years of service. This is indeed unusual and indicates that very clean water undoubtedly passed through the turbine. In many cases, excessive wear of the seals does develop. Consequently, the operating companies desire to make repairs with the minimum of difficulty. Fig. 24 of this discussion indicates a scheme which has proved very satisfactory in providing new seal rings with low outage time required.

The seal-ring inserts indicated are cast in segments and then individually peened into slightly dovetailed grooves. The ends of the segments are joined together by heating with a torch. Seal-ring segments are made of Parsons white brass which has proved to be very desirable from a wearing standpoint. Close clearances can be used without danger of galling between the stationary and moving seal ring, since the Parsons white-brass material has a tendency to wipe clear should contact develop.

The seal-ring inserts can be easily ground in the field to the exact diameter required to suit the runner wearing rings. In many instances, a spare set of rings is ordered with a turbine so that it becomes a very simple matter to make a complete replacement in the field. The worn rings can be repaired at the convenience of the operating personnel.

In addition to the foregoing maintenance points, wear is sometimes experienced in the butterfly-valve seats, guide vanes, liner plates, head cover, and runner throat. In modern turbines, these parts are either made replaceable or provided with replaceable liners so that maintenance work can be accomplished with the minimum of outage time. Special wear plates are sometimes provided in the head covers so that as wear, either from erosion or cavitation, takes place, the plates can be more easily repaired or replaced than the main part itself. Replaceable plate-steel liners are normally installed in the draft tube immediately below the runner. Cavitation frequently occurs at this point and, consequently, a replaceable steel liner is of considerable advantage in maintenance.

During the past several years considerable study has been undertaken by turbine manufacturers concerning cavitation problems. These studies have definitely disclosed that runner and draft-tube cavitation is a function of turbine setting, specific speed, and barometric pressure. As a result, proper operating conditions and unit setting can be established for a new installation so that cavitation and consequent maintenance problems are reduced greatly over what has been the case in the past. Such care in initial selection of units together with proper consideration for spare parts should greatly reduce maintenance problems of present-day turbines in contrast with older turbines.

Author's Closure

The discussions presented by various central-station engineers and manufacturers' representatives embrace the practice throughout the United States and contribute much to the scope of the subject.

By keeping the problems of maintenance before designing engineers, progress of improvement in performance of equipment must ensue.

Apparently, there is a gravitation toward the usage of 18-8 stainless steel as an "antidote" to cavitation, erosion, and corrosion. We have all tried various alloys from time to time, but thus far our experience indicates that none offer any advantage over the stainless steels.

Likewise, the metal-spray process has been employed with varying degrees of success. In general, it has been satisfactory on shafts and the like, but it has not done so well on flat surfaces

or shapes upon which the sprayed metal is in the form of a patch.

Packings of various materials and forms have also engaged the attention of maintenance men. No one type, however, is a "panacea" for all conditions. Metallic packings and rawhide seem to be less severe on the shaft or trunnion than others; especially the square-plaited flax.

Mr. J. F. Roberts is probably more intimately acquainted with the units at Pit River No. 1 powerhouse than any one else, for he "lived" with them when their service life was in its "infancy," and so his contribution supplies interesting information which otherwise would have been lacking in the paper.

Supplementing his comments, relating to the runner vibration, a complete revelation of this may be found in a paper by Roy Wilkins.[9] A unique device was employed to record graphically the magnitude and frequency of the vibrations. This is described and some of the records have been reproduced.

The efficiency tests on the Pit River No. 1 units, conducted by Prof. Charles M. Allen, in 1925, have not subsequently been repeated.

[9] "Study of Irregularity of Reaction Francis Turbines," by Roy Wilkins, *Journal of the American Institute of Electrical Engineers*, vol. 42, 1923, pp. 1141–1144.

With seal-ring clearance unchanged, and runner and guide-vane surfaces restored in 1942, the turbine efficiency should be about the same as it was in 1925. It may even be slightly higher, due to the reduction of the load on the Kingsbury thrust bearing, by virtue of the reduction of pressure on the top side of the runner brought about by the vent holes drilled in the runner cone.

The guide-vane stems of the Pit turbines were not worn sufficiently to warrant repairs. This is due to the absence of grit and foreign matter in the water and also to greasing at regular intervals, performed with the vanes in motion.

In answer to Mr. Roberts' inquiry concerning the seat rings of the Pit River No. 1 turbine butterfly valves, the only attention they have had in their 20 years of service is a dressing of the roughened areas by use of a drawfile. They were then readjusted to minimize the leakage.

We have more than twenty-four turbine butterfly valves in service at various hydroplants, varying in size from 36 in. to 9 ft, and operating under heads up to a maximum of 700 ft. Thus far, it has not been necessary to replace the seat rings in any of them. It is our practice to prevent the surfaces from becoming rough or cavitated by arc welding periodically or filing them, or both.

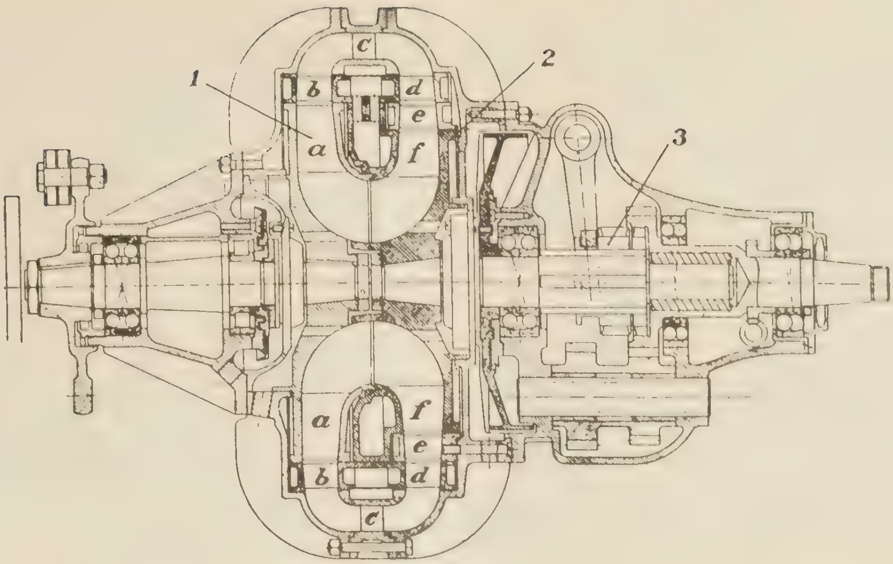

FIG. 1 LYSHOLM-SMITH CONVERTER WITH REVERSE GEAR
(a, pump; b, d, f, turbine; c, e, guide).

Development of the Lysholm-Smith Torque Converter

BY A. LYSHOLM,[1] STOCKHOLM, SWEDEN

The author's experience in the design and construction of hydraulic transmissions for automotive drives goes back over a 15-year period when the outstanding development was the Föttinger transformer, incorporating a coaxial pump impeller and turbine which was used as a speed reducer for ship propellers. Present-day designs of the Lysholm-Smith torque converter employ the Föttinger coaxial arrangement, but with improved multiple-stage turbine blading which enables the torque-speed range to become automatically variable within a wide range of high efficiency. The paper is concerned principally with the theory of the torque converter, details of early experiments in Sweden, and design features of hydraulic drives of various types as now widely applied in the automotive and railroad fields.

ABOUT fifteen years ago the author took up the problem of hydraulic transmission for automotive drives. At that time the Föttinger transformer which incorporates a coaxial pump-impeller and turbine had been used as a speed reducer for ship propellers. According to published information this converter had a very high efficiency at a certain speed ratio, but at other ratios the efficiency dropped off rather rapidly.

With the Lysholm-Smith torque converter which, like all modern hydrodynamic converters incorporates the Föttinger coaxial arrangement, it has been possible to obtain high efficiency as well as a flat efficiency curve, principally by means of improved multiple-stage turbine blading, which enables the torque-speed range to become automatically variable within a wide range of high efficiency. Fig. 1 shows the original test converter which has one pump impeller, three rotating turbine rings and two stationary guide-blade rings. An account will be given of some of the fundamental principles of the torque converter and the experimental work leading to the evolution of the present types, namely, the fixed-blade type, the adjustable-blade type, and the type with direct drive.

THEORY OF THE TORQUE CONVERTER

In a hydraulic gear of the turbine type, the following are the principal normal losses:

1 Friction loss in the pump wheel
2 Carry-over loss between pump and turbine
3 Friction loss in the turbine blading
4 Carry-over losses between the turbine stages
5 Carry-over loss between turbine and pump
6 Leakage losses
7 Rotational losses
8 Mechanical losses
9 Auxiliaries.

A summary of these losses is given in Table 1.

The friction losses in the pump wheel and turbine in the table include all losses due to the flow through the blading, together with the eddy-current losses at the inlet and outlet edges of the blades, but do not include leakage losses. When running at different torque-speed ratios, the blading losses due to varying inlet angles are the most important and account for nearly the total loss at the stalling point and racing speed. Therefore, only these losses will be considered in the following analysis, as this simpli-

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TABLE 1 BALANCE OF LOSSES

Item	First test converter	Standard bus converter	Standard rail-car converter
1 Friction, pump wheel	3.5	4.5	4.5
3 Friction turbine	2.5	3.0	3.0
6 Leakage	2.5	3.5	3.5
7 Rotation	1.5	1.5	1.5
8 Mechanical	0.5	1.5	1.0
2, (4), (5), (9) Others	0.5	1.0	1.0
Efficiency, per cent	89	85	85.5

fied assumption gives a fairly good view of the operation of a torque converter at varying speeds.

If the turbine is provided with only one runner, and if the leaving energy cannot be utilized, its efficiency at different speeds of the turbine shaft and with a constant supply of energy will, according to the principles of hydrodynamics, vary in accordance with a parabola. Conditions will be entirely different, however, if the leaving energy from each wheel is fully utilized in the next wheel. In the subsequent calculation, it is shown how the turbine "frictional" efficiency will then vary. Strictly speaking, the calculation is only valid for an axial-flow turbine and an infinite number of stages.

If c is the velocity of discharge from a blade and w designates the inlet velocity, then the useful pressure drop in the blade will be

$$\Delta h_u = \frac{c^2 - w^2}{2g} \qquad [1]$$

The pressure drop on account of the friction can be expressed by the following formula

$$\Delta h_f = \frac{f}{2} \cdot \frac{c^2 + w^2}{2g} \qquad [2]$$

where f is the friction coefficient, which also includes eddy losess.

Thus the total pressure drop in the blade is

$$\Delta h = \Delta h_u + \Delta h_f = \frac{c^2 - w^2}{2g} + \frac{c^2 + w^2}{2g} \cdot \frac{f}{2} \qquad [3]$$

The efficiency of each blade will evidently be obtained from the relation between Δh_u and Δh, or

$$\eta = \frac{\Delta h_u}{\Delta h} \qquad [4]$$

The values of Δh_u and Δh are obtained from the foregoing equations.

In the description to be given, the pump will be considered as an energy producer, and the efficiency of the turbine at different speeds will be examined, assuming a constant energy supply, that is, a constant product of pressure drop and quantity to be produced by the pump and absorbed by the turbine. Thus the pressure drop in the turbine, multiplied by the circulating-fluid quantity Q, must be constant, or

$$Q\Delta h = \text{const}$$

Assuming for the moment that Q is constant, then if a triangle for c, w, and u is drawn, and the peripheral speed u is assumed to vary with constant flow, the value of w will be seen to decrease from c, when the turbine is stationary, to a minimum value when the direction of w is perpendicular to u (in the following called the optimum point or the optimum speed), and then to increase to c again at racing speed. From Equations [1], [2], and [3], it will be seen that the value of Δh_u will vary between zero and a maximum value, the value of Δh_f between $f/2 \cdot (c^2/g)$ and a minimum value, and the total pressure drop Δh between $f/2 \cdot (c^2/g)$ and a maximum value. Thus at maximum torque at the stalling point, and zero torque at racing speed, all the available energy is absorbed by friction.

To get a constant product of $Q \times \Delta h$, however, the flow of liquid must be increased at all speeds in relation to the flow at the optimum point. At increased flow, the pressure drop varies with the second power of the flow, and thus the energy consumption must vary with the third power of the velocity of flow. The increased flow at a pressure drop Δh in relation to the flow at the optimum point will thus be obtained from the following relation

$$\sqrt[3]{\frac{\Delta h_{\max}}{\Delta h}}$$

where $\Delta h_{\max}$ denotes the pressure drop at the optimum point.

Hence it is clear that the velocity-ratio triangles must be enlarged in the same degree as shown in Fig. 2, where the triangles at starting and racing, as well as at a few intermediate speeds, are given. The triangles are drawn for different "relative speeds," that is, the speed of the turbine in relation to the speed of the turbine at the optimum point.

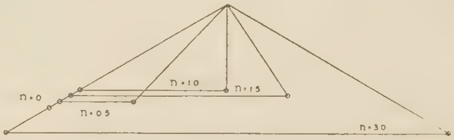


FIG. 2 VELOCITY TRIANGLES WITH $Q\Sigma\Delta h$ CONSTANT
(n = relative speed.)

It may be argued that the number of stages is of no importance for the efficiency of the converter, as the leaving loss always is utilized as inlet energy of the pump but that is not correct, since a certain percentage of the leaving velocity is always lost. Furthermore, it is possible with a multiple-stage converter to get a suitable stalling torque without using too large pump diameter and too much backward slinging of the pump blades which results in efficiency loss over the whole speed range.

Experience with converters, having similar types of blading, indicates that with turbines having one to three stages, the maximum efficiency may be about the same, but that with an increased number of stages, the stalling ratio as well as the over-all performance is improved. The three-stage converter is thus a good combination for most purposes.

When laying out the blading of a converter, the usual pump and turbine equations should be used; thus

$$H_p = \frac{y_p}{g}(U_2C_{2u} - U_1C_{1u})$$

$$H_t = \frac{1}{y_t g}\sum(U_1C_{1u} - U_2C_{2u})$$

where

H_p = pump head
H_t = turbine head
y_p = pump efficiency
y_t = turbine efficiency
C_{1u} = peripheral component of absolute inlet velocity
U_1 = peripheral velocity of inlet
C_{2u} = peripheral component of absolute outlet velocity
U_2 = peripheral velocity of outlet

Obviously the head of the pump and turbine must be equal and, further, the exhaust energy of the pump as well as of the last tur-

bine must be equal to the inlet energy of the turbine and pump, respectively, excepting of course carry-over losses.

Naturally due consideration must also be taken of the other losses mentioned in Table 1.

The foregoing equations are preferably solved by a cut-and-try method.

Early Experiments in Sweden

In order to investigate the blade friction at different speeds, some preliminary tests were made with a reaction balance, Fig. 3, by means of which the friction factors at different inlet angles could be studied. The results for two blade-section investigations (Nos. 3 and 2*b*) are given in Fig. 4. Blade section No. 3, Fig. 5, is a reaction blade of ordinary design and section 2*b* is especially suitable for varying inlet angles. However, the last-mentioned type of blade proved to be even better at ideal inlet angles of flow. In Fig. 4, a curve is drawn, calculated from later tests made on the complete hydraulic converter with blades principally of section No. 2*b*.

As will be seen, the results obtained with the converter followed closely the test values with the reaction balance for inlet angles between 50 and 145 deg. At angles smaller than 50 deg, the real values of f are, however, higher than those obtained with the reaction balance, and it is thought that this is due to the fact that the number of blade channels in the reaction balance was too small. However, this test gave sufficiently good results to enable construction of the blade system of the complete gear. It is interesting to note that the blade friction closely corresponds to the friction loss in a smooth tube, as determined by "Blasius" tests.

The blade system in the original Swedish test gear, Fig. 1, consisted of a pump wheel and three rotating blade rings, as well as two fixed guide-blade rings in the stator. Performance tests were made by measuring input torque and stator reaction, as well as primary and secondary speeds.

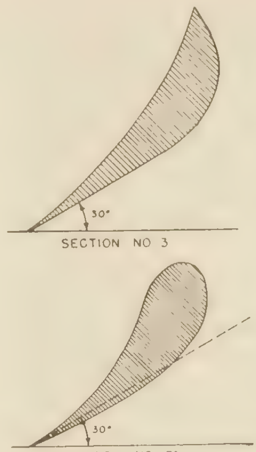


Fig. 5 Blade Sections

Fig. 6 shows the efficiency curve as a function of the turbine speed on the assumption that 50 hp is supplied on the primary side. This efficiency curve is very flat and attains a maximum value of 89 per cent. The corresponding turbine-torque curve is also shown. This curve has a hyberbolic shape which differs considerably from the straight line corresponding to a parabolic efficiency curve.

During the tests, the velocity of the circulating flow of liquid was also measured with a Pitot tube at 800 and 1000 rpm of the primary shaft. From these measurements it appeared that, at a constant primary input, the flow of liquid at stalling speed was 22 per cent, and at racing, 50 per cent higher than at the speed corresponding to the optimum point.

Fig. 7 shows the test results in another form. Here the efficiency curves are drawn for various constant primary speeds from 500 to 1600 rpm as a function of the secondary speed. The maximum efficiency for 500 primary rpm is 83 per cent, and at 1400 rpm the efficiency has increased to nearly 89 per cent.

Converter Input Characteristics. A normal three-stage converter with short third turbine blades absorbs more power at

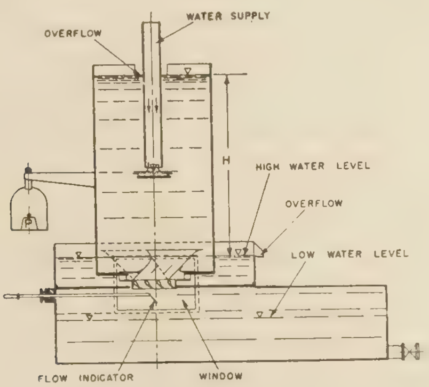
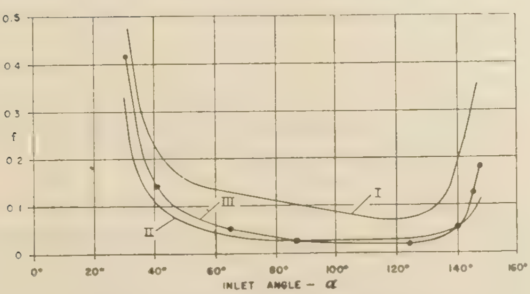


Fig. 3 Reaction Balance



I Profile No. 3 measured by reaction balance
II Profile No. 2b measured with reaction balance
III Profile No. 2b brake test with hydraulic gearing

Fig. 4 Test Values for Friction Factors

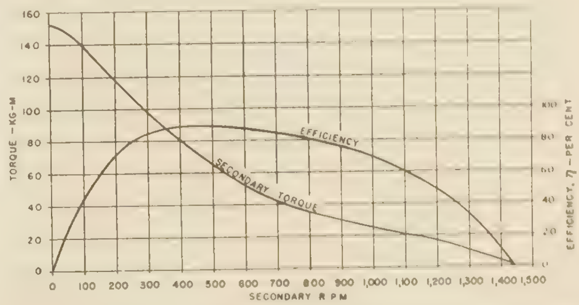
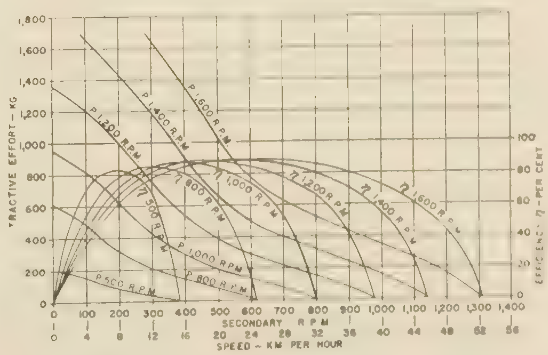


Fig. 6 Torque and Efficiency Curves at Constant Input of 50 Hp; Original Converter



50 H.P. OMNIBUS; AXLE RATIO $\frac{1}{4}$ – 1; EFFECTIVE WHEEL DIAMETER, 880 MM.

Fig. 7 Tractive Effort and Efficiency Curves; Original Converter

stalling through racing, and thus the engine speed at stalling decreased to about 70 per cent of racing speed. This is very important for automotive drives as it prevents overspeeding of the engine at stalling and flattens out the efficiency curve.

However, an increased stalling torque of about 15 to 20 per cent can be obtained by using long blades in the last turbine stage, reducing the ratio mentioned to 80 per cent. This is very useful for most industrial applications where engine speeds are generally lower. A further increase in stalling torque of 15 per cent is obtainable by using adjustable pump blades, which also permits control of the input power.

Range of Torque Converter. In order to obtain the correct performance from a torque converter, it must be used with an engine of suitable torque-speed characteristic, in the same way that a ship's propeller must be designed to suit the engine characteristic. Certain variations from the standard characteristic can, however, be made without material alteration of design, or efficiency. The easiest way to alter the torque-speed characteristic is as follows:

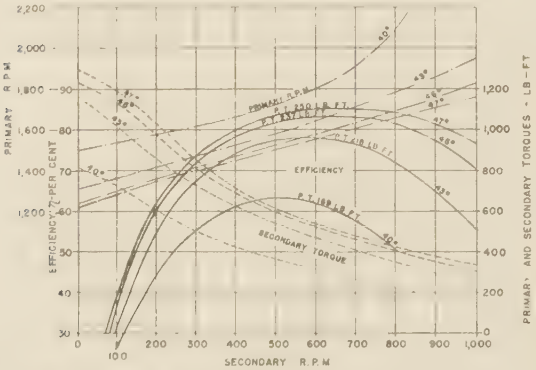
1 To reduce the external diameter of the pump wheel, if it is desired to use an engine of lower torque.

2 To increase the effective outlet angle of the pump blades by increasing the number of blades or the blade angle for an engine of higher torque.

3 To use third turbine blades, which increases the input power considerably at high speed ratios.

4 To incorporate a primary step-up or reduction gearing.

Some experiments showing the effect of altering the blade angle are illustrated in Fig. 8. The efficiency curves are shown



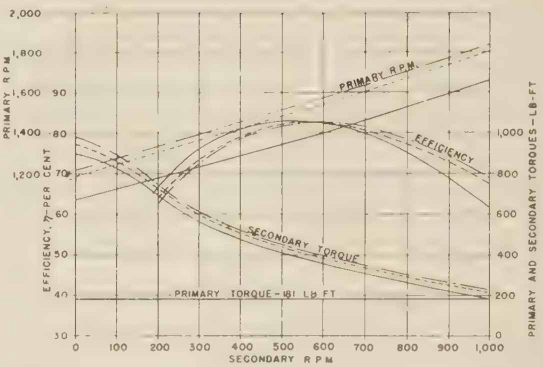
Fitted with pump wheel having 14 blades
Blade angles as given on curves
Curves at constant primary torque

FIG. 8 EFFICIENCY TESTS ON HYDRAULIC TORQUE CONVERTER

for constant primary torques, and the decreased range of the converter and lower efficiency as the angle is decreased will be noted. The available primary-speed range is also shown on the curves for each angle. The curves show the very great importance of having a correct pump-blade angle.

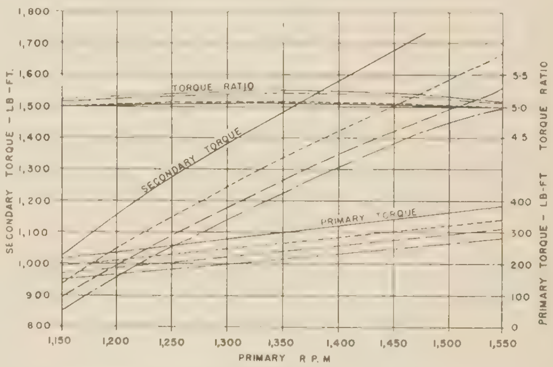
The effect of reducing the pump wheel diameter is shown in the curves in Figs. 9 and 10. It will be noted that, as the diameter is reduced, the efficiency at the lower end of the curve is reduced, and is increased at the higher end. Reducing the diameter displaces the curves along the secondary speed axis owing to the increased input. Fig. 11 shows the alteration of the stalling characteristic produced by reduction of the pump diameter.

From these results a suitable pump wheel diameter to suit any particular engine characteristic or conditions can be determined. The efficiency curves in Fig. 9 are shown for only one constant primary torque and only represent one condition. It will be



Standard pump wheel
Standard pump wheel, diameter reduced 7/16 inch
Standard pump wheel, diameter reduced 9/16 inch

FIG. 9 SPEED-EFFICIENCY CURVES FOR VARIOUS DIAMETERS OF PUMP WHEEL



Standard pump wheel
Standard pump wheel, diameter reduced 1/4 inch
Standard pump wheel, diameter reduced 7/16 inch
Standard pump wheel, diameter reduced 9/16 inch

FIG. 10 STALLING TESTS ON PUMP WHEELS OF VARIOUS DIAMETERS

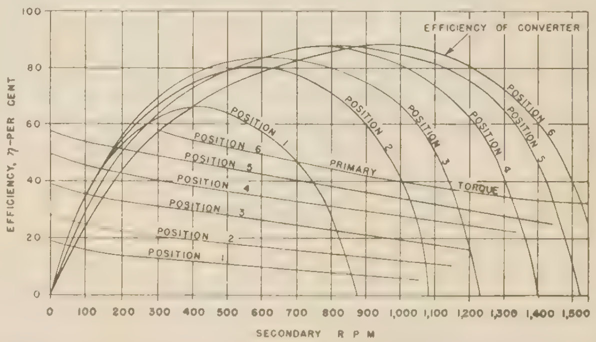


FIG. 11 EFFICIENCY AND PRIMARY TORQUE CURVES FOR HYDRAULIC TORQUE CONVERTER WITH ADJUSTABLE BLADES
(Constant primary speed, 1700 rpm.)

noted in Fig. 10, that the torque ratio at stall increases with decrease of wheel diameter, in about inverse proportion to the diameter.

Torque Converter With Adjustable Pump Blades. In Fig. 11, characteristic curves are given for a torque converter having adjustable pump blades. The numbers of the curves indicate different positions of the adjustable pump blades. It will be noted that the efficiency is less sensitive for different blade posi-

tions than in the case of the pump blades corresponding to Fig. 8. The advantages of the adjustable blades are:

1 Practically no secondary torque with closed blades.
2 Higher stalling torque ratio, up to 6:1.
3 Flatter efficiency curve.
4 Possibility of overloading the engine when desired.
5 Possibility of avoiding critical speeds of the engine crankshaft.

Effect of Turbine Variations. It is a very simple matter to alter a pump wheel on a production converter, as it may only have from 14 to 20 blades, but it is not so easy to make turbine-blade alterations, because of the large number of blades. However, as already mentioned, two different types of third-stage blades are used, a short blade and a long blade. It may be stated generally, however, that the calculations for the turbine are made for a speed which corresponds approximately to a flow perpendicular to the inlet of a blade ring. This "calculation" speed corresponds to the optimum speed previously mentioned. For the tested converters, the speed corresponding to the maximum efficiency is generally higher than the optimum speed owing to the type of blading used.

Maximum Efficiency Speed. The speed at the maximum efficiency point will depend upon (*a*) the number of stages, (*b*) the diameter of the blade rings, (*c*) the head to be utilized, and (*d*) other factors. Obviously, the optimum speed is increased by (*a*) a smaller number of stages, (*b*) a smaller diameter of stages, and (*c*) a larger head. This may also be expressed as a ratio which corresponds to the "Parsons figure" or "quality figure," as used for steam-turbine calculations, namely, $\Sigma u^2/h$, where Σu^2 is the sum of the squares of the turbine-blade velocities, and h is the head utilized in the turbine. For modern steam turbines, this ratio is generally about 2000 (metric units) for impulse turbines having some reaction, and up to 3500 for pure reaction turbines of the Ljungstrom double-rotation type, for instance. To get a direct comparison with these figures it is necessary to multiply the corresponding values for torque converters by the metric mechanical equivalent of heat, 427. The converters now made have a Parsons figure of 3200 to 3500 at "calculation" speed, and up to 50,000 at racing.

It is also possible to use the "specific speed" of the pump and turbine as a basis of comparison. For torque converters, the specific speed n_s will preferably be expressed by the following formula

$$n_s = 3.65\,\frac{n\sqrt{Q}}{h^{3/4}}$$

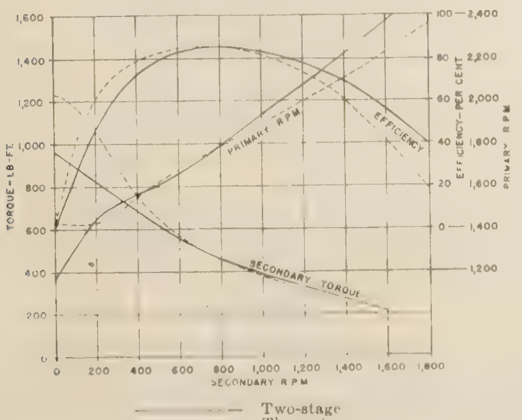


Fig. 12 Comparison of Two- and Three-Stage Turbines at Constant Primary Torque of 245 Lb-Ft

where n is the revolutions per minute of pump or turbine, Q is the quantity in cubic meters per second, and h is the total head in meters produced by the pump.

The average value of n_s for the pump in the present converters varies between 190 to 210, and for the turbine between 150 to 165 at "calculation" speed.

With a given diameter of converter, the number of stages can be reduced from three to two, but it is not easy to alter the diameter of the stages materially without altering the converter diameter.

Some of the results of changing from a three-stage to a two-stage converter in the same casing are shown in Fig. 12. From Fig. 12, it will be seen that the efficiency curve of the two-stage converter at constant torque is lower for the first 800 rpm of the secondary and, above this speed, is higher. There is also a considerable alteration in the stall ratio, that of the two-stage converter being constant at 4 to 1, and the three-stage varying from 4.5 to 1 to 5 to 1. These figures refer only to the two particular converters compared in the test. Higher stall ratios are available with different blade combinations. The effect of this on the secondary torque curve and engine speed at the commencement of converter operation is very marked. Both converters have the same maximum efficiency of 85 per cent.

Design of Torque Converter

In the design of an efficient and mechanically satisfactory torque converter, the following considerations are important:

1 It is necessary to prevent the short-circuiting of the fluid between the revolving parts and the casing. This is done by a number of labyrinth seals as shown in Fig. 13.

2 Seals must be provided to prevent loss of fluid. These seals are normally of the axial-sealing type generally using a steel ring against a carbon ring for sealing the surfaces having difference in rotational speed. One of these rings has axial movement by means of a copper bellows or a synthetic-rubber diaphragm.

3 Evacuation of gas from the converter must be provided for. New fluid under certain conditions of circulation and temperature is liable to gasify to a slight degree. To insure the removal of this gas, it is drawn through a small pipe from the top of the converter.

4 It is very necessary, in order to prevent cavitation, that the converter be kept full of fluid under pressure. The automatic replacement of fresh fluid is insured by an injector or mechanically driven pump which draws it from a reserve tank as required and feeds it under pressure to the converter.

5 To provide for a temperature rise in the working fluid which may result from prolonged use of the converter, a small cooler is placed in the fluid circulation. This cooler circuit also includes a filter.

Converter Blading. The major component parts of a modern converter are shown in Fig. 13.

Direct-Drive Type. This type was first developed in Sweden by the Ljungstrom Company and in England by Leyland Motors. By using a double-acting clutch, converter drive or direct drive may be obtained at will. In the latter case the pump impeller is cut out by the clutch and the turbine by means of a free wheel and thus all hydraulic losses are eliminated.

A later variety of this type, fitted with an angle drive, is shown in Fig. 14. This unit was built by the Spicer Company for General Motors Truck & Coach Company's buses. The direct-drive type is also very suitable for rail cars and rail-car trains.

The horsepower of the direct-drive converter varies generally between 100 and 300 hp. Many thousands of buses and several hundreds of rail cars equipped with this converter type are in service in Europe, Africa, Australia, and the Americas.

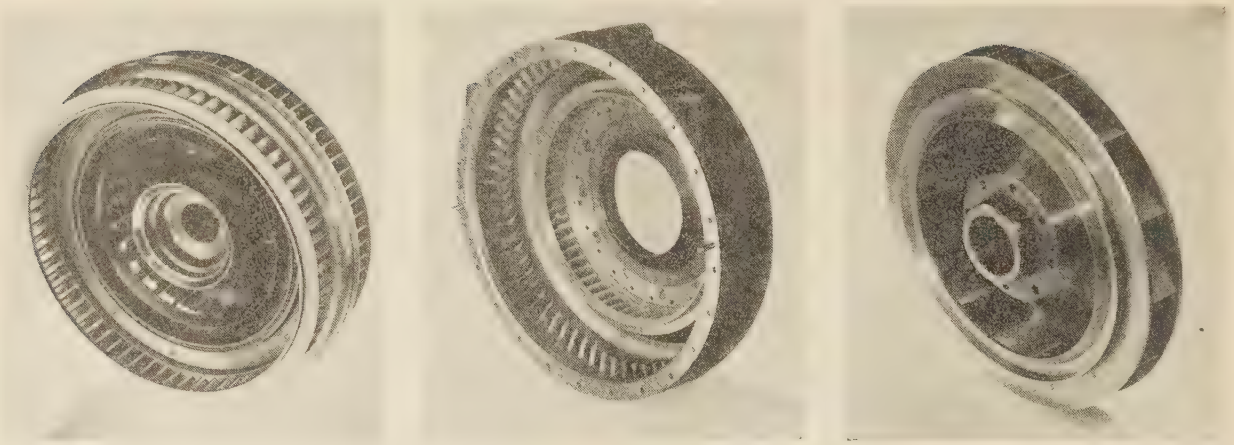


Fig. 13 Major Component Parts of Modern Converter

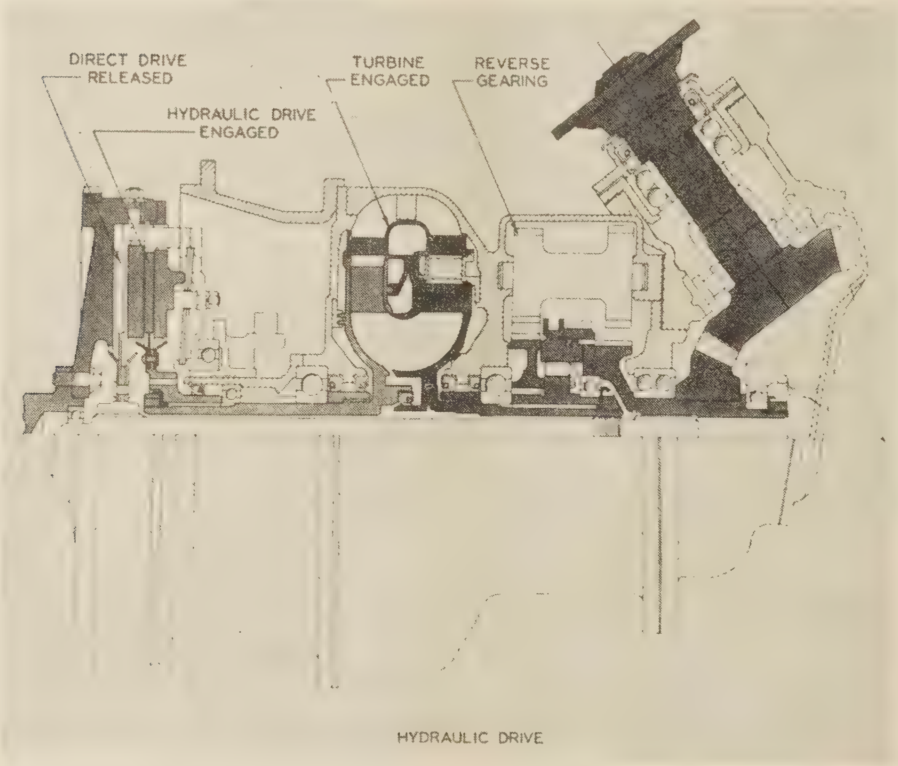


Fig. 14 Direct-Drive Type of Hydraulic Transmission, Fitted With Angle Gearing for Bus Operation

Fixed-Blade Type. This type has been used extensively by the Twin Disc Clutch Company for tractors, for oil-drilling rigs, for logging, for locomotives, and for construction and locomotive cranes. The horsepower generally ranges between 100 and 300 hp.

Adjustable-Blade Type. Fig. 15 shows an installation of a twin-disk adjustable-blade converter for oil-field application.

The largest converter of this type so far has been built for a 1200-hp Diesel-driven locomotive for the Bergen-Oslo Railway in Norway.

New Developments. It is regrettable that, owing to war restrictions, the latest available test data and improvements cannot be included in this paper. It may be said, however, that it seems possible to raise the maximum efficiency as well as widen the range of high efficiency operation considerably.

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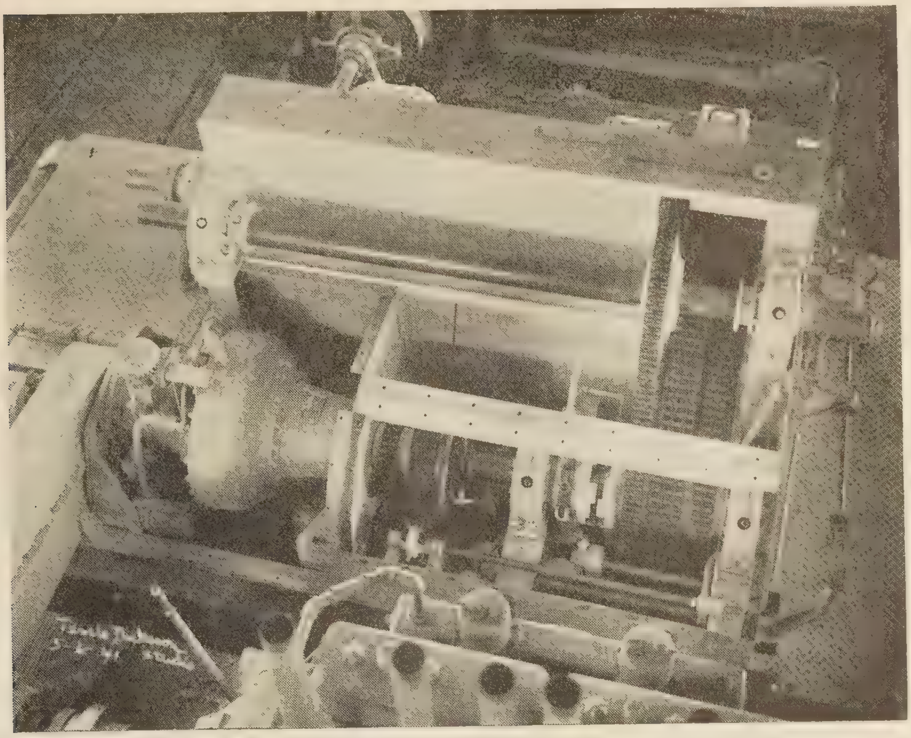

Fig. 15 Shows Installation of Twin-Disk Adjustable-Blade Converter for Oil-Field Application

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Discussion

R. Eksergian.[2] The desirable characteristic of any torque converter is a high-efficiency curve through a long speed range. The early Föttinger transformer, when modified to a variable-speed torque converter, had roughly a parabolic efficiency curve, with a decreasing linear output torque against increase of secondary speed. Through his excellent researches, using multistages on the turbine end together with blunt blade sections, the author has greatly improved the efficiency over the entire speed ranges, resulting in a more nearly hyperbolic secondary torque.

It is known in turbine theory that the multistage reaction turbine tends toward increased flow rates at reduced speeds, resulting in increased torque amplification over that of the impulse type with only a single or a few stages. The decreased pressure drop per stage on the turbine end, for a given total head, also results in reduced losses due to the decreased velocities, so that multireaction stages tend toward improved efficiency.

The torque-speed characteristic of a turbine with constant flow rate is approximately linear, decreasing with speed. This implies a parabolic output curve against speed. With improved efficiency, particularly in the speed ranges either above or below the optimum speed at maximum efficiency, the flow rate itself increases, resulting in increased starting torque and an improved efficiency throughout the entire speed range. The effective head is the difference between the impeller head, corresponding to the mechanical input, and the counter head of the turbine, corresponding to the output power. The effective head is balanced by the friction head, corresponding to the shock and friction losses in the circuit. As the secondary speed either decreases below or increases above the rated speed, the turbine-output counter head decreases much more rapidly than the impeller head since the latter operates at more nearly constant speed, and at constant input. At optimum speed, the losses at best are small, and then the impeller head more nearly balances the counter turbine head. At other than optimum speed, particularly at the more extreme over- and under-speeds, if the unbalance effective head is considerable, so that the friction head is small, the flow rate would in-

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crease rapidly. Actually, the shock losses increase very rapidly at both lower and overspeeds, thus increasing the friction head and checking the growth of circulation to a much smaller value.

Now the torque on the turbine at any given speed depends upon the change in angular momentum of the fluid per unit time at the several reaction stages and thus directly on the flow rate. Increasing the flow rate therefore increases the torque and output at any given speed. On the other hand, the increase of flow rate results in an increase in torque on the pump impeller, increasing roughly as a quadratic function of the flow rate. But the impeller torque also increases linearly with the speed. For this reason, to maintain constant engine torque and to permit increased flow rate in the starting zone requires a lower pump or engine speed, that is, the primary speed must increase with secondary speed at constant primary torque, as shown in the author's Fig. 8. In some cases it appears that it is not always possible to absorb the full engine output at all turbine-shaft speeds.

In a previous paper, the writer established a compatibility equation $F(Q, \omega_p, \omega_s) = 0$, where Q is the flow rate, ω_p and ω_s are the primary and secondary speeds. This equation was established by expressing the primary torque as a function of the flow rate and primary speed, and the secondary torque as a function of the flow rate and both secondary and primary speeds. On combining with the energy equation of the entire circulating flow and expressing the losses in terms of the flow rate and speeds, the foregoing function was obtained.

Provided the losses can be estimated accurately through proper interpolation of test data, the characteristics of the performance of a torque converter can be arrived at for any given design. For this reason, the author's experimental studies, while interesting for over-all characteristics, would further prove very valuable if the data could be presented in a more detailed form for estimating component losses.

For instance, it would be desirable to express such losses between blade groups in terms of blade angles and relative speeds with different flow rates. It would also be helpful in the present paper to give actual dimensional data of the various proportions of the torque converter in connection with the curves presented.

G. F. Wislicenus.[3] It is a recognized fact that, while there have been various modifications of the original form of the Föttinger transformer, the Lysholm-Smith torque converter represents a significant advance in this field. In view of this situation, the present paper constitutes a valuable contribution to the technical literature on the subject of hydrodynamic transmissions and, as such, might have been presented more fully. It is to be regretted that the paper was not worked out in sufficient detail to offer the reader an adequate introduction into the theory and fundamental characteristics of this type of machinery. There seem to be some omissions in the text which make it difficult to follow the derivations. In order to substantiate this statement, the writer will point out but a few items which he considers to be either incompletely or incorrectly presented.

The difficulties of determining the losses in various parts of hydrodynamic machines are generally recognized. For this reason, it would be most instructive if the method whose results are given in Table 1 could be explained in some detail. It also seems desirable to explain more fully what is meant by "carry-over losses" between the various parts of the machine.

With reference to the following derivations, it is hoped that the author can supplement the paper by a list of notations without which these equations cannot be understood properly. While Equation [1], for instance, is easy to understand with respect to stationary-vane systems, the same equation does not appear to be correct with respect to the moving-vane systems which are in the present case responsible for the greatest part of the head drop through the secondary part of the machine. If, for instance, c and w are interpreted as designating the "relative" velocity, it is seen that what is called the "useful pressure drop" in Equation [1] is actually only the change in "static" pressure in the vane system, always adhering to the author's assumption that this system is considered as one with axial-through-flow direction. The total useful pressure drop in such a vane system, however, differs appreciably from the static pressure drop. In instances under the assumption of a relative entrance angle of 90 deg, measured against the peripheral direction of the system and zero absolute rotation of the fluid at the discharge side, the static head drop without losses is $\frac{u^2}{2g}$ which corresponds to Equation [1], while the total head drop is $\frac{u^2}{g}$, as could be calculated for instance from the equation for H_t, given below Fig. 2 of the paper. It is seen that in this case the total head drop differs from the static head drop by a factor of 2, so that Equation [1] can in general not be used for the total head change in the systems considered. The subsequent derivations, however, seem to indicate that c and w are absolute velocities. Also under the latter assumption, Equation [1] cannot be used for moving-vane systems. For the example just given, the actual head drop through the system would again be twice as great as that given by Equation [1].

Equation [2] for the friction head does not appear justified without further explanations. This equation gives one the impression that the friction losses are calculated as the sum of two friction losses, one being proportional to the velocity head at the entrance and the other to the velocity head at the discharge of the system. If this is true, it is not easy to understand why the same friction factor is applied in both cases. Furthermore, the friction factor must certainly be expected to change over wide limits as the flow conditions vary as a function of the secondary speed, so that the corresponding variations in the total pressure drop cannot even be approximated under the assumption that f is a constant.

From a theoretical point of view, the result that the rate of flow through the transformer reaches a minimum at the point of best efficiency appears to this writer highly improbable, chiefly because of the variations of the primary torque at constant primary speed, as described by the test results given. At constant primary speed, the primary torque appears to fall off as the secondary speed is increased. This fact, in connection with a primary or pump runner of the type used, seems to indicate a reduction in pump capacity. Even if the primary speed is then increased to obtain constant power input (as assumed by the author), the capacity cannot regain its original value, because the primary runner operates at another point of its head-capacity curve than that of best efficiency. In view of this apparent contradiction, it would be highly desirable if the author would give additional information on the method used in determining the rate of flow through the converter by Pitot tube. These test results, if substantiated, would seem to be of far-reaching significance with respect to the theory of this type of machinery.

As indicated in the beginning, the writer believes that most of the foregoing difficulties could be cleared up if the author would have the opportunity to explain his theoretical approach more fully than was apparently possible in this paper. In view of the great interest which exists throughout the industry in this kind of machinery, it is believed that a fuller presentation of the theory would be fully justified.

[Because of wartime conditions, we have been unable to obtain a closure from the author, Mr. Lysholm.—Editor.]

[3] Research Engineer, Worthington Pump & Machinery Corp., Harrison, N. J. Mem. A.S.M.E.

Gas Turbines and Turbosuperchargers

By SANFORD A. MOSS,[1] PH.D., LL.D., WEST LYNN, MASS.

Since his university days, the author has been associated with many of the developments relating to the gas turbine and particularly to the perfection of the airplane-engine turbosupercharger. It is from this intimate and interested viewpoint that he outlines the history of the gas turbine and its potential use as a prime mover. A comprehensive Bibliography of the art of gas-turbine design has been included in the paper to support the author's purpose of outlining the evolution of each component of the modern gas turbine from its earliest inception. United States and foreign gas-turbine research projects are referred to, and then details of advancements in such items as nozzles, compressors, and the like, are discussed. While the gas turbine as a prime mover has not been applied except in isolated cases, such as the Neuchâtel, Switzerland, 4000-kw emergency gas-turbine plant and the Swiss Federal Railway gas-turbine locomotive, partial gas-turbine cycles are in somewhat extensive commercial operation. Examples of these are the Diesel exhaust-gas turbine, the Diesel boiler cycle, the divided-manifold cycle, all of which are mentioned in some detail. The final part of the paper is devoted to consideration of the extremely vital development of the constant-pressure aviation turbosupercharger, which has been brought to its highest state of development by the author's company, in which work the author has taken a leading role, from its inception in the last war down to its present state of extensive application.

PRODUCTION of power by direct action of products of combustion on a turbine wheel has been an engineering dream for a century and a half. A number of approaches to the complete scheme now are in successful commercial operation, and it seems possible that these are paving the way for the long-sought accomplishment of the primary plan of the gas turbine as an efficient prime mover. So it would now seem opportune to give some history and reminiscence (without any thermodynamics), list the partial accomplishments now in commercial use, and speculate on the future. The author can do this from personal association with many of the items ever since he became, in 1895, one of the many who have had dreams in the matter.

This association with gas-turbine developments has occurred during an engineering career which began in San Francisco in 1889, with a machinist apprenticeship, following with work as draftsman, in the shop of Mr. Edward A. Rix (an early master draftsman in compressed-air engineering); and continuing as a draftsman in various gas-engine shops. This was before the days of electric transmission and there were several plans of compressed-air power and transmission in successful operation. One was use by Mr. Rix of the early impulse water wheel of Pelton to drive an air compressor, with a transmission pipe line some miles long, and then a reciprocating hoisting engine using the transmitted air after it had been reheated by coal or coke so as to increase its volume (1).[2] So this really was a combustion heat-engine cycle. Its thermodynamics, which is that of the "Brayton constant-pressure cycle," is an early item of the present subject. Mr. Rix and his associates also had a street railway in New York City operating on compressed air stored in steel cylinders. But the author remembers how these and similar schemes soon were replaced by the early electric transmissions.

Gas-Turbine Proposals

Beginning about 1889 both the steam turbine and the internal-combustion reciprocating engine began the competition which finally resulted in the elimination of the reciprocating steam engine as a prime mover, except for the steam locomotive, and some ship installations.

Ever since the beginning of this period it keeps occurring independently to students of thermodynamics that the steam turbine and internal-combustion engine might be combined, and so they "invent" the gas turbine. The usual form of this, shown in Fig. 1, has a chamber into which is forced compressed air and fuel, usually oil, so as to give combustion under pressure. The products pass through a nozzle so as to drive a turbine wheel, part of whose power drives the combustion-chamber compressor, and the remainder (if any) is the net power derived from the fuel burned.

The author was one of the many of these "inventors," while a student in the classes in thermodynamics and hydrodynamics of Prof. Frederick G. Hesse at the University of California in 1895. Since then, the author's engineering career, now drawing to a close, always has been directed toward the gas turbine. He also has tried to keep posted on the numerous advances in other fields which could help the gas turbine, as well as on the many gas-turbine publications, some of which are given in the references.

Like most of the other inventors, the author at first thought he was alone in the gas-turbine field. In 1900 an account was written as a Master's thesis (2) and filed in the University of California Library, which gives early proposals of some details.

Associated with Mr. Rix and the compressed-air street railway in New York City was a Chicago firm, "The Vimotum Company," engaged in promoting operation of railway cars by internal combustion. The author participated during 1898-1899 in the actual operation on the "Big 4" railway tracks between Columbus, Ind., and Indianapolis, Ind., and later on the Pennsylvania Railroad tracks between Chester and Darby, Pa., of full-sized railway passenger cars, self-propelled by gasoline engines, using the "variable-speed transmission" of the Reeves Pulley Company of Columbus, Ind. This probably was the predecessor of all modern internal-combustion locomotives. Plans were formulated in 1898 for replacing the gasoline engine by a gas turbine as was set forth in the Master's thesis (2) cited. This is the earliest proposal that the author knows of for a gas-turbine

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Contributed by the Aviation Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses are references contained in the Bibliography at the end of the paper. An attempt is made in the figures herein and the references to give the first published proposal of each item. Of course, this cannot be done with certainty, but at any rate the figures and references give early if not the original proposals. The author would appreciate references to earlier proposals of any items.

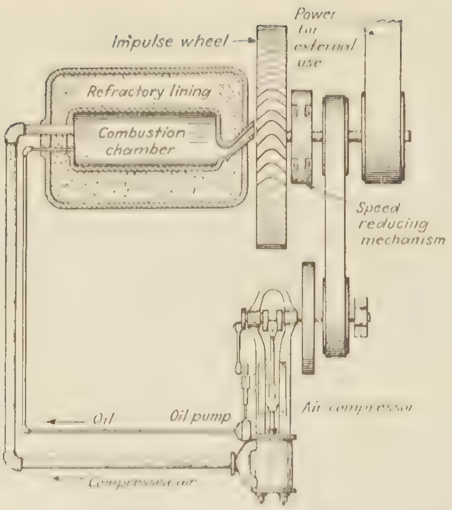

Fig. 1 Diagrammatic Plan of Fundamental Gas Turbine
(Reproduced from the Doctor's Thesis, Ref. 9, of 1903. This diagram does not include "Heat Interchanger," of the Theses, Refs. 2, 9.)

locomotive. But the proposition never came to a head, which was just as well, because then it never could have been made operative.

During the planning for this proposed gas-turbine locomotive, patent searches were made, and unexpectedly there was found a British patent of November 30, 1791, to John Barber (3), giving probably for the first time the fundamental gas-turbine plan. There was a combustion chamber into which fuel and air were introduced, with a nozzle from which the products of combustion issued onto a turbine wheel. This supplied power to drive the compressor, as well as external power.

An early diagram of Barber's cycle is given in Fig. 1, from the Thesis (9). Many gas turbines proposed since differ in plan only by the addition of a heat interchanger for transferring the heat of the exhaust gases to the compressed air proceeding to the combustion chamber. An early mention of this is in the Doctor's Thesis (9) and an experimental model of 1905 is shown in Fig. 4. It is not probable that the apparatus shown in Barber's British patent ever was operated.

Davey (8) lists some partial approaches to the gas turbine and gives a list of British patents beginning in 1856. The earliest complete United States gas-turbine patent of which the author has personal knowledge is to Charles G. Curtis, June 24, 1895 (4). Mr. Curtis had extensive connections with regard to steam turbines, with the General Electric Company. The author has some knowledge of these, but none of any actual work on gas turbines by Mr. Curtis.

At this same time was a United States Patent to Leon LePontois (5) filed December 5, 1895, on an explosion combustion chamber for a gas turbine. This was the subject of a Doctor's Thesis at Cornell University by Walter O. Amsler in 1897 (6). Actual tests were made of LePontois' explosion apparatus, as well as a theoretical study, with unfavorable conclusions.

United States Gas-Turbine Research

The author always has had in mind continuous combustion, and after the gas-turbine-locomotive scheme, proposed in 1898, came to naught, he started gas-turbine research in 1901 in the Sibley College Laboratory, Cornell University. Laboratory time for a year was required to get a continuous-combustion chamber in stable operation. It frequently went out and then the oil on the red-hot firebrick lining filled the neighborhood with dense black smoke, so the Sibley College people well knew of the gas-turbine research.

The first De Laval steam turbine in the United States, Fig. 2, which had been exhibited at the World's Fair in Chicago in 1893, was used for the De Laval nozzle-patent demonstration mentioned

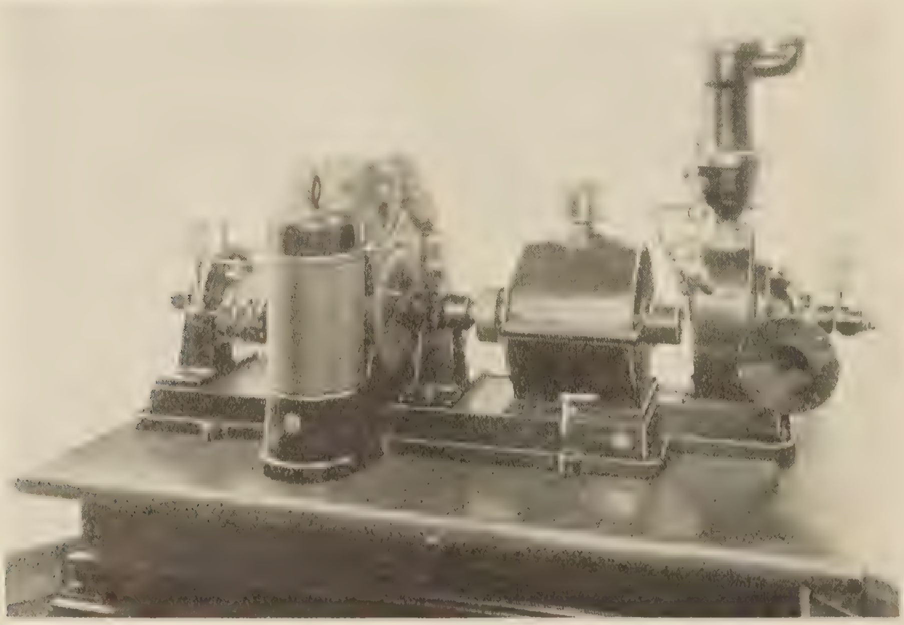

Fig. 2 First De Laval Steam Turbine in the United States, 1893
(Exhibited in the Chicago World's Fair and then sent to Sibley College, Cornell University. The wheel of this turbine was used in 1902 in the gas-turbine research of the Doctor's Thesis, Ref. 9, and was the first turbine operated on products of combustion in the United States and possibly the first anywhere. Reproduced from De Laval pamphlet of Ref. 10.)

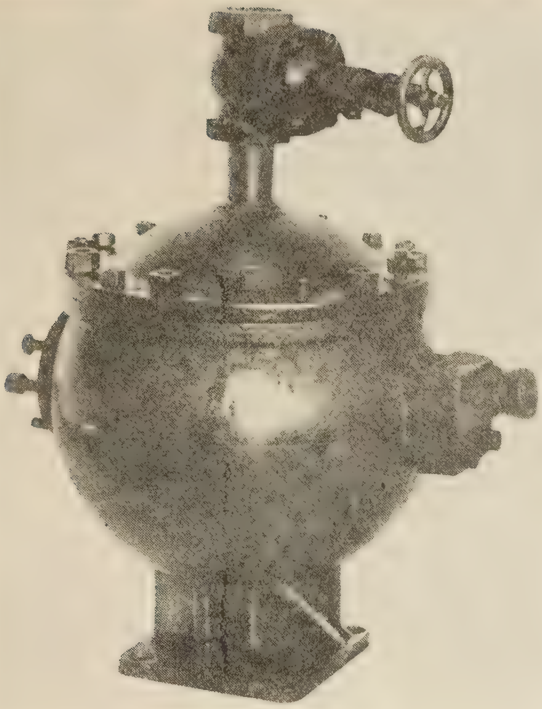

Fig. 3 Combustion Chamber of General Electric Gas-Turbine Research, 1903

(This was made spherical to give minimum radiating surface. This research started in Schenectady in 1903 and was transferred to Factory K, General Electric Company, West Lynn Works, in 1904.)

later (10), and then went to Cornell. So the bucket wheel of this turbine was borrowed and operated with the combustion chamber, about the end of 1902. This probably was the first turbine wheel actually operated by products of combustion in the United States, and possibly was the first such turbine wheel ever operated. The entire power was absorbed by a prony brake, and the air for compression was furnished by a steam compressor, with computation of the power required.

But, as with many other experimental gas turbines, the power for compression was more than the turbine power. So except for the historical fact that the combustion chamber actually operated the turbine wheel, the experiment was a flat failure. A Doctor's Thesis (9) was prepared and presented to Charles P. Steinmetz and Ernest J. Berg of the General Electric Company. The author previously had been a draftsman on steam-turbine work with General Electric. In June, 1903, the connection was resumed and has continued ever since, with work on various items more or less connected with gas turbines.

A number of General Electric models of gas turbines successively were gotten into operation, first at Schenectady, and beginning in 1904 at Lynn, with consultation with Prof. Elihu Thomson and Richard H. Rice. There were combustion chamber, heat interchanger, nozzle, and single-stage impulse turbine wheel; but with power for compression from an independent air compressor and allowed for by computations. Some U. S. patents were obtained during this research ($9^1/_2$). Figs. 3, 4, and 5, respectively, show the combustion chamber, heat interchanger, and nozzle jet of 1903 and 1905. The results of this research will be given later in the paper. The patent drawings ($9^1/_2$) show fairly well the actual apparatus that was used.

Foreign Gas-Turbine Research

A gas-turbine patent was applied for by Charles Lemale (8) about 1901 and he began association with Réné Armengaud in 1903 in actual operation in France of a continuous-combustion gas turbine (8, 21). This probably was the earliest elaborate gas-turbine experimental work. From then on, there have been many proposals and tests of various sorts of gas turbines.

Numerous publications have been made, a few of which are given herewith, i.e., references, (4, 5, 6, 7, 8, 12, 13, 21, 22, 23, 24, 25, 26, 27). Others are in the Engineering Index Annual and in references (7, 8). Some of these operate with a hydraulic column; operate below atmospheric pressure; have cooling by water injection; have hollow turbine buckets with cooling; have cooling by air excess; use pulverized coal; use reheating combustion; or have cooled compressors. But most of these have not led to any conclusive results. The cycle now receiving the most attention is that of Fig. 1 with the addition of the heat interchanger of the previously mentioned theses (2, 9).

Nozzles for Steam and Gas Turbines

Beginning about 1885 there were many arguments concerning the power that could be obtained by expansion in a nozzle of a jet

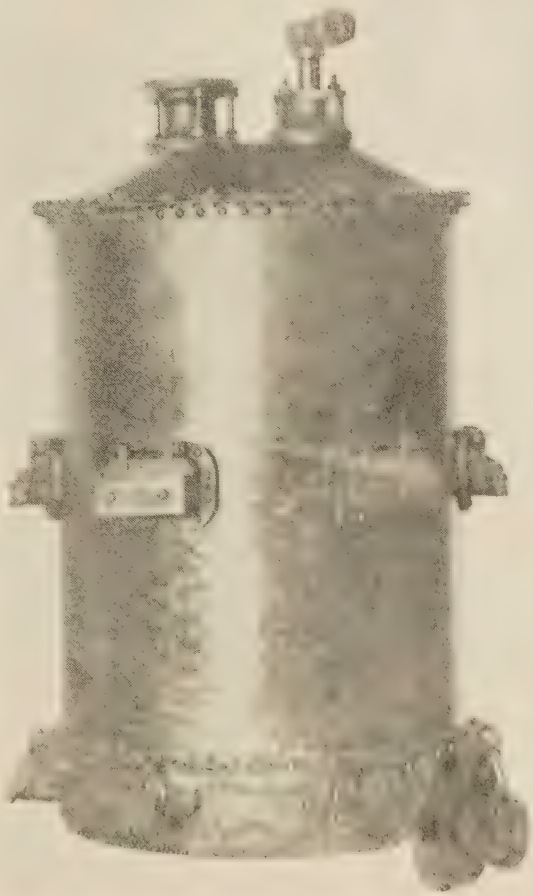

Fig. 4 Heat Interchanger of General Electric Gas-Turbine Research, 1905

(The air from the compressor passed through some tubular coils on its way to the combustion chamber. The turbine exhaust gases circulated outside of these coils.)

FIG. 5 NOZZLE JET OF GENERAL ELECTRIC GAS-TURBINE RESEARCH, 1905

(Curious white-hot nodes appeared in this jet of products of combustion, being of one type when the mouth of the convergent-divergent nozzle was too small for the pressure ratio; of another type when it was too large, and vanishing between when it was about the right size. Similar nodes later have been observed with steam nozzles.)

directed on a turbine wheel. Napier had shown experimentally that when steam was discharged from an opening in a vessel into a region of lower pressure, the flow increased as the pressure in the outer region decreased, until the outer pressure became about one half of the pressure in the vessel. For lower values of the outer pressure there was no increase in flow. Therefore, it was argued that full theoretical velocity of a steam-nozzle jet could not be obtained when the pressure ratio was less than 0.5. It seemed reasonable to assume that the theoretical power from a steam-turbine wheel was the same as from a reciprocating steam engine, for pressure ratios between 1.0 and 0.5. But Napier's law meant that for pressure ratios less than 0.5, a steam turbine would not theoretically give the same power as a steam engine.

Carl Gustaf Patrick De Laval was building single-stage steam turbines in Sweden with steam pressures of 115 psia or so, exhausting into atmospheric pressure of about 15 psi. He said he was avoiding the disability of Napier's law by using a nozzle first convergent and then divergent, popularly called an "expanding nozzle" because of the final divergent portion. De Laval tried to get a United States patent on this shape, and the official file shows that Napier's law was cited against him, with the support of scientific experts, including Gustave Zeuner, the famous German engineer. Finally, De Laval with his 1893 Chicago World's Fair turbine of Fig. 2 gave an actual demonstration that he was right (10).

The theoretical reasons for a nozzle first convergent and then divergent already had been given by Osborne Reynolds (11), with the first diagram of such a nozzle, reproduced as Fig. 6. But this was buried in a philosophical magazine and does not seem to have been appreciated by engineers. At a much later date it began to be understood that the theoretical available energy for a steam turbine with proper nozzles is the same as for the corresponding reciprocating engine. But there was a period within the memory of the author when engineering ignorance in this matter held back steam-turbine development.

Next it was argued (12, 13) that, while proper power could be developed by use of a steam nozzle, it was because of proximity of the fluid to condensation. But, so it was said, circumstances differed with gases far removed from condensation, such as products of combustion, so that a gas-turbine nozzle never could be efficient. Experimental measurements were published (12) of temperatures measured along a nozzle producing a jet of a nearly perfect gas, showing that the temperature did not fall with lowering of pressure as was predicted by the formulas for isentropic expansion. It was pointed out that with an expanding gas, mechanical energy only could be developed by conversion of internal energy, as manifested by temperature drop. Thus it was argued that proper conversion of pressure drop into velocity could not be accomplished by a gas-turbine nozzle.

To give further support to this contention, a test was published (13) in which a De Laval single-stage turbine was operated at full speed with compressed air. To avoid complication, no power was abstracted from the turbine shaft. There was measurement of temperature in the pipe preceding the nozzle, and in the exhaust pipe. The pressures in the inlet pipe varied in different tests from about 12 to 123 psi, with atmospheric pressure in the exhaust pipe. But in spite of these pressure drops, which corresponded to appreciable drops of isentropic temperature, the temperatures in the exhaust pipe always were nearly the same as in the inlet pipe. Since compressed air within the pressure limits used is nearly a perfect gas, the equality of temperature showed that in spite of passage through the nozzle no internal energy had been abstracted corresponding to an isentropic drop of temperature. Thus a nearly perfect gas such as air or products of combustion would not liberate its energy by isentropic drop in a turbine nozzle, and so a gas-turbine could not be efficient.

This experiment was published in all seriousness as a demonstration of the point made. But if a student of engineering has not by now detected the fallacy, he has fallen into a trap which often before has been set by the author for the unwary thermodynamist. As was mentioned with planned casualness, no energy was abstracted from the turbine shaft, so that all of the kinetic energy of the jet, which no doubt was nearly the theoretical isentropic value, was converted into heat by turbine-bucket and rotation-loss friction. With the nearly perfect gas used, this restored the original temperature, as could be expected.

The temperature-measurement experiment (12) was equally fallacious. The jet flowing along the nozzle no doubt had the full isentropic velocity and low temperature at each point corresponding to the pressure drop. But the temperature-measuring apparatus destroyed this velocity, converting it into heat, and so, with a perfect gas, restoring the initial temperature.

The theory of this, possibly for the first time, was published in

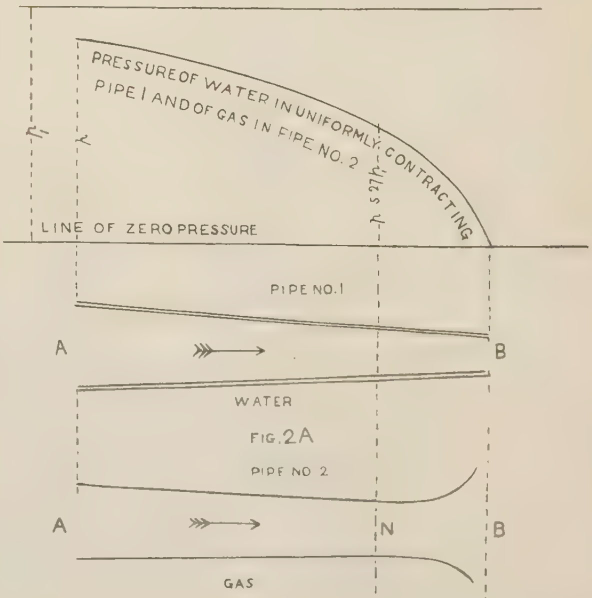

FIG. 6 FIRST PROPOSAL FOR A NOZZLE FOR ELASTIC FLUID TURBINES, WHICH WAS FIRST CONVERGENT AND THEN DIVERGENT

(Reproduction of Fig. 2A of the 1885 paper of Osborne Reynolds, Ref. 11.)

1916 (14). This paper describes an experiment made by the author about 1905. Steam, initially slightly wet, and at an appreciable pressure, was discharged into the atmosphere through a properly shaped nozzle. As is well known, the initially wet steam became wetter as it expanded to atmospheric pressure, and so the jet leaving the nozzle had the "static" temperature of wet steam at atmospheric pressure, which is 212 F. But a thermometer inserted in the nozzle jet showed a much higher temperature, the result of conversion of some of the jet velocity into heat. Complete conversion would give a temperature called "impact temperature" in the 1916 reference (14), but now beginning to be called "total temperature." The situation nowadays is fairly well understood ($14^1/_2$), and it seems to be remembered that it was explained in the 1916 publication (14) of the 1905 experiment.

But although the fallacies of the two publications mentioned (12, 13) are very evident now, and cannot be cited against the ultimate success of the gas turbine, and were partially refuted at the time (12, 13), they were disturbing factors during the General Electric gas-turbine research and had to be argued against.

Compressors for Gas Turbines

A part of a gas-turbine plant is an air compressor to force air into the combustion chamber, which must be driven by part of the power from the turbine wheel. In the fall of 1903, the General Electric gas-turbine research began to include development of a "centrifugal compressor," with rotative speed suitable for direct connection to the gas-turbine wheel. High-speed wheels previously had been proposed for air compression (15, 16), with some sort of "reversed nozzle," since called a "diffuser," for converting into pressure the absolute velocity of air leaving the wheel periphery. Such diffusers had begun to be used for incompressible fluids. They probably first had been used in 1895 in a centrifugal pump of the famous English scientist Osborne Reynolds (17), which is shown in Fig. 7.

But De Laval's plan of a turbine nozzle, first convergent and then divergent, for a compressible fluid, now was firmly fixed. With this was the idea that a "reversed nozzle" for converting velocity into pressure with a compressible fluid such as air always should be the reverse of the De Laval expanding nozzle. Therefore the foregoing compressor references (15, 16) show such passages beyond the high-speed wheels, with the minimum section or throat of De Laval, and speak of "throat conversions of velocity into pressure." The famous French engineer August Rateau was working on high-speed centrifugal air compressors about this time (18), but the author never has been able to find out if he then used diffusers, and if so what his early ideas were about their shape. Rateau supplied centrifugal compressors for the Armengaud-Lamale gas-turbine work (7, 8, 18, 21).

In a patent applied for in 1904 (19), it was set forth correctly, probably for the first time, that with velocities below the velocity of sound, in the case of a diffuser for a compressible fluid, the velocity changes are the major items, so that passage areas change inversely as velocities, just as with an incompressible fluid. But with velocities greater than the velocity of sound, the change of volume with a compressible fluid, due to change of pressure, is the major item, so that increase of velocity in a nozzle requires increase of area, and decrease of velocity in a diffuser requires decrease of area. So with a centrifugal air compressor with impeller exit velocity less than the velocity of sound, the diffuser must be divergent, just as with a centrifugal pump. But for velocities greater than the velocity of sound, the theoretical diffuser first is convergent, has a throat, and then is divergent. All of this now is elementary knowledge, but in 1904 it was not evident.

The velocities of the General Electric centrifugal compressor

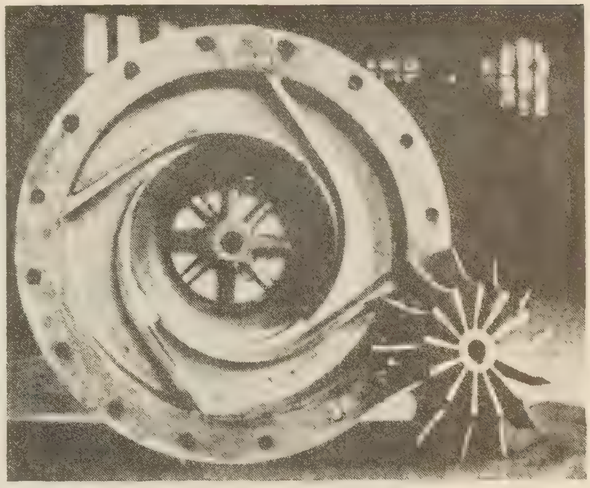

Fig. 7 First (or Very Early) Fluid Pump With Impeller Having Axial-Flow Inlet and Radial Vanes at Exit, and Diffuser for Converting Into Pressure the Impeller Exit Absolute Velocity

(Mather-Reynolds pump of 1895 from Ref. 17, p. 112.)

for the proposed gas turbine always were such as to require theoretically a divergent diffuser, and development work on this basis was begun in 1904 under the supervision of Richard H. Rice, with the collaboration of the author and other General Electric engineers. When the primary gas-turbine research ceased in 1907, the centrifugal compressor had developed so that it became a commercial product of itself. One early result was the use of centrifugal compressors for blast furnaces for production of pig iron (20). This met early opposition from advocates of reciprocating "blowing tubs," and it was alleged that a centrifugal compressor, operated by a flow-metering device to deliver a constant weight flow of air, nevertheless was subject to a process called "churning" whereby the impellers would rotate without delivering any air, in spite of the indication of the flow meters. This was shown to be a fallacy. Also it was shown (20) that even though internal-combustion engines operated by blast-furnace gas might have somewhat better thermal efficiency, the small fixed and maintenance charges of the centrifugal compressor gave lower total cost of blast-furnace operation.

Rateau did some early centrifugal-compressor work already mentioned, a great deal of it for gas turbines (7, 8, 18, 21). Some work probably also was done elsewhere in Europe. The Ingersoll-Rand Company has continued the development in the United States. As one result, centrifugal compressors, started in the United States as a gas-turbine by-product, now are almost exclusively used throughout the world for blast-furnace blowing, as well as for many other purposes.

As another part of the General Electric development, geared centrifugal superchargers for United States aviation engines began about 1925 and now are in enormous production (28), using impellers based on the original Osborne Reynolds design (17), Fig. 7. But in spite of the fact that some of this centrifugal-compressor development was inspired by the gas turbine, there is present evidence that the centrifugal compressor may be exceeded in efficiency by a comparatively recent competitor, the "axial-flow blower." If so, this blower will be the one used in the ultimate gas turbine. These blowers are used in the Neuchatel gas-turbine power plant (34) and Swiss gas-turbine locomotive (35). But on the other hand they have a shorter operating range and many more stages than the centrifugal compressor, and progress is being made in improvement in effi-

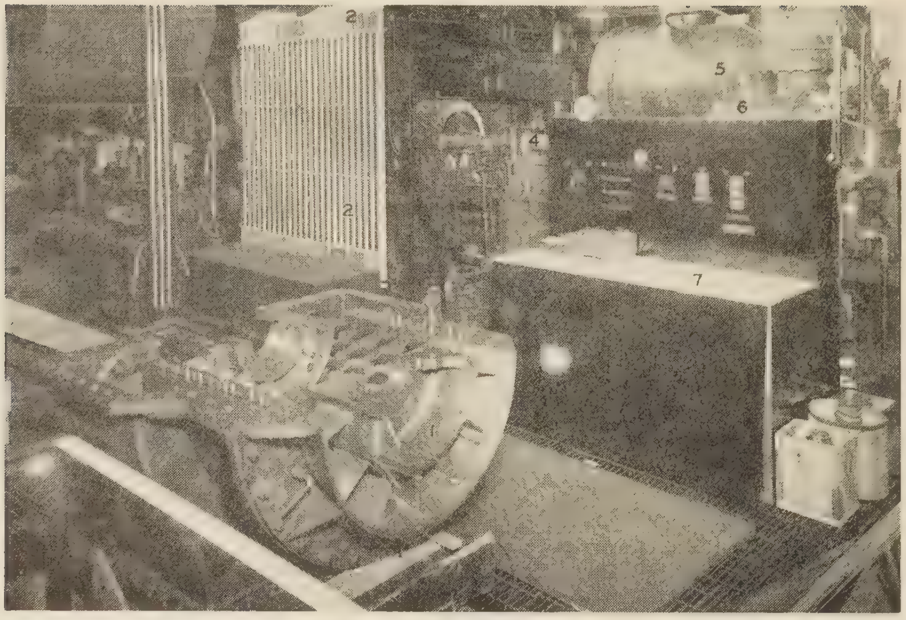

Fig. 8 Gas-Turbine Laboratory Now in Operation in the United States

ciency of the latter. However, this is a matter for the future and cannot be settled here. At any rate, the gas turbine had its part in introducing centrifugal compressors for blast furnaces, many industrial purposes, and present geared and turbosuperchargers.

United States Gas-Turbine Research Continued

The General Electric gas-turbine work, previously mentioned as beginning in June, 1903, continued until about 1907. At first the power for compression consumed all of the turbine power. Some approximate figures will show why. Suppose a gas-turbine wheel theoretically delivered 100 hp, and suppose the compressor theoretically required 49 hp. Then there would be theoretically 51 net hp. But the turbine wheel might have an efficiency of 70 per cent so that it actually would deliver 70 hp; and the compressor might have an efficiency of 70 per cent, so the net horsepower would come out $100 \times 0.70 - 49/0.70$. Before the reader is cruel enough to laugh at this result, let him put himself in the place of lots of us gas-turbine inventors who have sweated blood through years of research only to come out this way or worse. The reason is that with the 51 theoretical hp net, we have apparatus of 149 theoretical hp, with consequent losses, instead of the losses directly corresponding to 51 hp. A reciprocating internal-combustion engine deducts the power for compression from the gross power, before any power is delivered, and so the losses are based upon the net power output rather than upon the sum of powers of compression and expansion.

This paper purports to give items other than the vast number of gas-turbine thermodynamic computations which have been published by many persons, the author among them. But a simple approximate computation can be made which will show the other side of the foregoing picture of a gas turbine with a zero net power. This is on the basis that the products of combustion act as a perfect gas and have mass and properties of the same nature as those of the original air for combustion. Suppose the initial pressure for the compressor and the exhaust pressure of the turbine is 14.7 psia and suppose the final pressure of the compressor and the nozzle inlet pressure of the turbine is 49 psig or 63.7 psia. With such equality of initial and final pressures for turbine and compressor, the ratio of the theoretical power from the turbine to the theoretical power required for the compressor is equal to the ratio of the absolute temperature at nozzle inlet to the theoretical absolute temperature at the end of isentropic air compression.

A test of the Neuchatel gas turbine is cited later (34), and its figures should be before every gas-turbine student. The ratio of the theoretical powers in test No. 2 is 1.83 and the ratio of the absolute temperatures is 1.82 which shows the character of the approximation here being made. With 70 F inlet temperature the compressor isentropic final temperature is 802 F abs. Suppose the temperature at the turbine nozzle inlet is 1946 F. This is a high temperature, but within the bounds of possibility. The absolute temperature at the nozzle inlet then is 2406 F. The ratio of absolute temperatures, which is approximately the ratio of the theoretical powers, then is 3, that is, the theoretical power for the compressor is 33 per cent of the theoretical turbine power.

Using the efficiencies of the Neuchatel gas-turbine test (34), the compressor efficiency is about 85 per cent, so the actual power for the compressor is 0.33/0.85 or 39 per cent of the theoretical turbine power. With 88 per cent efficiency of the turbine, the actual power from the turbine is 88 per cent of the turbine theoretical power. That is to say, the net power is $(0.88 - 0.39)/0.88$ or 55 per cent of the turbine actual power. This figure is far beyond the 1907 possibility and is on the basis of all of the progress since that date, and then some. The Neuchatel gas-turbine test has a turbine temperature of about 1000 F, much lower than the 1946 F previously assumed, so that the net power only is about 27 per cent of the actual turbine power. But while in 1907 we hoped for everything that since has been shown by the Neuchatel test, the turbosupercharger, and all the other progress, the actual reality was far different.

So the General Electric gas-turbine research continued until a fuel consumption was in sight, of 4 lb of kerosene per net hphr. At that time good oil engines were using 1 lb of oil per net hphr. Even though it seemed certain that the clouds cast upon the operation of a gas-turbine nozzle (12, 13) had no basis in fact, it was thought desirable to measure actual spouting velocity of products of combustion. So the combustion chamber was flexibly mounted and nozzle reaction measured.

The work was difficult to conduct with accuracy, but enough was done to indicate that the only corrections needed were to take account of departures from the perfect gas laws of the products of combustion. Similar nozzle-reaction experiments later were conducted by Mr. Glenn B. Warren (30) with similar results. Recently, Dr. Chester W. Smith and other General Electric engineers have worked on tables of available energy of products of combustion for turbosuperchargers, with exact values of specific heat. But probably more work needs to be done to get exact available energies for gas turbines.

All early General Electric gas-turbine work was conducted with temperatures possible with materials then known, and no vision was had of the possibilities which the turbosupercharger work to be described later has developed. So the temperatures were kept low by air excess or water injection, both resulting in inefficiencies. No way then seemed open to do better, and so the gas-turbine part of the research was postponed, but with vigorous commercial work on centrifugal compressors, as already mentioned.

Among the many persons who long have had gas-turbine ideas are two with whom the author has had the pleasure of personal acquaintance, and who have made some publication, Dr. Harvey N. Davis (29), president of Stevens Institute of Technology, and Mr. Glenn B. Warren (30), designing engineer of the Steam Turbine Division of the General Electric Company.

Explosive and Continuous Combustion

The gas turbine had been expected to replace the reciprocating internal-combustion engine, and so there was an early idea that it was essential gas-turbine combustion should be similar. This led to a number of proposals for explosion gas turbines, as in the case of LePontois (5, 6) already cited. A German engineer, Hans Holzwarth, has made extensive publications on such apparatus through many years. But there seems doubt as to whether he ever reported performance of a complete machine which compressed its own air (27).

Opposed to these explosion gas turbines are those based on continuous combustion under pressure. The possibility of efficient operation once was doubted. Since then many continuous combustion chambers have been operated with complete success, so demonstration on this point no longer is needed.

An original, or at least an early continuous-combustion chamber, was that of the Doctor's Thesis (9). Such combustion in a chamber at a pressure of any amount above atmospheric proceeds exactly as it does in the open combustion chamber of a steam boiler, where the absolute pressure is about 14.7 psi; and the same heat of combustion is liberated. Because of constancy of conditions, the nozzle-spouting velocity is constant, which is a necessary condition for efficient operation of a turbine. On the other hand an explosion combustion chamber delivers a jet with variable velocity, which only can be proper for the turbine speed during a fraction of the time and so results in inefficiency. At any rate modern gas turbines are wholly on the continuous-combustion basis, and explosion outfits are not now attracting any attention.

Gas-Turbine Prime Movers

Thus the gas-turbine idea has been kept alive through many years, with some partial approaches, as will be discussed later, but always with the hope among enthusiasts that someday it will compete with such high-efficiency prime movers as the modern steam central station or the internal-combustion reciprocating engine. Prof. Lionel S. Marks, now professor emeritus of the Graduate School of Engineering of Harvard University, long has been a gas-turbine student (31), and on December 14, 1939, he gave a lecture (32) which presented a realistic view of immediate gas-turbine prime-mover probabilities. He pointed out that the successful use of the temperatures necessary to give high efficiency might be far off, but that there were many immediate prospects with lower temperatures. Among these were locomotives to compete with the modern steam locomotives of moderate efficiencies, plants where water is scarce, and stand-by plants where low first costs would give total charges competing with more efficient but expensive units. Similar ideas were expressed by Dr. Adolphe Meyer of Brown, Boveri & Company (33), in a paper which mentioned the starting in the Brown-Boveri shops of the Neuchatel 4000-kw gas turbine (34) and the Swiss Federal Railway gas-turbine locomotive (35), whose completion is described next.

The town of Neuchâtel, Switzerland, arranged for an emergency power plant, in a bombproof place, as a war protection, and a 4000-kw gas-turbine plant was constructed. An efficiency test of this in actual operation was published by the famous Swiss turbine authority, then in his old age, Dr. A. Stodola, and seems to be a worth-while operation of a gas-turbine prime mover (34).

Recently actual test results have been published of the gas-turbine locomotive mentioned by Dr. Meyer as being constructed by Brown-Boveri for the Swiss Federal Railway (35), which seems ready for actual use. This is another operation of the sort of gas-turbine prime mover which has been sought through many years. Both of these gas-turbine prime movers are of the sort that Professor Marks and Dr. Meyer mention, where useful purposes are served by temperatures of about 1000 F. But like the other cycles here mentioned, they pave the way for such higher temperatures as will make high gas-turbine efficiency a primary object. Admiral Mills recently has mentioned development by the United States Navy of gas turbines for ship propulsion ($35^1/_2$). Fig. 8 shows part of the laboratory in which gas-turbine research now is being conducted in the United States.

Partial Gas-Turbine Cycles

There are some gas turbines in extensive commercial operation, but not as complete prime movers. An early one is the gas turbine in extensive use by Brown, Boveri & Company and others, to supply compressed air to the supercharged combustion chambers of the steam boilers for the Brown Boveri Velox System (36) and other similar systems.

Many partial gas-turbine cycles have been in successful operation for years, for oil refineries, built by the Allis-Chalmers Company (37), which has some connection with Brown-Boveri. Outfits of a similar sort have been constructed or planned by the General Electric Company (38) and others. Fig. 9 shows an example.

All of the foregoing outfits are planned for the comparatively moderate temperature of about 1000 F. They use gases produced in the course of chemical and combustion operations, and drive air compressors. They do not have the obligation imposed on a gas-turbine prime mover, which has to produce net power as well as to compress the air which produces by combustion or otherwise the fluid which drives the turbine. But they are definite steps in gas-turbine progress.

In the meantime, temperatures used in steam turbines have been rising, and many are operating at temperatures around 900 to 1000 F and show present general use of such temperatures (39). But to a "red-hot" enthusiast who lathers himself into a white heat in trying to get white-hot gas turbines, a mere 900 F is almost a refrigerator temperature. But it is a step on the way.

Reciprocating Engines Exhausting Into Gas Turbines

A gas-turbine enthusiast dreams of complete prime movers such as the Neuchatel power plant (34) and the gas-turbine

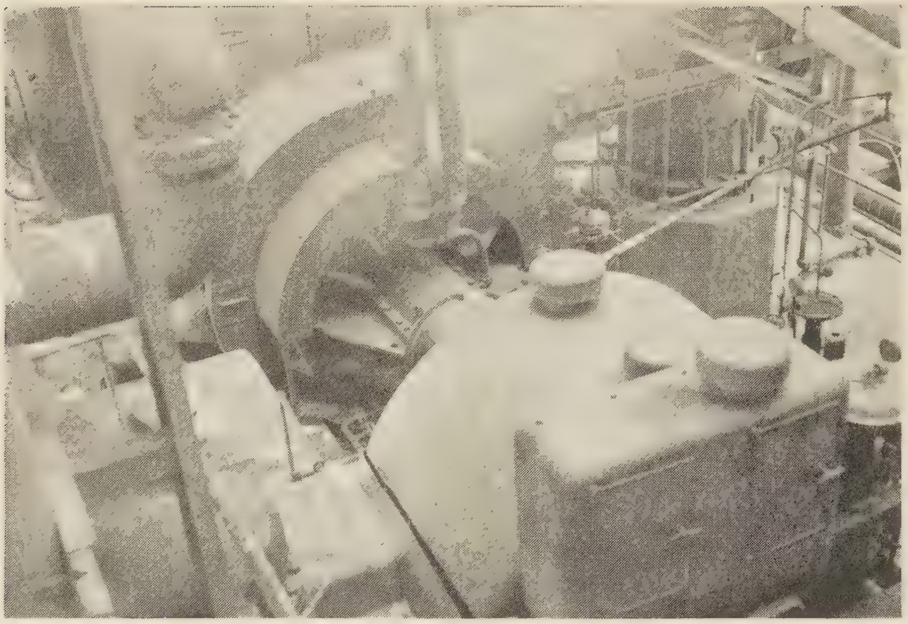

Fig. 9 Gas Turbine Operating on Products of Combustion at About 1000 F, at Union Oil Company, Oleum, Calif. (Manufactured by General Electric Company.)

locomotive (35), but with such temperatures as will give efficiencies competitive with any other cycles. However, there are those who argue that for moderate powers, the upper part of the high-temperature range which is necessary for high prime-mover efficiencies, can best be handled in a reciprocating engine, with the gas turbine to take the gases at a reduced temperature and carry the expansion to the limit. But whether or not this is going to be the ultimate is a question. There are three sorts of such cycles in present use where reciprocating internal-combustion engines exhaust into gas turbines. These may be called (*a*) Diesel-boiler or Götaverken-Sulzer cycle, (*b*) Büchi divided-manifold cycle, and (*c*) constant-pressure-turbosupercharger cycle; and will be discussed in this order.

(*a*) *Diesel-Boiler Cycle.* An early system was proposed by the engineer Johannsen (40) of Götaverken, a prominent shipbuilding concern of Gothenburg, Sweden, of which Mr. Hugo Heyman has been managing-director. A two-cycle Diesel engine exhausts at the high back pressure of about 4 atm and drives only its scavenging reciprocating air compressor, which is mounted on its crankshaft. The exhaust, cooled by a large amount of scavenging air, proceeds to the turbine, all of whose power is net. Sulzer Brothers, Ltd., of Winterthur, Switzerland, has been working on various modifications of the same general plan (41), as shown in Fig. 10. Free pistons also have been used.

(*b*) *Divided-Manifold Cycle.* Dr. Alfred Büchi of Winterthur, Switzerland, long has worked on turbosuperchargers and, as discussed next (43), made a very early proposal for the modern type with constant pressure, since used by others. Büchi and his associates, Brown, Boveri & Company, also have used what may be called a divided manifold (42). This has a Diesel engine with the usual sort of turbosupercharger, next discussed, except that nozzle-box sections and exhaust-manifold passages are divided so that a given exhaust passage into which exhaust is proceeding from one cylinder is never connected to another cylinder which is at the end of exhaust and beginning of intake. Büchi says there never can be "blowback" of exhaust gas into the intake manifold during the overlap period when both exhaust and intake valves of a cylinder are open at the same time. This cycle also has been used by Rateau.

In most cases, the cycle is used with Diesel engines having low exhaust temperatures and low speeds, as compared with aviation engines. The divided-manifold sections are of such reduced area as to avoid sudden drop of exhaust pressure. Furthermore, each cylinder exhausts by itself until its exhaust pressure dies out.

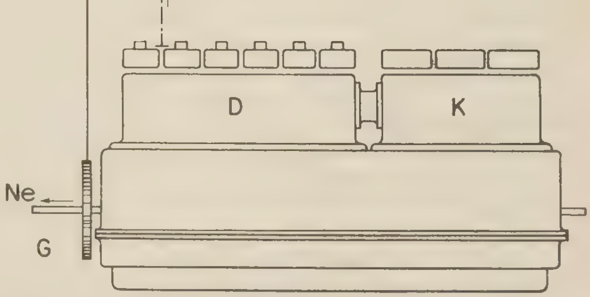

Fig. 10 Highly Supercharged Two-Stroke Diesel Engine, With Direct-Coupled Reciprocating Compressor and Mechanical Transmission Between Exhaust-Gas Turbine and Diesel Engine

(The exhaust-gas turbine is coupled by gearing, represented diagrammatically as chain gear, to the crankshaft of the two-stroke engine; Ref. 41.)

D Two-stroke Diesel engine
K Compressor
T Exhaust-gas turbine
G Gear between Diesel engine and exhaust-gas turbine
Ne Effective output at the shaft

So there is a variable nozzle-spouting velocity, which does not match the turbine-wheel speed during the entire time and thus causes a loss. The system of interlocking sections of the exhaust manifold connected to the different cylinders causes appreciable complexity.

With a turbosupercharger system of such efficiency that the intake-manifold pressure is nearly equal to or less than the exhaust-manifold pressure, the divided-manifold-system advocates say that it avoids "blowback" and gives scavenging instead, and so maintains full engine power. But with the aviation turbosuperchargers next discussed, the efficiency is such that the intake-manifold pressure is higher than the exhaust-manifold pressure, and so there is proper scavenging with the constant-pressure-turbosupercharger cycle.

History of Aviation Turbosuperchargers

The constant-pressure-turbosupercharger cycle first began practical use with high-altitude aviation. The constant pressure started as that of sea-level atmosphere, 30 in. of mercury abs, and the expansion and compression are to and from the low pressure of altitude atmosphere. The part of this aviation turbosupercharger in the gas-turbine picture will be begun with its history, by taking a flashback to World War I. Military aviation was in its infancy, but already there was difficulty in the loss of engine power even at the moderate altitudes then possible. Hence efforts were made to attach, by gearing, air compressors of various sorts, to give airplane-engine supercharging, but none was successful. General Electric later supplied geared superchargers having the design of the centrifugal compressor developed during the gas-turbine research, but these did not come into use until 1925.

During World War I Dr. William F. Durand was chairman of the National Advisory Committee for Aeronautics, and in the course of one of his duties of examination of schemes for aviation improvement, there came to him in the fall of 1917 a plan of the French engineer, Rateau, for using a turbine wheel driven by the products of combustion of the engine exhaust, at constant pressure, to drive a centrifugal compressor to do the supercharging. Dr. Durand had been a professor at Sibley College, Cornell University, in 1901, and he remembered the black smoke from the gas-turbine research and knew of the extensive business that the General Electric Company then had in steam turbines and centrifugal compressors. So he asked General Electric ($42^1/_2$) to undertake turbosupercharger development in the United States. General Electric engineers had their own ideas about design details from their experience with gas turbines, steam turbines, and centrifugal compressors, and so started a development entirely independent of Rateau, which has continued to this day, with production successively increasing on account of the present war.

The first General Electric turbosupercharger was constructed with wartime speed and was assembled on one of the Liberty motors then being used, by the U. S. Army Air Corps, McCook Field, Dayton, Ohio, in May, 1918, as shown in Fig. 11. It was operated there so far as possible at sea level and then tested at the 14,000-ft summit of Pike's Peak, Col.

The airplane turbosupercharger is so arranged that as the plane ascends to the successively lower atmospheric pressures at altitude, there continues to be maintained in the exhaust manifold and turbine nozzle box the normal sea-level pressure. The difference between this and the low pressure of altitude gives the nozzle jet that drives the turbine wheel. This drives a centrifugal-compressor impeller on the same shaft, which compresses the low-pressure air of altitude and supplies it at sea-level pressure to the engine intake manifold. So, regardless of altitude, the engine exhausts at sea-level pressure and receives its charge at sea-level pressure. So it continues to deliver sea-level power at all altitudes, up to the critical one where the turbine wheel reaches its maximum safe speed. Due to the low resistance of the low-density air at altitude, sea-level engine power gives increased airplane speed.

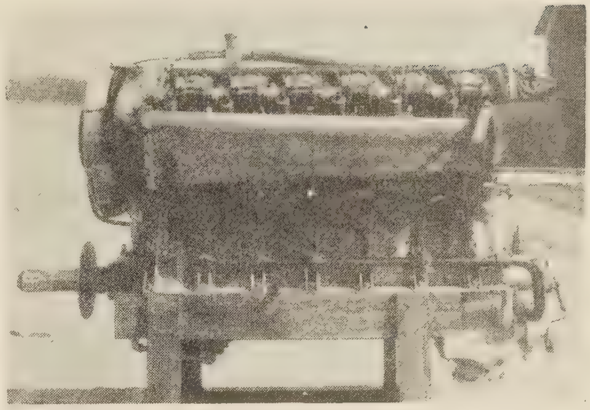

Fig. 11 First American Turbosupercharger on a Liberty Motor at McCook Field, Dayton, Ohio, May, 1918

The constant pressure of the engine-intake manifolds of the early airplane turbosupercharger was 30 in. of mercury. This is greatly exceeded nowadays. Values around 50 in. are usual for take off or short periods, and continuous operation is common with "boost" of intake-manifold pressure as discussed later.

Rateau's turbosupercharger development in France continued for a while after World War I and developments were started in England and Germany. But the resources devoted to the project abroad do not seem to have been sufficient to bring it to a state that warranted continuation. At any rate there is no evidence of any turbosuperchargers other than those of General Electric being used in the present conflict. Other differing American designs recently have begun to be developed.

It once was thought that the 1917 work in France was first mention of the modern turbosupercharger plan, but later searches show what seems to be an earlier publication by Büchi (43). Nevertheless, Rateau made important contributions (44).

In the United States, the development encountered the usual experience of the "glassy eye." But there were enough enthusiasts among the officers and engineers of the United States Army Air Forces, and enough help from them ($44^1/_2$), from engine and airplane manufacturers, and from metallurgists, and sufficient resources available to General Electric, to bring the project to its present originally undreamed of state (45).

The principal interest that the turbosupercharger has in connection with the gas-turbine problem is its development of a turbine to operate at high temperatures of products of combustion. It is true that the aviation turbosupercharger has only short-time operation at extreme conditions and does not have to meet the conditions of continuous operation in a power plant. Nevertheless, many problems have been solved, due to the impetus of World War II, which advance the prospects of a gas turbine as a prime mover. Thus the turbosupercharger, originally a gas-turbine by-product, has not only proved very useful of itself, but has accelerated gas-turbine development.

The turbosupercharger in the United States started in 1918 with a turbine wheel suited only for steam temperature, provided in a rush by the De Laval Steam Turbine Company. Since then there has been continuous progress in materials, fabrication, heat-treating, design, and cooling, of all of the parts, whether exposed to exhaust-gas temperatures or not.

Originally the nozzle area covered only a portion of the wheel

periphery, with cooling between, and this may be an item in the ultimate gas turbine. But for many years the turbosupercharger has had a 360-deg nozzle arc. For years, the rated temperature at the nozzle-box entrance was 1200 F. This meant that the entire nozzle box and the outer portion of the turbine buckets were visibly red-hot, at rotative speeds of about 22,000 rpm. Successive improvements have permitted much greater rated temperatures. Since about 1930 a part of the General Electric turbosupercharger research has been a "hot-gas test" where turbines were operated in the laboratory on products of combustion, with all conditions as when flying at altitude. By opening a peekhole in the altitude chamber in which the turbine wheel was running, there was seen for the first time operation with red-hot turbine buckets, and colored photographs were taken. Since the author always has considered the turbosupercharger as merely a step on the way to the gas turbine, he has often exhibited as gas-turbine propaganda this really inspiring sight of a turbine wheel operating at about 1200 fps with buckets red hot.

Fig. 12 is an accurate reproduction of a water color taken from life, showing a recent turbosupercharger undergoing a standard factory endurance test with products of combustion. In this test, and in actual service, the exhaust-gas conduit from engine to turbine is incipient white, and the nozzle box varies from white to red. A few inches away is the air inlet of the compressor, with a temperature at high altitude of about 50 F below zero, the reproduction of which in the test covers the pipe with frost. Similar conditions exist in the many turbosuperchargers which are reported as successfully flying at high altitude in the war in Europe, Asia, Africa, and the South Pacific, with the *Flying Fortress* B-17 made by Boeing, the *Lightning* P-38 made by Lockheed, the *Liberator* B-24 made by Consolidated, and the *Thunderbolt* P-47 made by Republic. Some of this is shown in the Army illustration, Fig. 13. This is not the place to give details of turbosuperchargers, as this has been done by Reginald G. Standerwick and W. J. King, engineers of the General Electric supercharger Division, in a paper read at the Semi-Annual Meeting of the A.S.M.E., in Los Angeles, Calif., June, 1943, and published in the Transactions of the A.S.M.E. for January, 1944.

(*c*) *Constant-Pressure-Turbosupercharger Cycle.* Turbosuperchargers on American airplanes are examples of the third of the cycles previously mentioned, where reciprocating engines exhaust into gas turbines. They are designed for pressure ratios which give absolute pressures above sea-level value, when flying at altitude. This means that absolute pressures in the engine-intake manifold appreciably above sea-level value are easily possible at moderate altitudes. Such pressures are used for take-off or for regular operation under special circumstances, which is called "boost." Pilots flying airplanes with turbosuperchargers have instruments showing airplane altitude, intake-manifold pressure, turbosupercharger speed, various temperatures, and so on. They have rules and technical orders which give values for the readings of the various instruments which give values for the boost of engine-intake-manifold pressures safe for the engine and turbosupercharger. The greater the boost, the greater the engine horsepower and the greater the airplane speed.

Thus when a pilot with a turbosupercharger is in battle, he has means of greatly increasing his plane speeds at most altitudes, by increased boost. So, when an enemy plane is on his tail, does the pilot take out his technical orders, look at his instruments, and figure out what he ought to do? He does *not*, but subjects the turbosupercharger to pressures far beyond its test rating.

Extension of such operation, which easily is possible independent of aviation, would give a four-stroke Otto- or Diesel-cycle engine with a constant-pressure turbosupercharger, which is this third one of the cycles where a reciprocating internal-combustion engine exhausts into a gas turbine. It has been called a "compound internal-combustion cycle."

As already mentioned, reports of operation of turbosuperchargers in war show that they are subjected to conditions far exceeding their test rating. Therefore the average of the conditions in war has some relation to conditions of long-time operation at sea level to which a gas-turbine prime mover would be subjected.

High-Temperature Progress

One major item in operation of turbine wheels at high temperature is "creep," which is continuous nonelastic elongation, finally resulting in rupture. Hence stresses for a gas turbine to be operated in a power plant for years are lower than for a turbosupercharger in war. Information about creep is rapidly accumulating in connection with turbosupercharger as well as with other turbine work (46).

Reference already has been made (42) to Büchi, Brown-Boveri, and Rateau work on turbosuperchargers with divided manifolds, probably always with low-exhaust-temperature, low-speed Diesel engines. Accounts also have been published (47) of Swiss, French, and German use of turbosuperchargers on aviation engines, with Diesel or Otto cycles and with divided or constant-pressure manifolds. None of the apparatus thus referred to is in use in World War II, the only known turbosuperchargers being those of the United States Army Air Forces, which as already mentioned are in very extensive use.

Nevertheless, some progress is being made abroad with turbine wheels operated by products of combustion. One example is a publication by Brown-Boveri (48), which includes a night picture of a turbosupercharger for an aviation engine (probably with a constant-pressure exhaust manifold) in a factory test at about 1800 F and 30,000 rpm. Glowing exhaust-gas conduit and nozzle box are shown with the remark, "The set presents an exceptional and fascinating appearance. It is surrounded by the glowing gas pipe and carries the mind back to some medieval machine."

Thus the thousands of turbosupercharger turbines, in operation by the United States Air Forces, and being investigated abroad, are adding to the other gas-turbine items already mentioned, in paving the way for temperatures much higher than those which have been used, and with a minimum of loss from cooling. No thermodynamic studies can be given here, but it is well understood that such higher temperatures mean higher gas-turbine efficiencies. So the long sought gas-turbine prime mover, with temperatures much beyond the 1000 F now in use, and consequent higher efficiency, seems to be getting nearer. Of course, this same progress is paving the way for higher temperatures and higher efficiencies with the regular steam cycle and with two fluid cycles. So we only can speculate as to whether the gas turbine with its compressor, combustion chamber, and heat interchanger is going to compete with the other cycles which instead have economizers, feedwater heaters, feed and air pumps, boilers, and condensers.

Acknowledgment

The author is greatly indebted to many individuals and companies who have been kind enough to furnish data, references, and illustrations which are used in this paper.

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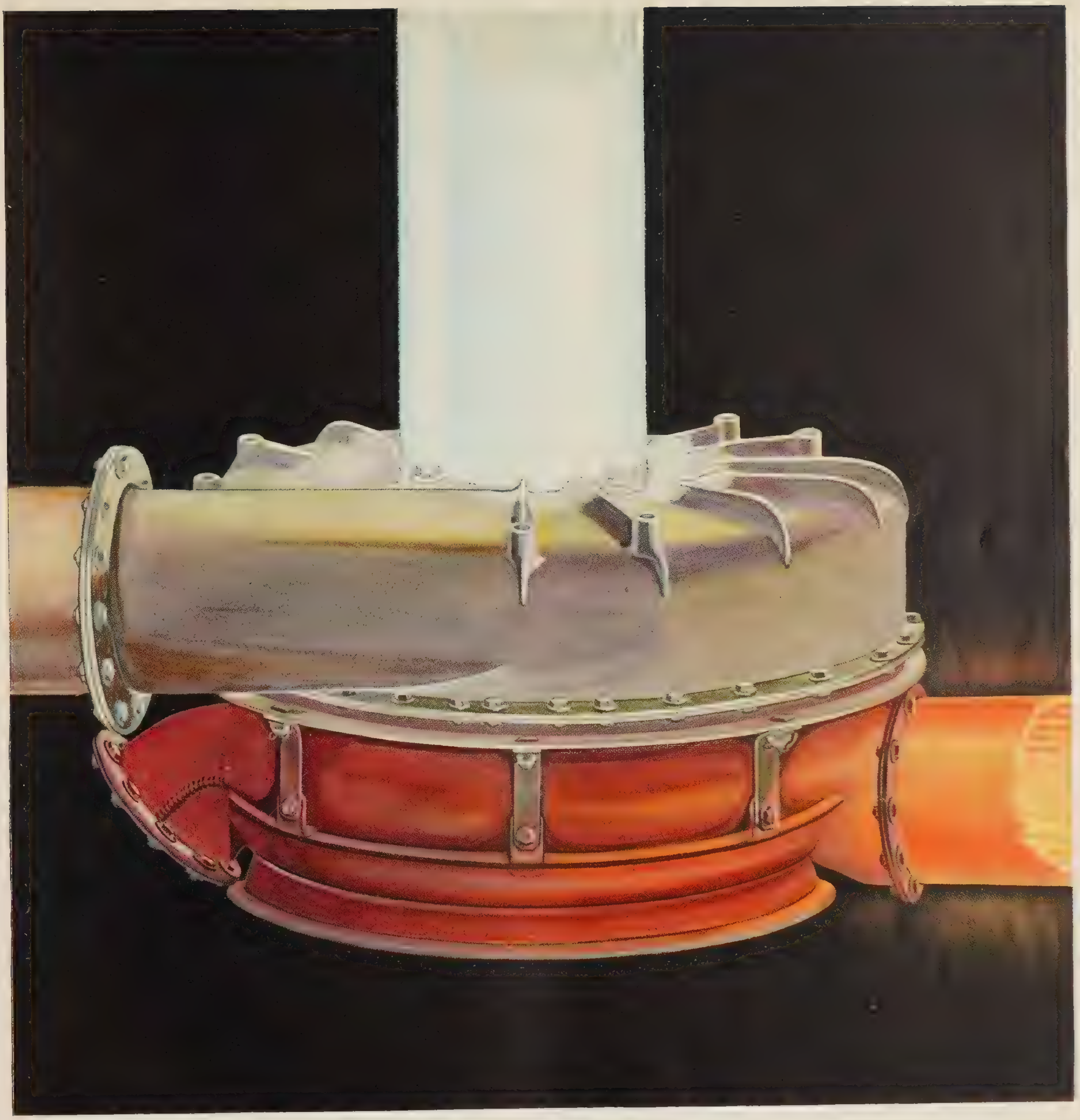

FIG. 12. GENERAL ELECTRIC TURBOSUPERCHARGER OPERATING UNDER STRATOSPHERE CONDITIONS
(Painting from laboratory test.)

14½ "Measurement of High Temperatures in High-Velocity Gas Streams," by W. J. King. Trans. A.S.M.E., July, 1943, pp. 421–428.

15 U. S. Patent 1086755, "Elastic-Fluid Compressor," to Charles G. Curtis; filed Aug. 4, 1896; issued Feb. 10, 1914.

16 U. S. Patent 1086754, "Rotary Compressor, Blower and Pump," to Charles G. Curtis; filed June 14, 1902; issued Feb. 10, 1914.

17 *Engineering* (London), Jan. 26, 1912, pp. 112–113, gives a description and illustration of a Mather-Reynolds centrifugal pump of 1895, showing a diffuser and an impeller with axial inlet buckets; here reproduced as Fig. 7.

18 "Ventilateaurs et Pompes centrifuges Pour hautes Pressions," by A. Rateau. Extrait du Bulletin de la Société de l'Industrie Minerale, 1902. Engineering Societies Library, New York, N. Y.

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"Notes on Blast Furnace Operation With a Turbo Blower," by S. G. Valentine (with collaboration of Sanford A. Moss), Trans., American Institute of Mining Engineers, vol. 50, 1915, pp. 90–103. Details of operation of a machine, as given in the preceding paper.

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U. S. Patent 161733 to Moss and Robinson, Feb. 28, 1927.

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23 "La Turbine a gaz-son rendement," by Alfred Barbezat, *Revue D'Électricité*, Nov. 12–19, 1904, and *Schweizerische Bauzeitung*, no. 9, vol. 44.

24 "The Gas Turbine," by M. L. Sekutowicz, Memoirs of the Society of Civil Engineers of France, February, 1906, pp. 195–300.

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26 "The Lorenzen Gas Turbine and Supercharger for Gasoline and Diesel Engines," by Christian Lorenzen, *Mechanical Engineering*, vol. 52, 1930, pp. 665–672.

27 "The Internal-Combustion Turbine," by W. J. Stern, report No. 54 of Aeronautical Research Committee of British Air Ministry, September, 1920. Discusses a number of gas turbines, including the Holzwarth and Karavodine explosion turbines, and says: "Mr. Holzwarth's calculations are made in this slipshod manner, even when on non-controversial grounds; he makes errors and gets inconsistent results . . . . . the claimed horsepower results must be fallacious."

28 "Geared Centrifugal Superchargers for Airplane Engines," by Sanford A. Moss, *Journal*, Society of Automotive Engineers, August, 1930, pp. 148–153, 160.

"Centrifugal Compressors for Diesel Engines," by Sanford A. Moss, *Mechanical Engineering*, vol. 47, 1925, pp. 1075–1084, and vol. 48, 1926, pp. 473–474.

29 "A PQ Plane for Thermodynamic Cyclic Analysis," by Dr. Harvey N. Davis, Proceedings of the American Academy of Arts and Sciences, April, 1905.

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"Gas Turbine Combustion Chamber," by Glenn B. Warren, *Power*, June 28, 1921, pp. 1059–1060.

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31 "Gas Turbines," by Lionel S. Marks and M. Danilov, Trans. A.S.M.E., vol. 46, 1924, pp. 1095–1129.

32 "Recent Developments in Power Generation," by Lionel S. Marks, *American Scientist*, vol. 30, no. 3, summer issue, 1942, p. 171. Beginning p. 186 is an abstract of Professor Marks' 1939 lecture before the Harvard Engineering Society.

33 "The Combustion Gas Turbine," by Dr. Adolphe Meyer, Journal and Proceedings of The Institution of Mechanical Engineers, vol. 141, no. 3, May, 1939, pp. 197–222. Reprinted, with changes and additions, *Mechanical Engineering*, vol. 61, 1939, p. 645, and *Journal*, American Society of Naval Engineers, August, 1939, with discussion in *The Engineer*, London, March 31, 1939, p. 406. Also see *Brown Boveri Review*, June, 1939.

"The Brown Boveri Constant Pressure Gas Turbine, a Waterless Power Station," by Ad. Baumann and W. Broggi, *Brown Boveri Review*, April–May, 1939, pp. 103–106.

34 "Load Tests of a 4000-Kw Combustion-Turbine Set," by Prof. A. Stodola, *Engineering* (London, England), vol. 149, January 5, 1940, p. 1; reprinted, *Power*, February, 1940, pp. 62–63; *Mechanical Engineering*, vol. 62, 1940, p. 239. Describes a 4000-kw gas turbine stand-by power station in Neuchatel, Switzerland, with a thermal efficiency shown by test, of 16 to 17 per cent at about 1000 F.

35 "A New Gas Turbine Electric Locomotive in Switzerland," by F. Steiner, *Diesel Railway Traction*, Supplement of *Railway Gazette* (London, England), September, 1942, pp. 100–103; October, 1942, pp. 106–109 (with continuation).

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35½ "Some Aspects of Diesel Engines for Navy Main Propulsion," by E. W. Mills, *Mechanical Engineering*, vol. 65, 1943, pp. 625–627.

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38 "Turbines for Power Generation From Industrial-Process Gases," by John Goldsbury and J. R. Henderson, of the Lynn Turbine Division of General Electric Company, Trans. A.S.M.E., vol. 64, 1942, pp. 287–298. This includes details, Figs. 1, 2, and 3, of the plant herein shown in actual operation as Fig. 9.

39 "Electrical Developments of 1942, Power Generation Transmission, Measurement," by Guy Bartlett, *General Electric Review*, vol. 46, January, 1943, pp. 8, 9, 10; see also *Power*, September, 1940, pp. 561–567; and September, 1941, pp. 655–663. Gives tables with details of many steam plants above 900 F.

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42 "Supercharging of Internal Combustion Engines With Blowers Driven by Exhaust Gas Turbines," by Dr. Alfred Büchi, Trans. A.S.M.E., vol. 49, 1937, pp. 85–96; vol. 60, 1938, pp. 430–456. See also many references in *Brown Boveri Review*.

42½ Copy of letter of 1917:

National Advisory Committee for Aeronautics
Washington, D. C., November 19, 1917

Mr. E. W. Rice
General Electric Company

Dear Sir:

At the present time in connection with the development of the airplane and the Liberty engine for driving same, there is not a problem which is perhaps more seriously before the Government officials than that of maintaining at high altitudes the power of the engine. In this connection we are especially desirous of developing the possibilities of a precompression of the air before going to the engine carburetor, thus maintaining as nearly as may be constant conditions regarding pressure at all altitudes.

In this connection we believe that your Dr. Sanford A. Moss could be of very great service in advising with us regarding the possibilities

of turbo-compression as applied to this problem, and I have written informally to Dr. Moss inviting his attention to the problem as outlined.

I desire, therefore, to ask your approval of this request, and that, if possible, you will authorize Dr. Moss to give to us the results of his experience in the study of such matters.

Thanking you in advance for your anticipated cooperation in this matter, I remain,

Very truly yours,
/s/ W. F. Durand
Vice-Chairman

43 "Über Vebrennungskraftmaschinen," by Dr. Alfred Büchi, *Zeitschrift fur das gesamte Turbeninwesen*, 1909, pp. 313–316, 322–332, 347–351. Clearly shows a turbine wheel driven by engine exhaust gases on a shaft separate from the engine shaft, and driving a "turbokompressor" which supplies air through a "kuhler" to the cylinders of a reciprocating internal engine of either what is now called the "Otto" or "Diesel" types. Mentions the use of the process on "aircraft, automobiles, motorboats, railway motors, stationary engines," etc. Refers to German patent 204, 630, filed Nov. 16, 1905, issued Nov. 28, 1908; but this shows turbine and compressor on the engine shaft.

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"The First Gas Turbine Locomotive," by Dr. Adolf Meyer, The Institution of Mechanical Engineers (London), Journal and Proceedings, November, 1943.

Discussion

A. G. Christie.[3] The author has presented an interesting history of the development of the gas turbine and turbosupercharger. He discusses his early experimental work at Cornell University using the DeLaval steam-turbine wheel of the unit that was shown at the Chicago World's Fair and which was said to be the first De Laval unit in this country.

The writer went to Cornell in October, 1904, as an instructor. When the late Prof. R. C. Carpenter, then in charge of the mechanical laboratories, learned that the writer had built steam turbines, he asked whether or not the old De Laval turbine could be fixed up and run. Professor Carpenter said that a former student had tried to use this rotor as a gas turbine and, as a result of overheating, the shaft had broken.

Balancing was not a refined art in those days. The De Laval designers provided a flexible shaft to care for wheel unbalance. This shaft was reduced to a small diameter (about ⅜ in. as nearly as can be recalled) on either side of the disk. The fracture occurred in one of these small sections. The shaft was either repaired by a sleeve or replaced, for the turbine was in service the following winter for laboratory purposes.

The blading of the wheel showed no ill effects of its operation as a gas turbine. In view of our more extended knowledge of fatigue failures, it is quite certain that the shaft fracture was due to fatigue rather than to Dr. Moss's use of it as a gas turbine. The shaft was well away from the wheel, and the wheel itself showed no ill effects from the application of gas. The writer recalls that Professor Carpenter said the wheel had been "very hot," so his curiosity was aroused to see what had been the effect on the blades and disk. Apparently, the effect was not serious for the unit ran for some time afterward in regular laboratory service.

Later, about 1912, when the writer was at the University of Wisconsin, the late Prof. Carl C. Thomas had him fit up a small gasoline engine with a quick-opening valve on the indicator-cock connection. This valve was opened somewhat before the end of the power stroke and the gas was discharged through a nozzle onto a small turbine wheel. This ran satisfactorily and developed a small amount of power but the hot gases quickly de-

[3] Professor of Mechanical Engineering, The Johns Hopkins University, Baltimore, Md. Fellow, A.S.M.E.

stroyed the valve. The writer believes that Professor Thomas later filed an application for a patent on this device as a power-generating unit but no further construction or experimental work was done.

W. F. Durand.[4] In this paper, the author has presented a vast amount of historical information on the development of the gas turbine which will be of the greatest interest and value to all who are concerned with this new form of combustion engine. My own small part in this history goes little beyond the letter which I wrote in the fall of 1917, to Mr. Rice, president of the General Electric Company, asking him to detail Dr. Moss on the problem of the exhaust turbosupercharger, as an aid to the airplane in reaching higher atmospheric levels.

In 1917, the members of the National Advisory Committee for Aeronautics were becoming acutely impressed with the need of some means for the maintenance of sea-level power at high altitudes. We knew something of what Rateau in France had been doing in this line, and the salvaging of some of the power wasted in the exhaust seemed a promising means for achieving this end. As Dr. Moss has noted in his paper, I had been in contact with his work at Cornell in 1901 and knew of his connection with the General Electric Company in 1917; and hence it seemed to be a most natural approach to address the letter to Mr. Rice as he has noted. The response was prompt and generous and the work was started.

The growth, since that beginning in 1917, has far outrun anything that either Mr. Rice, Dr. Moss, or I could have possibly foreseen or even imagined at that time. The turbosupercharger is firmly established as an absolutely essential feature of aircraft motive equipment where navigation at high altitudes is contemplated, and the rate of production in recent months has reached figures which, 26 years ago, would have seemed nothing short of a fantastic dream.

The splendid record of achievement in this particular domain of power drive may well be an indication of continually improving adaptation of the gas turbine as a prime mover, to the requirements of engineering and industry broadly.

J. S. Haverstick.[5] The author has presented an interesting and instructive account of his work, and the work of others, in attempting to develop a commercial gas-turbine power plant. It is to be hoped that recent wartime developments, both in this country and abroad, may soon be published for the benefit of those who design this type of equipment.

This year is the fiftieth anniversary of the first De Laval steam turbine in the United States. We were interested to note that its bucket wheel was probably the first to operate on exhaust gas; and, further, that 25 years later De Laval was the source of supply for high-speed wheels.

Since mention has been made of the bucket wheel supplied to the author in 1918, we feel that we should add a few notes on the other development program which paralleled that of the author, and with which De Laval was connected more intimately.

Fig. 14 of this discussion is a copy made from the drawing by Rateau and is probably the starting point for turbosuperchargers in this country.

During 1918, tests were conducted at De Laval for the Fergus Motors of America, Inc., using the turbosupercharger developed the previous year by E. H. Sherbondy. Several changes in design were made as the tests progressed, for it was soon found that

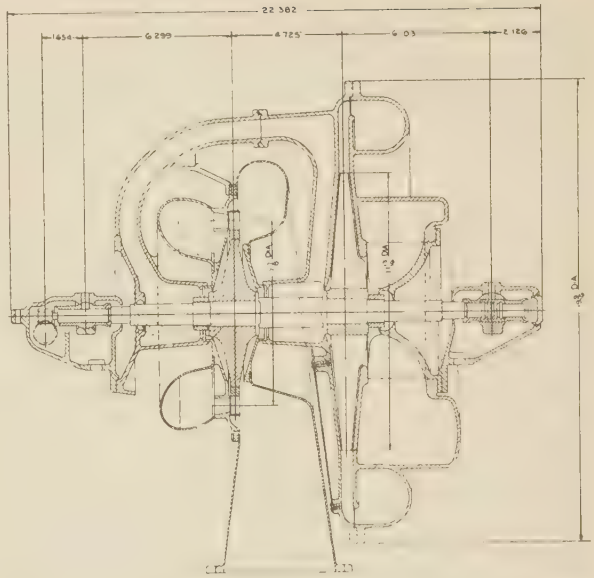

Fig. 14 Turbocompressor Operating at 20,000 Rpm for 330-Hp Motor

(Copy of French drawing from Rateau, dated November 3, 1917.)

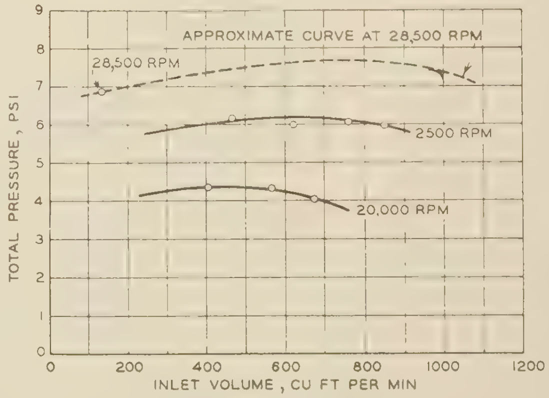

Fig. 15 Characteristic Curve, Showing Performance of Centrifugal Compressor

the unit could not be brought to its full speed of 30,000 rpm because of congestion in the flow passages. Fig. 15 is a characteristic curve of the compressor taken before completion of the tests.

Unfortunately, this program was discontinued after the last war. For a description of the machine one should refer to Stodola's text (25),[6] where it is shown in cross section alongside the General Electric turbosupercharger of that period.

Richard Herold.[7] This paper presents a valuable chronological history of the gas turbine with a quite natural emphasis on the work done in this country. Due no doubt to the imposition of wartime secrecy, the paper does not add anything to the facts already known; it does not submit any figures about present results, and it does not offer suggestions about the probable aspect of further gas-turbine development.

The writer feels therefore that while the paper is of great

[4] Chairman, Special Committee on Jet Propulsion, National Advisory Committee for Aeronautics, Washington, D. C. Past-President, and Honorary Member, A.S.M.E.

[5] Mechanical Engineer, De Laval Steam Turbine Company, Trenton, N. J.

[6] Numbers in parentheses in this discussion refer to the author's Bibliography.

[7] Sulzer Bros., Ltd., New York, N. Y. Jun. A.S.M.E.

value for all those interested in gas turbines, it does not offer material for discussion. It is regretted that the author has not mentioned the studies which he has surely made, with a view to using gas turbines for jet-propulsion high-altitude aircraft. This again, is unquestionably due to wartime secrecy.

W. J. King.[8] One of the most remarkable aspects of this subject of gas turbines is the rapidity with which interest has developed within the very recent past, as manifested in the published literature and in the transactions of several of our engineering societies. Along with this, there seems to be a similar growth of interest in various combinations of the gas turbine with other types of heat engines, such as the Velox boiler and the "Diesel boiler" or Götaverken cycle which the author mentions.

After reading the article on the Diesel boiler, to which the author refers, the writer became interested in comparing the theoretical efficiency of the particular cycle described with that of the simple gas-turbine cycle. Of course the actual performance of either cycle will depend to a major degree upon the efficiencies of the component machines, but it was felt that the results of a basic thermodynamic analysis using idealized compression and expansion, with adiabatic or 100 per cent efficiency, would be of some interest.

The cycles assumed in this study are indicated in Fig. 16 of this

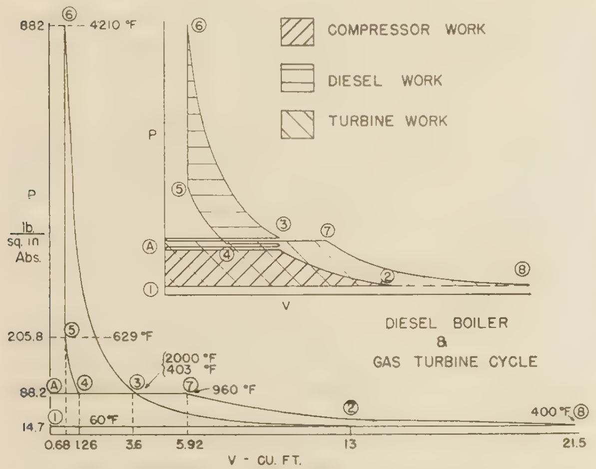

Fig. 16 Theoretical Indicator Diagram for Gas Turbine and "Diesel Boiler" or Götaverken Cycles

(Lower diagram is to scale, upper diagram is schematic, to show sequence of operation.)

discussion, in which the "Diesel boiler" cycle is superposed upon the ordinary gas-turbine diagram. In either case, atmospheric air is initially compressed through a 6:1 ratio in a separate compressor, which discharges it at 88.2 psia and 403 F, at point 3. In the simple gas turbine, this air is heated in the combustion chamber to a temperature of 960 F, at constant pressure. It is then expanded down to atmospheric pressure in the turbine. The efficiency, defined as the difference between the turbine work and the compressor work divided by the heat input, is 40 per cent.

In the Götaverken cycle, all of the air discharged from the compressor is taken into the Diesel cylinder, point 3, but on the compression stroke the exhaust valve is not closed until 65 per cent of the charge has been expelled into the exhaust-receiver feeding the turbine. The remaining 35 per cent is compressed along line 4–5 to a pressure of 205.8 lb, where it is mixed with fuel, ignited, and heated to 4210 F at 882 lb, point 6. It is then expanded isentropically to point 3, where it is discharged into the receiver at 2000 F. The mixing of this hot gas with the 403-deg air, previously expelled into the receiver, results in the same temperature, pressure, and weight of gas at point 7 as before. The work done in the turbine is, therefore, exactly the same as in the previous case, but the cycle has been so chosen that the net output of the Diesel supplies the work done in the compressor. The efficiency is, therefore, equal to the total turbine output divided by the heat input between points 5 and 6, which turns out to be 49.2 per cent. This particular cycle is thus 23 per cent more efficient, on this idealized basis, than the simple gas turbine.

The advantage of the Diesel boiler cycle in this instance cannot be attributed to the fact that the maximum temperature of the cycle is much higher than in the simple gas turbine. With isentropic compression and expansion, as assumed herein, it is interesting to note that the over-all efficiency of the gas turbine is independent of the temperature of the gas entering the turbine. The superiority of the Diesel cycle lies in the fact that it supplies the compressor work more efficiently by virtue of the greater expansion ratio in the Diesel cylinder, 5.30, as compared with 3.63 in the gas turbine.

On the basis of actual compressor and turbine efficiencies, substituting the rather optimistic figure of 85 per cent for both units, the over-all gas-turbine efficiency drops from 40 to 16.4 per cent. If the Diesel output is likewise 85 per cent of the theoretical value, the actual efficiency of the Götaverken cycle would be reduced to about 30 per cent. With lower efficiencies of the components, it may be seen that the gas-turbine cycle rapidly collapses to zero output, at the specified inlet temperature of 960 F, unless the Diesel or some other engine is provided to drive the compressor. These considerations indicate that the addition of a relatively small piston engine in this manner may make it possible to utilize a gas turbine as the final drive under conditions where it would otherwise be impracticable.

R. C. Muir.[9] The author's closing statement shows great restraint, when one considers the faith that he has in the ultimate successful development of the gas turbine, a faith supported by study, experiment, and research on the gas turbine throughout his entire engineering career of nearly 50 years; and those who know the author know that these were years of great activity.

It must be with great satisfaction that he has seen the gas turbine become such an important part of the gas-and-oil-engine cycle, more specifically in the form of the turbosupercharger which has given us superior airplanes.

However, he does speak encouragingly when he says the gas-turbine prime mover seems to be getting nearer, notwithstanding the equities and probable improvements in other heat cycles.

The gas-turbine prime mover has been slow in coming because its principal elements, the compressor and the turbine, had to be highly perfected before the gas turbine had any output. Other forms of prime movers gave some output even though each element was very inefficient. With efficiencies of turbines and compressors in the 80 to 90 per cent bracket, and with materials capable of high stress and low creep at temperatures of more than 1000 F, we have arrived at a point where the gas-turbine prime mover is a fact.

There are some applications where light weight, low space factor, and scarcity of water are necessary, and these are the applications for which the gas turbine should be studied today,

[8] Supercharger Engineering Division, General Electric Company, West Lynn, Mass. Mem. A.S.M.E.

[9] Vice-President in Charge of Engineering, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

even though its thermal efficiency is less than that of some other heat cycles.

However, it is probable that higher efficiencies of turbine and compressor will be obtained, and that materials capable of higher temperatures under stress conditions in the turbine will come through metallurgical research, and with these the gas-turbine prime mover will become a legitimate competitor of other forms of prime movers in more extended fields of application.

Frederick Nettel.[10] The story of the good old days of the gas turbine, when everybody was happy if it turned at all, as related by the Nestor of American gas-turbine engineers, is truly fascinating and will forever form a valuable contribution to the history of this oldest and, at the same time, newest, power machine.

The historic references may be rounded out by mentioning a few pioneer thinkers, whose contributions to the art, even though in the form of patent specifications only, seem important enough, or may become so, considering the more recent trend of developments in gas or air turbines. These are as follows:

1 U. S. Patent 18,435 to Philander Shaw of Boston, dated May 2, 1854: Reissue No. 1014, dated July 17, 1860, which discloses the principle of passing the air exhausted from a hot-air engine through the fire (in the furnace) for purposes of economizing heat. This is important in connection with efforts to realize coal-burning air-turbine plants.

2 British Patent 18,435, convention date Norway, Aug. 26, 1903, to J. W. A. Elling of Christiania (Oslo) showing an early example of gas-turbine plant with multiple-stage centrifugal compressor embodying precooling and intercooling of the air in two stages, in combination with one turbine and recuperator.

3 German Patent 216,191 of Sept. 12, 1906, to S. Z. de Ferranti of Sheffield, England, illustrating probably for the first time useful power turbine and compressor driving turbine on separate shafts. The turbines are arranged in series as regards the flow of the combustion gases, with the low-pressure turbine driving the compressor. Included is the first discussion of regulating problems in multishaft gas-turbine plants.

4 U. S. Patent 853,124 to E. L. Schaun of Baltimore, May 7, 1907, describes for the first time starting of a gas turbine by compressed air.

5 U. S. Patent 1,316,234 of Sept. 16, 1919, to J. O. Heinze of Springfield, Ohio, mentions most likely for the first time gas turbines with oppositely rotating runners used for neutralizing gyroscopic action on vehicles with particular reference to airplanes. (This was in the meantime reinvented several times!)

While the author quite logically omits detailed thermodynamic computations in his predominantly historical paper, he draws certain conclusions from a simple approximate calculation which at this date tempts the "red-hot" gas-turbine enthusiast to a few remarks:

Much has happened since the days when it was thought possible to dispose of the gas-turbine problems "comprehensively" or even "definitively" by thermodynamical calculations of probable efficiencies along textbook lines.

Without wishing to go into details in this discussion, it appears timely to take cognizance of the fact that during the last few years much fundamental work has been done all over the world on the thermodynamics of the gas and air turbine. This brought to the fore, among other things, the long underestimated value of intercooling and reheating of the working fluid. Due to the war, and other considerations, much of this work will have to remain unpublished for some time. However, it seems opportune here to emphasize in the interest of the healthy development of the gas turbine, that its future as a serious competitor of the steam turbine, and under certain circumstances even of the Diesel engine, is today no more predicated upon the employment of extraordinarily high temperatures.

[10] Manhasset, L. I., N. Y.

In short, a modern gas-turbine plant of about 4000 hp, or preferably more, working at a maximum gas temperature of 1050 F, consisting of two stage compressors with interposed intercooler, two turbines with interposed reheater, and a recuperator of acceptable size, can be designed and built to give a rated-load thermal over-all efficiency of 30 per cent, referred to the coupling of the useful power turbine. The efficiencies assumed in that plant for both turbines and compressors, as well as the parasitic losses (pressure drops, etc.) are well within the range of figures measured in actual plants (Neuchâtel, etc.)

Steam-turbine designers know only too well why they prefer not to go much above 1000 F at present, in spite of the fact that the steam turbine would also profit from a rise in superheat, particularly in high-pressure plants. If they thus avoid what they consider grave risks, evaluated from vast experience, there is little basis for crediting gas-tu bine designers, with much less to build upon, with greater ingenuity or wizardry in dealing with 1200 F, or even 1500 F.

Since, as mentioned, the gas turbine must be considered a potential thermal and—on account of savings in weight, space, and cooling water—economic rival of high-pressure steam plants, even when employing gas temperatures in the range of what has been accepted for some years for steam turbines, higher temperatures should be resorted to only with the greatest caution and gradually.

Setbacks due to gambling with higher temperatures are liable to be unjustly blamed on the gas turbine and are bound to retard rather than to assist its healthy development.

J. T. Rettaliata.[11] The author, in devoting his efforts since the beginning of the century to the advancement of the gas turbine, deserves much credit for the recognition it is receiving at the present time. In his interesting paper, he rightly points out the antiquity associated with the gas-turbine art, thus correcting any erroneous beliefs, which may have arisen due to the recent attention being accorded it, that the gas turbine is a modern invention.

The author may well be proud of his contributions in the many years of development of the turbosupercharger, a device playing such a vital part in the present conflict. The modern turbosupercharger is exhibiting admirable performance in the type of installation now being employed. The efficiencies of its respective compressor and turbine elements are adequate in the present application where no additional power is required from the turbine beyond that needed to effect compression.

Since the aim of the aviation industry is ever to decrease the weight per horsepower of the power plant, the next logical modification of the turbosupercharger would be to achieve compressor and turbine efficiencies of the highest order so as to produce an excess of power. This additional power could be fed back into the engine or propeller shafts or utilized in the form of a jet-propulsion effect from the turbine exhaust gases.

The large-scale prime-mover type of combustion-gas turbine has benefited and will, undoubtedly benefit from turbosupercharger experience but it should be remembered that they are basically different types of units. The supercharger approaches a limit design in that minimum-weight requirements necessitate relatively high speeds and stresses resulting in a limited service life. Some of the high-temperature metallurgical problems of the supercharger therefore differ from those of the more conservatively designed, longer-life, prime-mover type of gas turbine. For the supercharger, materials having good stress-rupture properties are desirable, whereas, for longer-life units creep char-

[11] Chief Research Engineer, Steam Turbine Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Jun. A.S.M.E.

acteristics may be more important. Further, high-temperature alloy castings and forgings in sizes suitable for superchargers are easier to produce than those required for large-size gas-turbine units.

As a prime mover, the combustion-gas turbine is quite versatile owing to the many different kinds of cycles that can be employed; giving a wide range of thermal efficiencies. The basic cycle, embodying a single turbine, compressor, and combustion chamber, will represent the lowest part of the range. Fortunately, it can be modified so, when operated with temperatures permissible with materials now available, good thermal efficiencies are possible.

One such cycle modification, comprising intercooling in the compressor, reheating between the power and compressor turbines, and preheating of the compressor discharge air with the turbine-exhaust gas, is shown in Fig. 17 of this discussion. This cycle is capable of yielding a coupling thermal efficiency of 30 per cent when operated with turbine inlet temperatures of 1200 F, a pressure ratio of 5, and 1500 and 5000 sq ft of surface in the intercooler and heat exchanger, respectively.

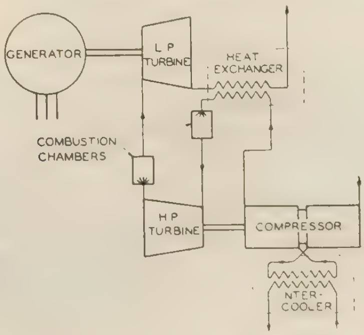

Fig. 17 Gas-Turbine Regenerative Cycle With Reheating and Intercooling

As has been pointed out previously on many occasions the gas turbine has progressed mainly through the development of better materials and more efficient compressors. As a further step, assuming thermal efficiencies of 30 per cent for the present, it will probably be necessary to adapt pulverized fuel to justify economically the general use of the gas turbine in central-station practice. Such a procedure appears to have merit considering the appreciable difference in cost between coal and oil and the possible future scarcity of the latter.

The major commercial application in this country of the combustion-gas turbine in prime-mover sizes has been in oil refineries employing the Houdry catalytic cracking process in the manufacture of high-octane gasoline. Figs. 21 and 22 of Mr. Sidler's discussion show the first of these machines supplied by Brown Boveri. Many other units of the same type have since been built by Allis-Chalmers under Brown Boveri license. To date Allis-Chalmers has supplied 27 such gas turbines to the oil industry and much valuable operating experience has been obtained with these units.

The well-recognized limitations of the gas turbine preclude the possibility of its being considered as the answer to all future power problems. It is believed, however, that its inherent characteristics make it an ideal aspirant for certain applications in land, heavy traction, aeronautical, and marine fields.

P. R. Sidler.[12] In the course of the last 3 or 4 years several papers on gas turbines have appeared in the technical literature; some also have been presented before meetings of this Society. It is perhaps more than a coincidence that quite a few of them, notably those covering the subject in greater detail, devote very considerable space to a historical survey. Possibly Dr. Meyer's paper (33)[13] before The Institution of Mechanical Engineers and before this Society in 1939 started this trend. At any rate, several of the early gas-turbine "inventors" mentioned there have since reappeared in a number of papers, only to be summarily dealt with, as having contributed rather little that could be of use today.

It is interesting to speculate on the purpose of such extensive backward glances, especially if it is admitted that most of the gas-turbine proposals advanced during the first decade of this century were impractical or not workable at all. One good reason may be the desire to establish priority of ideas; another to make a case for continuity of research and study undeterred by early failures or lack of appreciation; still another could be to show why there is as yet in many quarters only a modest beginning on the road toward real gas-turbine power machinery.

Whatever the reasons, such historical surveys do help to focus attention on the many difficulties that stood in the way of a practical gas turbine which actually delivers useful power. They also emphasize that modern technical accomplishments are not the invention of any individual but the result of a co-operative effort of many minds, a structure slowly built up on primitive and often rather unsafe foundations. Again, the gas-turbine development shows that even large numbers of patents, issued through the years to many corporations and individuals in various countries, do not necessarily insure success. What really counts is the slow but consistent accumulation through tests and research of data and experiences, the building-up of shop techniques, and of "know-how."

The present paper is a very interesting account of the author's struggle with gas-turbine problems for over 40 years and of his successes in the field of turbosuperchargers for airplanes. The very detailed Bibliography should also be of definite value to the student of gas-turbine matters.

The fundamental work done in this field by Prof. A. Stodola (25) would seem to merit greater emphasis. A large part of his life work as research scientist, teacher, and consultant had been devoted to the development of the gas turbine, and he had the satisfaction of seeing his dreams come true (34), before he died on Christmas Day, 1942.

In the author's discussion on compressors for gas turbines there are a few points where the picture might be rounded out on the basis of available records.

It is stated that Rateau supplied centrifugal compressors for the Armengaud-Lemale gas-turbine work carried out in France. It is known that Professor Rateau had some preliminary models of single-stage centrifugal compressors built for this experimental gas-turbine work, but when the definite gas-turbine design was evolved in 1906, and the need for a larger centrifugal compressor developed, Mr. Armengaud turned to Brown Boveri. Fig. 18 of this discussion shows this machine on the test floor of the Brown Boveri factory in Switzerland. This is a three-cylinder unit with 25 impellers, built for about 2150 cfm at 55 psig, 4000 rpm, and required about 328 hp. It showed an efficiency between 65 and 70 per cent, depending upon the load.

This was incidentally the first commercial centrifugal compressor in the world and it was so successful that Brown, Boveri & Company immediately launched a manufacturing program for this new machine. Between 1906 and the end of 1910, a total of nineteen centrifugal compressors or blowers for blast-furnace operation were installed, among them six in England, three each in France and Chile, two each in Belgium and in Canada, and single units in a few other countries. Their air

[12] Resident Engineer, Brown, Boveri & Co., Ltd., of Baden, Switzerland, 19 Rector St., New York, N. Y. Mem. A.S.M.E.

[13] Numbers in parenthesis in this and subsequent discussions refer to the author's Bibliography.

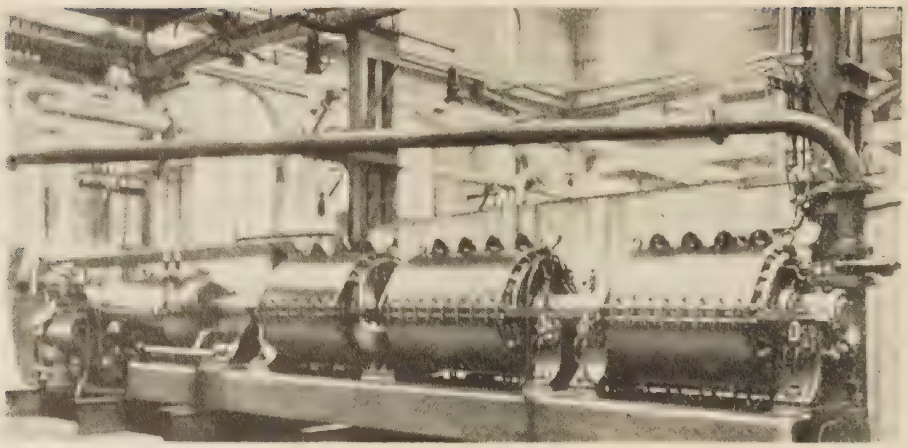

Fig. 18 Three-Cylinder Turbocompressor Built in 1906 for Armengaud-Lemale Gas Turbine

volume varied between 5000 and 32,500 cfm. During the same period Brown Boveri also completed eight turbocompressors for pressures between 62 and 103 psig and for installations in France, England, and other countries, besides about a dozen centrifugal blowers in various applications. It is interesting to note that this development finds a close parallel in the decisions of the General Electric Company, which installed the first two blast-furnace turbocompressors in this country in 1910.

With this background in centrifugal-compressor work and fully realizing its limitations, Brown Boveri worked on the development of a more efficient rotating machine and found the solution around 1931, in an improved axial-flow compressor, the principle of which had already been known for over 20 years. In fact Parsons had built several machines of this type in the period, 1905–1908, but had abandoned the design as it did not then show any advantage over the centrifugal compressor. It was only by applying newly developed data from the field of aerodynamics that a more efficient axial-flow compressor could be built. Since 1932, there have been put into commercial operation over seventy-five multistage axial-flow compressors in Velox plants, some forty units in gas-turbine applications, and about a dozen large machines up to 128,000 cfm in other fields, such as wind tunnels for airplane research, refrigerating plants, etc. Operating results on some of these plants have been published repeatedly and prove that the axial-flow compressor is indeed much more efficient than a centrifugal machine. Adiabatic efficiencies between 82 and 86 per cent have been consistently obtained (34).

Fig. 19 of this discussion shows the cylinder of an axial compressor built about 1933, for a Velox plant. The machine has 10 blade rows and produces a pressure of 36 psig. Fig. 20 shows the rotor for the same machine. To be sure, the pressure ratio of the axial-flow blower is at present somewhat lower than that of centrifugal compressors, but this is not considered a serious handicap for successful gas-turbine operation. Their greater number of stages on the other hand is no cause for apprehension either. There are many thousand steam turbines of the reaction type in service in the world today which have even more blade rows and work under more severe conditions of pressure and temperature.

This higher efficiency of the axial-flow compressor is one of the basic reasons for the existence of a practical gas turbine today, that is, a prime mover which actually delivers useful power and does not use up all its output for driving the air compressor.

Incidentally, the axial-flow compressor is a perfect illustration for the statement that patents by themselves do not necessarily result in practical machinery. It is rather the careful attention to details in manufacturing, rigid inspection, and thorough testing, in short, "know-how," which insures success. Even seemingly minor deviations from these rules may spell the difference between a machine that goes into service immediately after erection and another that requires perhaps several months of adjustment and partial rebuilding.

It is true that the application of such machines in the Houdry catalytic cracking process does not constitute a self-contained prime mover. Their main purpose in this process is the delivery of substantial air volumes. Nevertheless, all these units (nearly thirty machines are now in operation in the United States, the first eight being imported from Switzerland between 1936 and 1939) show a net surplus of power and are thus properly classified as gas turbines. In most cases this surplus is fed back into the electrical system, except where local conditions or contractual arrangements make it desirable to take it out in the form of heat.

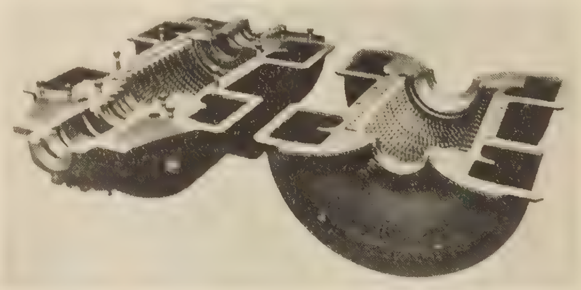

Fig. 19 Upper and Lower Halves of Axial-Compressor Cylinder for Velox Boiler

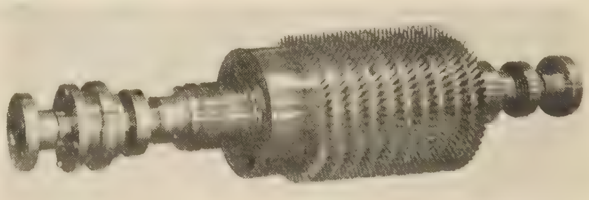

Fig. 20 Rotor of Axial Compressor as per Fig. 19, With Roller Bearings and Coupling Flanges

Fig. 21 shows the rotating element of the first gas-turbine unit for the Houdry process, built in 1936, and in continuous operation since January, 1937, except for a few days of inspection and maintenance work on the whole plant, every 6 months. To the right is the five-stage gas turbine, to the left the 21-stage axial compressor, which is designed for 40,000 cfm at a delivery pressure of 45 psig and a speed of about 5100 rpm. The gear and the electric generator for the surplus power are not shown here, but may be seen in Fig. 22; at the extreme right is the starting motor for bringing the whole set up to speed, which can be uncoupled after the gas turbine takes over.

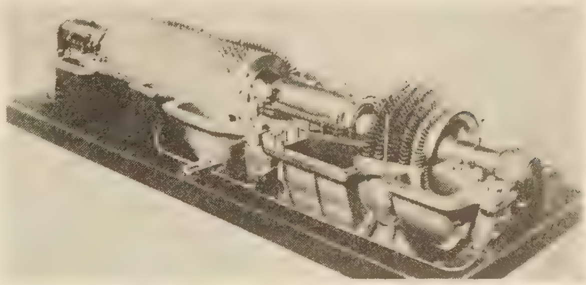

Fig. 21 Gas-Turbine-Driven Compressor Set for Houdry Process With Top Casing Removed

Beside the 4000-kw Neuchatel gas turbine and the 2200-hp unit powering the Swiss locomotive our company completed, during 1939 and 1940, four other self-contained gas turbines which went into operation after extensive tests. Two of them are working in blast-furnace plants, delivering the blast air and burning furnace gas. Unfortunately, it has not been possible to publish any performance data on these units due to wartime restrictions, but there is room for the hope that such information will be available before long.

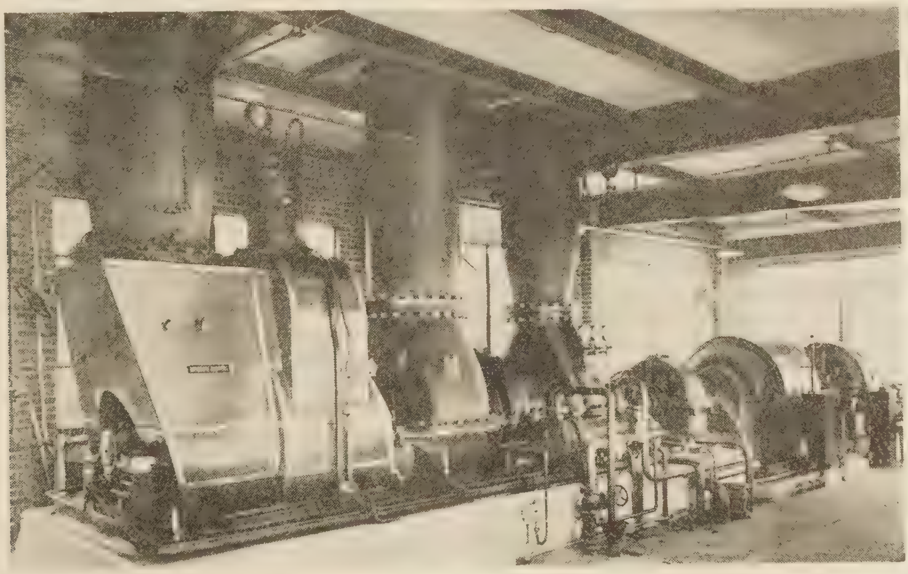

Fig. 22 First Gas-Turbine-Driven Compressor Set for Houdry Process, Installed in Eastern Refinery

In the field of superchargers, the author indicates that the first gear-driven machines for United States aviation engines were built in 1925. It might be noted in passing that several hundred gear-driven centrifugal superchargers, following Rateau or Brown Boveri design, were in use in Europe during the first world war.

As to exhaust-turbine-driven superchargers, the writer has noted the statement that no turbosuperchargers other than those of General Electric are in use in the present conflict. He does know, however, of rather extensive developmental programs started several years prior to the outbreak of this war with well-known airplane-engine builders in England and France, following the Brown Boveri - Büchi system. It would seem likely that these efforts have produced some results, although no supporting evidence is available due to censorship restrictions.

Under the heading "High-Temperature Progress," the author mentions some tests made in the Brown Boveri shops with an airplane supercharger running at 30,000 rpm and about 1800 F. It should be added that the diameter of the turbine wheel is in the range of 10–12 in. This is considerably smaller than that of turbosuperchargers used in this country and running at substantially lower speeds and temperatures.

After discussing some controversial points, the writer is very glad to conclude with a note of entire accord, namely, in regard to the important contributions of the author's work on turbosuperchargers, especially in the line of high-temperature operation, for the advancement of the combustion-gas turbine. This experience will be particularly interesting for checks and comparison with the results of research work that has been going on in other quarters, all directed toward the development of alloys which will stand up in gas turbines for continuous service with gas temperatures of 1500 to 1700 F.

By using these results in conjunction with the other known and developed elements of the gas-turbine cycle, particularly the efficient axial-flow compressor, we shall surely construct in the coming peace years gas-turbine power plants of high thermal efficiency which will find ready acceptance in many fields that are now being regarded as the exclusive domain of the steam turbine or the Diesel engine.

W. E. Trumpler.[14] In this presentation, attention has been called to the importance of heat-resisting materials, high compressor and turbine efficiencies. It is felt that not enough emphasis has been placed on the problem of blade cooling. While

[14] Development Engineer, Carrier Corporation, Syracuse, N. Y Mem. A.S.M.E.

remarkable progress has been made recently toward the development of heat-resisting materials, combustion-gas temperatures will always reach a higher level than materials can stand, and cooling may offer attractive solutions.

It is obvious that cooling in a gas turbine is of a different nature that in combustion engines. Cooling by air circulation must supersede the cooling-water jacket. Air cooling to be effective must use similarly high velocities as occur in the turbine-blade channel. To do this without involving additional losses, the cooling circulation must serve some other useful purpose, such as compression of the air into the combustion chamber. The losses then are only slightly higher than with compression performed independently. Lorenzen (26) is to the writer's knowledge the first to have used passages in the turbine wheel and through hollow blades to produce centrifugal compression and cooling combined. His arrangement suffered from hydraulic imperfection and failed to obtain practical success.

Following the same basic principle, the writer has devised an arrangement wherein the turbine wheel is equipped with specially shaped blades suitable for both compression and expansion and partially admitted to the flow of each function. The blade in question is bound to be a compromise design and the blade efficiency will suffer, also the change from one operation to the other twice during one revolution involves additional losses. Very complex hydraulic problems must be solved to keep these losses to a tolerable figure. It has been proved, however, even at this early stage of development, that the advantage gained by cooling the blades more than offsets the additional hydraulic and thermal losses. It is impossible to predict the extent of future improvements in this direction, but it is certain that blade cooling will have an economic justification in the gas turbine of the future.

G. B. Warren.[15] The development of the gas turbine has been the grand passion of the author's life. His not inconsiderable participation in the development of the centrifugal compressor, then of the steam turbine, and finally of the airplane turbo and geared centrifugal supercharger, has, if the writer judges him correctly, been but a stepping stone in his quest for the ultimate development of the gas turbine itself. It is altogether fitting and proper that as a result to an appreciable extent of these efforts, combined, to be sure, with the achievements now of many other men, the author can begin to see the real possibilities of the development of several successful gas turbines on the part of numerous groups. This new but old form of prime mover is destined to play a very important role in our industrial life.

Another important achievement, for which it is believed the author is more responsible than any other one man and regarding which he says nothing in his paper, but which came as a byproduct of his other work, was the development of the gear-driven centrifugal supercharger for airplane engines. This machine is now used on practically every military airplane engine in the world whether or not a turbosupercharger is used. A very large number of these have been produced from the designs of the division of the General Electric Company that the author organized.

It is, of course, unfortunate that the requirements of the national war effort limit the matters which the author could have discussed in this paper. Such will not always be so, and the sequel to this paper written by the author or others will make interesting reading when that time comes.

The very complete Bibliography will undoubtedly prove to be a particularly helpful tool to the many others now working in this field.

[15] Turbine Engineering Division, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

Author's Closure

The author feels highly complimented by the discussions which have been contributed and wishes to express his thanks and appreciation. A number of the discussions are by eminent authorities who have had actual experience with the subject so that they add a great deal to the author's presentation. The author also is thankful for a number of new references which may increase the bibliographical value of the paper. Some of these are given in the discussions, and a few have been communicated to the author and are added to the list of references.

As several of the discussers have mentioned, some of the items presented in the paper already are known to some extent. However, correspondence which the author has received indicates that even though some items may be known to some of the older generation, it is desirable to include them all for the sake of completeness and for the benefit of the younger generation. They are the ones who are to have the task of bringing the gas turbine to the successful fruition which is foretold by the present situation.

In spite of some belief to the contrary, the author feels that the ultimate gas turbine will successfully and conservatively operate at very high temperatures. Fig. 12 of the paper, and colored photographs that have been taken, of nearly white-hot nozzle boxes and red-hot turbine wheels, are visual evidence of the purpose of the portion of the paper devoted to turbosuperchargers, which is to show the enormous contributions that the thousands upon thousands of these turbosuperchargers now in use are making to the art of operating turbines safely at very high temperatures.

The author can add from personal knowledge, to the discussion kindly given by Mr. Haverstick of the De Laval Company. The Sherbondy apparatus that Mr. Haverstick mentions started out as a design sent from France by Rateau to his American agents, the Rateau-Battu-Smoot Company, and no doubt shown in Mr. Haverstick's Fig. 14. But there was great change from this design of Rateau in the course of the development work by Sherbondy and Fergus Motors. One of the changed models is given in the 1927 New York edition of Stodola (25),[16] as Mr. Haverstick states. But the accompanying diagram (Fig. 1086) is not, as stated by Stodola, the General Electric apparatus, but is a general diagram of the cycle, probably first presented by Rateau. The General Electric details always differed materially from Rateau or Sherbondy. Major George E. A. Hallet had charge during this period of the Powerplant Division of the United States Army Air Corps, McCook Field, Dayton. He wrote at the time ($44^1/_2$,) which gives further details of this period, "....... no way was seen to overcome the faults of the Sherbondy machine. Therefore, the latter was dropped and General Electric was given a contract to rebuild the old supercharger designed by Dr. Moss." Major Hallett recently gave similar details, which have been quoted by Frederick Neely.[17] The Rateau machine did operate in France and was given a test in England. But as the author and other General Electric engineers have learned the hard way, it is one thing to have apparatus that operates on paper or on test, and another to obtain actual service on many airplanes at high altitude. The "know-how" that Mr. Sidler of Brown Boveri appropriately mentions only is obtained through such experience.

There has been included the oft-repeated gas-turbine thermodynamics only by citation of references. But one thermodynamic item that probably is novel has been called to the author's attention by Mr. F. K. Fischer of the Westinghouse

[16] Bibliography (25), Fig. 1086a, p. 1252.

[17] "Conquest of the Stratosphere," by Frederick Neely, *Air Transport*, March, 1944.

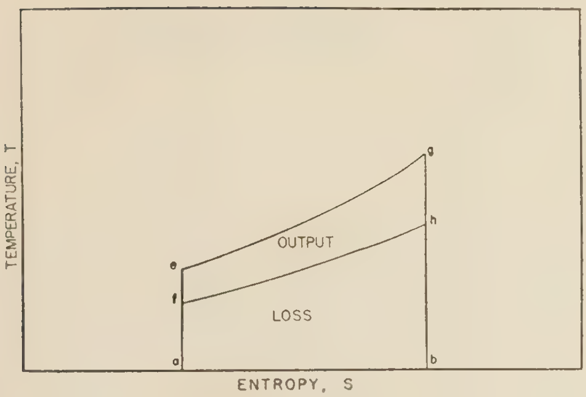

Fig. 23 Simple Gas-Turbine Cycle

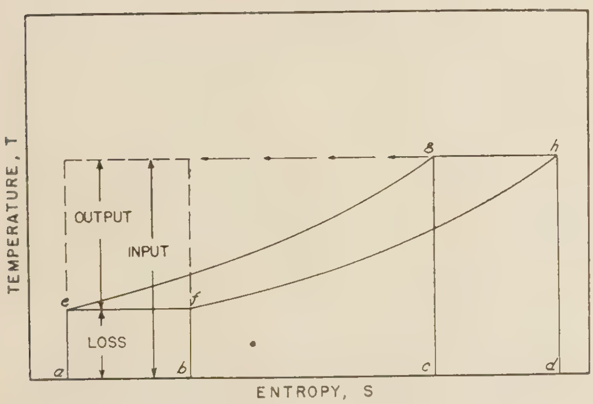

Fig. 24 Gas-Turbine Cycle With Complete Reheating, Regeneration, and Intercooling

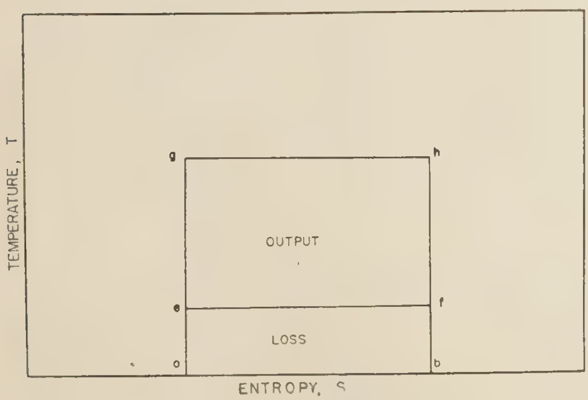

Fig. 25 Carnot Cycle

Company, Philadelphia, and Mr. Lee J. Fischer of the General Electric Supercharger Division, Lynn, Mass., and seems worthy of inclusion as follows:

Although the simple gas-turbine cycle, Fig. 23, has a lower efficiency than the Carnot cycle for given temperature limits, a plan with complete reheat, regeneration, and intercooling will give a theoretical cycle, Fig. 24, which has the full Carnot efficiency, which is the maximum of efficiency obtainable with given temperature limits Fig. 25. Dr. Rettaliata's Fig. 17 shows a well-known partial approach to this scheme.

The simple gas-turbine cycle, Fig. 23, consists ideally of an isentropic compression (f to e), a constant pressure heating (e to g), an isentropic expansion (g to h), and a rejection of the exhaust gases at constant pressure (h to f).

The improved cycle Fig. 24, has compression (f to e) in a compressor with sufficient intercooling to maintain practically constant temperature. The constant pressure heating (e to g) is completely accomplished through regeneration by a theoretically infinite heat exchanger, utilizing the heat of the exhaust gases (h to f, area b-f-h-d being essentially equal to area a-e-g-c). The gases are expanded (g to h), in a turbine having sufficient reheat to maintain constant temperature.

The only external heat added to the improved cycle is, therefore, that put in during the reheating process (area c-g-h-d). The heat rejected is that rejected to the compressor cooling medium (area a-e-f-b). The work done (area e-g-h-f), Fig. 24, practically is equivalent to a rectangular area of the same base and height. This assumes that expansion and compression through the same pressure ratio gives the same entropy increment, (which is the case with a perfect gas). Consequently, for the same temperatures, the efficiency of this improved cycle is the equal of the Carnot cycle, Fig. 25, in which the compression (e to g) practically is isentropic, the heating process (g to h) practically is isothermal, the expansion (h to f) is isentropic and the rejection of heat (f to e) also practically is isothermal.

In actual practice, therefore, if sufficient heat-exchanger surface and properly high efficiency of the various elements could be obtained, the gas-turbine cycle would be able to approach Carnot efficiency as a limit.

Of course, this only is a theoretical dream and there are a lot of limitations and practical difficulties to be overcome before its realization even can be approached. But it is inspiring to realize that there is such a theoretical possibility even though it may take a long time to approach it.

The author has had many years of pleasant association with Mr. Glenn B. Warren and must admit the statement made in his discussion, that the development of the gas turbine has been the "grand passion" of the author's engineering career. This career no doubt now is drawing to a close; but nevertheless the author expects to see many power-plant-gas turbines commercially developed by the younger generation.

Bursting Tests of Steam-Turbine Disk Wheels

By ERNEST L. ROBINSON,[1] SCHENECTADY, N. Y.

Bursting tests on model disks of steam-turbine-rotor material both with and without a central hole show that the average hoop stress at bursting speed for a good alloy steel is likely to exceed the tensile strength of the material as determined in an ordinary tensile test whether or not the hole is present. With a good turbine-wheel material, the maximum stress at bursting speed in a flat disk, calculated on the basis of elastic theory, greatly exceeds the tensile strength of the material; and if there is a central hole, it is more than double the tensile strength. Thus a comparison of tensile strength with average hoop stress in a wheel gives a fairly reasonable estimate of the factor of safety against actual bursting. Test results are also given on low-carbon and on high-alloy materials.

During the past 10 years and especially since steam temperatures have increased to the level where materials suffer a gradual deformation called creep, it has become quite common to construct turbine rotors out of solid forgings instead of assembling a succession of wheels on a shaft and expecting them to stay tight by virtue of their elastic grip on the shaft. Such rotors are often quite large, and it has been thought good practice to bore an inspection hole through the geometrical center in order to be sure of the soundness of the material. Under such conditions a calculation of the elastic-stress distribution in the immediate vicinity of the central hole shows the tangential stress to rise at the edge of the hole to double the value in the center of the rotor without any hole.

There is no question as to the existence of this elastic-stress concentration, but only a limited amount of material which depends on the size of the hole is subjected to the extra stress. Thus with a ductile material it seems likely that no ill effects will be experienced in having such a local stress exceed ordinary rules of design. Thus it may be reasonable to be guided by the bore stress in a shrunk-on wheel, but it does not seem equally reasonable to count the local stress at the boundary of an inspection hole. Of course, this stress can be avoided by not boring such a hole. However, such a decision would tend to place too much reliance on the attainment of a perfect calculation and would blindly assume perfection of the material. It has therefore been deemed good practice to bore the hole and not count the local stress concentration.

The several series of tests described herewith were originally undertaken to make sure that this judgment was well founded, and this turned out to be the case.

Heat-Treated Nickel-Chrome-Molybdenum Steel, S.A.E. 4340

Chemical Composition: (Per cent) C—0.39, Ni—1.81, Cr—0.74, Mo—0.38, V—0.09, Mn—0.57, Si—0.18

Heat-Treatment: 950 C (1742 F) 8 hr furnace cool, 100 C (180 F) per hr, 650 C (1202 F) 4 hr furnace cool

The first series of wheels consisted of four flat disks 11 in. diam and $^9/_{16}$ in. thick, two without any center hole and two with a center hole 1.5 in. in diam. Fig. 1 shows the elastic-stress distribution in these wheels. It may be noted on the right-hand scale that the maximum stress at the center (without any hole), as given by the ordinary elastic theory, is 10,000 psi at 10,000 rpm, while the average bursting stress is 80 per cent of the maximum, or 8000 psi. The diagram also gives at the left the stresses at 40,000 rpm, the stress being greater in proportion to the square of the speed.

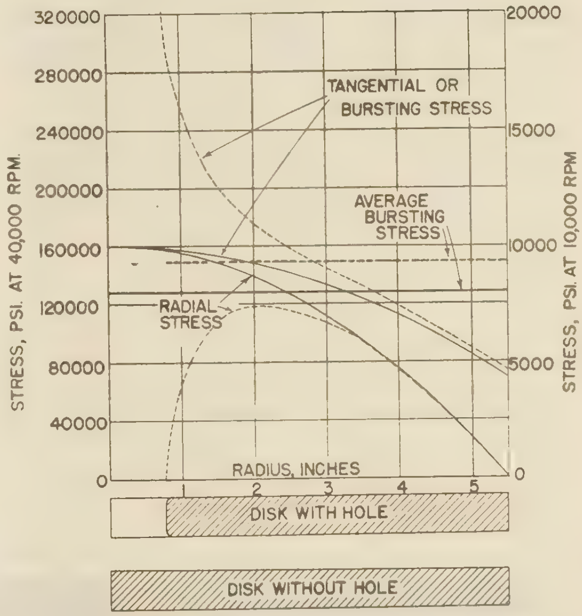

Fig. 1 Elastic Stress Distribution in Rotating Disks With and Without a Central Hole

(The solid lines represent the stress distribution in the solid disk and the dotted lines the disk with hole.)

Table 1 gives the physical properties and the results of the bursting tests. Regardless of whether or not the central hole was present, these wheels burst with average stresses ranging from 121,600 psi to 131,000 psi, or 98.9 to 106.5 per cent of the tensile strength. The presence of the central hole thus reduced the actual bursting strength just about in proportion to the amount of material cut out.

The maximum center or bore stresses which would have existed at bursting speed if the wheels had remained elastic amounted to 130.1 and 131.3 per cent of tensile strength for the solid wheels and 229.3 and 213.0 per cent of tensile strength for the wheels with holes. Thus with this typical steam-turbine-rotor material the average bursting stress is an excellent measure of ultimate strength.

The measured deformation also given in Table 1 may or may not be significant. WB4 with center hole enlarged almost 4

[1] Turbine Engineering Division, General Electric Company. Mem. A.S.M.E.

Contributed by the Applied Mechanics and Power Divisions and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

TABLE 1 PROPERTIES AND RESULTS OF BURSTING TESTS ON WHEELS OF HEAT-TREATED NICKEL-CHROME-MOLY STEEL, S.A.E. 4340

PHYSICAL PROPERTIES

Tensile strength, 123,000 psi — Elongation, 15.0 per cent
Elastic limit, 80,400 psi — Reduction in area, 39.1 per cent

STRESSES AT BURSTING SPEED

Disk no.	Speed, rpm	Average stress, psi	Per cent of T.S.	Elastic stress, psi	Per cent of T.S.
	11-in-diam flat wheels without hole				
WB1	40000	129000	104.9	160000	130.1
WB3	40200	130400	106.0	161500	131.3
	11-in-diam flat wheels with 1.5-in. center hole				
WB2	37500	131000	106.5	282000	229.3
WB4	36120	121600	98.9	262000	213.0

DEFORMATION IN DISK, PER CENT

Disk no.	Rim diam	Rim thickness	Center thickness	Bore diam	Bore thickness
	11-in-diam flat wheels without hole				
WB1	0.91[a]	—2.67	—10.9	...	...
WB3	2.28	—2.67	—16.3	...	...
	11-in-diam flat wheels with 1.5-in. center hole				
WB2	2.0	—2.13		31.3	—11.4
WB4	3.72	—1.07		32.8	—10.7

[a] Based on diameter after last run before bursting.

per cent on the diameter; whereas, the solid wheel WB1 had enlarged less than 1 per cent on the last run before bursting. This wheel went into very small fragments, the largest "piece of pie" being just about $^1/_8$ of the whole. Three of these fragments had radii smaller than the original wheel as if bursting had released a star- or spider-shaped void in the center and allowed the rim to go scallop-shaped. Undoubtedly it is necessary to take note of such behavior in order to understand the results and assign proper significance to them.

original 11 in. may be seen plainly on the foot rule. The thinning toward the center is also perceptible below the straight edge of the rule. The white paper sticking out of the center hole shows how much it has enlarged over the original 1.5 in.

LOW-CARBON STEEL, S.A.E. 1020

Chemical Composition: (Per cent) C—0.15 to 0.25, Mn—0.30 to 0.60

Heat-Treatment: 850 C (1562 F) oil quench, 650 C (1202 F) air cool

The second series of wheels consisted of four flat disks just like those of the first series. In addition, there were two more disks made with the bottle-shaped cross section necessary to give a uniform distribution of stress over the entire interior of the wheel as shown in Fig. 3. These two wheels were designed to get the maximum peripheral speed out of a given material. They are directly comparable with the flat wheels of 11-in. diam, since the elastic stress at 10,000 rpm is also 10,000 psi. This stress, as with the flat wheels, would rise to 160,000 psi at 40,000 rpm. However, whereas the stresses in the flat wheel fall off from the maximum at the center to much lower values toward the outside, both the radial and tangential stresses remain equal and of constant magnitude from the center to the inside of the small rim. In the rim there is a slight falling off of the hoop stress, and within the small increase of dimension the radial stress drops rapidly from its maximum value to zero at the outside.

Thus these six wheels of the same material all have strictly

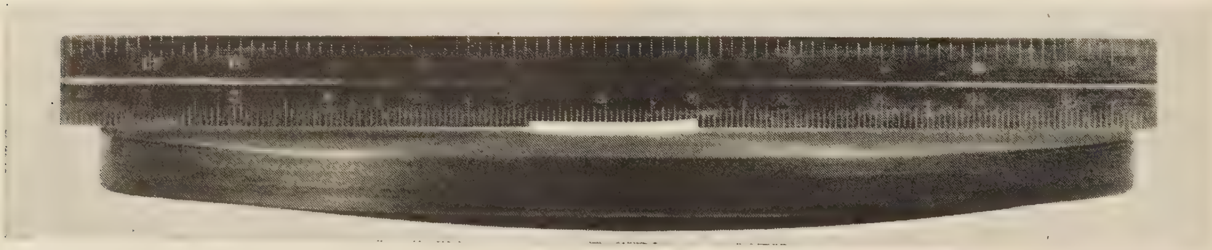

FIG. 2 NICKEL-CHROME-MOLYBDENUM STEEL DISK, WB4, 11 IN. DIAM AFTER RUN AT 36,000 RPM
(Enlargement at rim, 2.2 per cent; at hole, 21.2 per cent; thinning at rim, 1 per cent; at hole, 9.7 per cent.)

FIG. 3 LOW-CARBON-STEEL DISK OF UNIFORM STRESS
(The small rectangular rim takes the place of a bucket loading and is the only portion of the wheel in which the uniform stress of 10,000 psi at 10,000 rpm is not maintained. The thickness in inches, t, of this wheel at any radius r in inches is given by the formula $t = 0.670\ e^{-0.04r^2}$ where 0.670 in. is the center thickness.)

The solid wheel WB3 thinned 16.3 per cent at the center, an amount in excess of the elongation in the tensile test. The other wheels thinned about $^2/_3$ as much. The center holes enlarged approximately one third from 1.5 in. diam to about 2 in. diam.

Fig. 2 shows one of these disks, WB4, with the central hole, after it had been run to 36,000 rpm, very close below the bursting speed of 36,120 rpm. The enlargement of diameter over the

comparable elastic-stress distributions at the same speed. In the two solid wheels the maximum elastic stresses are the same, but localized around the center of the flat wheel and of wide extent in the bottle-shaped wheel. In the flat wheel with the central hole the elastic-stress distribution in the outer half radius or $^3/_4$ of the volume is very close to that in the flat wheel without any hole. However, in the inner half radius, comprising less than $^1/_4$ of the volume of the wheel, the elastic stress rises

TABLE 2 PROPERTIES AND RESULTS OF BURSTING TESTS ON WHEELS OF LOW-CARBON STEEL, SAE 1020

PHYSICAL PROPERTIES

Tensile strength, 68,650 psi — Elongation, 37.5 per cent
Elastic limit, 47,150 psi — Reduction in area, 60 per cent

STRESSES AT BURSTING SPEED

Disk no.	Speed, rpm	Average stress, psi	Per cent of T.S.	Elastic stress, psi	Per cent of T.S.
	11-in-diam flat wheels without hole				
WB5	29000	67800	98.8	84100	122.5
WB7	29400	69700	101.6	86500	126.0
	11-in-diam flat wheels with 1.5-in. center hole				
WB6	24300	55000	80.1	118400	172.5
WB8	24400	55500	80.8	119400	174.0
	16.6-in-diam wheels of uniform stress (no hole)				
WB9	25500	65000	94.7	65000	94.7
WB10	25200	63500	92.5	63500	92.5

DEFORMATION IN DISK, PER CENT

Disk no.	Rim diam	Rim thickness	Center thickness	Bore diam	Bore thickness
	11-in-diam flat wheels without hole				
WB5	9.09	—4.97	—38.4	...	...
WB7	7.37	—4.80	—38.4	...	...
	11-in-diam flat wheels with 1.5-in. center hole				
WB6	3.91	—3.2	...	46.0	—19.5
WB8	3.84	—3.55	...	42.8	—18.2
	16.6-in-diam wheels of uniform stress (no hole)				
WB9	9.48[a]	—6.17	—18.9	...	...
WB10	2.83[a]	—8.63	—26.1	...	...

[a] Based on diameter after last run before bursting.

to a maximum value at the bore which is twice what it is in the other wheels. (Precisely speaking, this ratio is nearer 201 per cent than 200 per cent.)

This material was chosen as representative of very ductile materials, and it was heat-treated for maximum ductility. In examining the results, it is necessary to keep these facts in mind. The tensile test shows a real "yield point" where the material refuses to hold until it has stretched a considerable amount. After this stretching has occurred, it holds again, but this kind of ductility has its disadvantages.

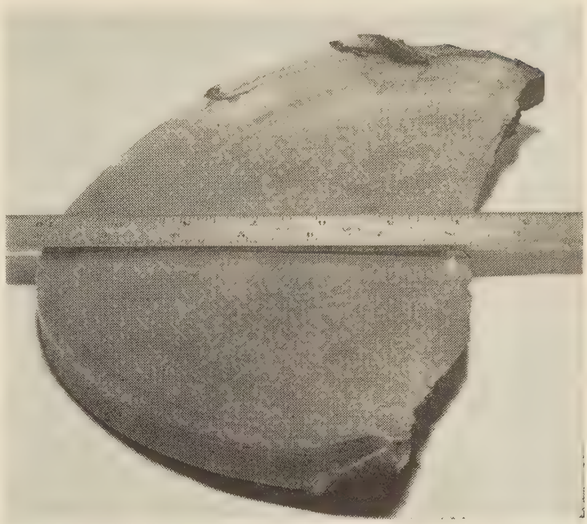

FIG. 5 LOW-CARBON-STEEL DISK WB5, 11 IN. DIAM AFTER BURSTING AT 29,000 RPM

(Enlargement at rim 9.1 per cent; thinning at rim, 5 per cent; at center, 38.4 per cent. Note that this is just about a full half wheel. The rupture is close to perpendicular to the plane of the disk.)

much as it got longer which is like a tensile bar. The wheel as a whole got thinner at the center, 38.4 per cent in each case, as compared to 37.5 per cent elongation in the tensile test. In each case this is the deformation in the direction of the axis of deformation, although the nature of the deformation is completely opposite. In the tensile test, the bar is pulled along its axis and goes in from all sides. The bursting disk is pulled out in all directions and goes in along its axis.

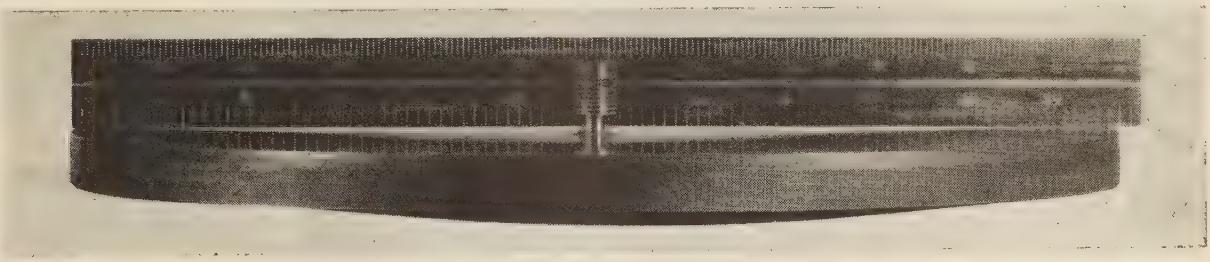

FIG. 4 LOW-CARBON-STEEL DISK WB7, 11 IN. DIAM AFTER RUN AT 28,320 RPM

(Enlargement at rim, 5.5 per cent; thinning at rim, 2.7 per cent; at center, 9.7 per cent.)

Table 2 gives the physical properties and the results of the bursting tests. The two solid wheels, WB5 and WB7, burst with average stresses based on the original dimensions of 98.8 per cent and 101.6 per cent of the tensile strength. However, the wheels enlarged 9.09 per cent and 7.37 per cent, respectively, with corresponding increases in the actual centrifugal bursting forces. Thus, going by the enlarged diameters, these wheels actually had average stresses at bursting of approximately 108 per cent of tensile strength. A pretty good case could be made out for basing the actual stress on the square of the enlarged diameter; but, since the centrifugal force itself goes only with the first power of the radius, such a procedure would correspond with the occasional practice of computing tensile strength on the basis of the reduced section instead of on the original dimensions. Analysis in terms of the original dimensions has the advantage that such dimensions are known in advance.

Note that the circumference got thinner about one half as

The two wheels with holes burst at too low speeds to develop average stresses equal to the tensile strength even if credit is allowed for the enlargement at the rim. (This has not been done in the table where all calculations are based on original dimensions.)

The rims got thinner almost as much as they got larger, but the center holes enlarged more than 40 per cent and only thinned less than 20 per cent.

The enlargement of these wheels was less than one half as much as the solid wheels. Apparently this material has a kind of ductility that does not allow all portions of these wheels to participate in the job of resisting the centrifugal force. The burst takes place at a time when the rim is not able to do its share in carrying the load or at least has not yet picked up its full share.

Fig. 4 illustrates WB7 after having been run at 28,320 rpm which is a little more than 1000 rpm below its bursting speed of 29,400 rpm. The foot rule in this picture shows plainly the

Fig. 6 Strain Lines Spreading Outward From Edge of Hole As They Appeared on Surface of Low-Carbon Steel Disk WB8, 11 In. Diam, After Run at 20,100 Rpm

Fig. 7 Strain Lines Reaching to Rim of Disk WB8, After Run at 21,300 Rpm

Fig. 8 Strain Lines Disappearing Off Rim of Low-Carbon Steel Disk WB6, 11 In. Diam, After Run at 22,380 Rpm

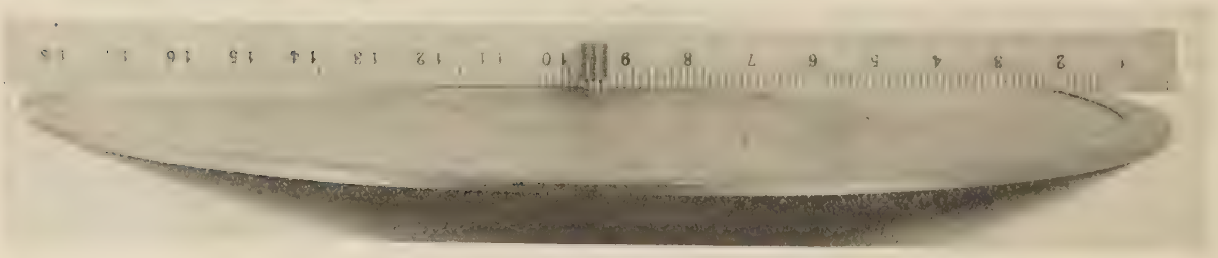

Fig. 9 Low-Carbon Steel Disk WB9, 16.606 In. Diam, Proportioned for Uniform Stress Distribution, After Run at 25,200 Rpm
(Enlargement at rim, 9.5 per cent.)

enlargement of the diameter of the disk and its thinning toward the center.

Fig. 5 shows the fracture in the case of disk WB5. This appears to be a rather characteristic fracture. It seems necessary to conclude that, however instantaneous the burst, the fracture started at the center and finished up at the outside. There appears to be a tendency for these wheels in the final stages of enlargement to try to neck down on two perpendicular diameters, thus ever so slightly trying to make a square out of a circle.

Strain Lines. Fig. 6 shows the appearance of strain lines on the surface of disk WB8 after having been run to 20,100 rpm. The markings originate at the bore which is the first part of the wheel to attain a stress in excess of the yield point of the material. Fig. 7 shows the appearance of the same disk after having been run to 21,300 rpm. The markings have now reached the periphery of the disk while the region around the central hole has cleared up. Fig. 8 shows the appearance of the other identical disk WB6 after having been run to 22,380 rpm. The entire inner quarter of the surface out to the half radius of the disk has now cleared up.

Uniformly Stressed Wheels. The two wheels of uniform stress developed slightly less than full tensile strength on the basis of their original dimensions. However, it should be noted that WB9, shown in Fig. 9, enlarged from $16^5/_8$ to $18^3/_{16}$ in. diam or almost 10 per cent without bursting. Based on the actual centrifugal force with these enlarged dimensions, this wheel developed at least 105 per cent of tensile strength when it burst. While the flat wheels appear to have all burst from the center, these wheels of uniform stress both lost their rims, and neither gave way anywhere throughout the whole central web.

Peripheral Speeds. It is noteworthy that WB3 which was just a flat wheel with a tensile strength of 123,000 psi attained a peripheral speed of 1975 fps at bursting, and that WB9 with the theoretical shape and a tensile strength of only 68,650 psi attained a peripheral speed of 2025 fps at bursting. (In each case credit is here taken for the enlargement of diameter.) These figures give some indication as to the real value of design shape. Thus in this comparison the special contour of the wheel of uniform stress enabled a material 56 per cent as strong to attain fully as high a speed, or even a trifle higher.

Cyclops 17W

Chemical Composition: (Per cent). C—0.49, Mn—0.68, Si—0.60, Cr—13.30, S—0.012, P—0.020, Ni—19.31, Mo—0.68, W—2.30 (Allegheny Heat 34413).

Heat-Treatment: Preheat 5 hr at 1500 F; transfer to 2100 F and hold for 1 hr; forge to within 10 per cent of finished forging thickness; cold work at 1200 F to finished size; anneal 2 hr at 1200 F.

The third series of tests consisted of four 10-in-diam wheels of Cyclops 17W material $^7/_{16}$ in. thick, two without center holes and two with center holes 1.364 in. diam. This size of center hole bears the same relation to the over-all diameter as the 1.5-in-diam hole in the 11-in-diam disk. Thus all stresses are comparative with the first two series at a speed ratio of 11/10. In other words, the stresses in the 10-in. disks when running at 10 per cent higher speed are the same as in the 11-in. disks.

The results of the bursting tests are given in Table 3. The two solid wheels burst with average stresses of 77.4 and 81.3 per cent of the tensile strength. If perfectly elastic at bursting, the maximum stresses would have been 95.7 per cent and 100.6 per cent of tensile strength.

The two wheels with holes in them burst with average stresses 79.7 per cent and 81.2 per cent of the tensile strength. If they had remained elastic, the maximum stresses would have been 171.3 and 174.8 per cent of the tensile strength.

It is a noteworthy happenstance that the two wheels without

TABLE 3 PROPERTIES AND RESULTS OF BURSTING TESTS ON WHEELS OF CYCLOPS 17W MATERIAL

Physical Properties

(Standard $^1/_4$-in-diam test specimen from rim)

Disk no.	Tensile strength, psi	Proportional limit, psi	Elongation, per cent	Reduction of area, per cent	Brinell No. Rim	Brinell No. Center
W1.......	131200	73400	18.0	22.6	255	269
W2.......	139500	89600	15.0	26.6	285	269
W3.......	135500	89600	15.0	26.6	277	255
W4.......	131200	77400	17.0	26.0	269	255

Stresses at Bursting Speed

Disk no.	Speed, rpm	Average stress, psi	Per cent of T.S.	Elastic stress, psi	Per cent of T.S.
	10-in-diam flat wheels without hole				
W1............	39000	101500	77.4	125500	95.7
W4............	40000	106700	81.3	132000	100.6
	10-in-diam flat wheels with 1.364-in-diam center hole				
W2............	38000	111200	79.7	239000	171.3
W3............	37800	110000	81.2	237000	174.8

Deformation in Disk, Per Cent

Disk no.	Rim diam	Rim thickness	Center thickness	Bore diam	Bore thickness
	10-in-diam flat wheels without hole				
W1............	2.65	—0.16	—6.25	...	...
W4............	0.99	—0.80	—5.97	...	...
	10-in-diam flat wheels with 1.364-in-diam center hole				
W2............	1.20	—0.07	...	6.84	—2.76
W3............	1.56	—0.16	...	5.09	—2.38

holes in them burst when the maximum elastic stress reached 100 per cent of the tensile strength. This is not a reliable criterion for bursting, however, because the maximum elastic stress in the wheels with holes reached almost 75 per cent over the tensile strength.

For these wheels the best rule for bursting would be to say that they would develop an average stress equal to 80 per cent of the tensile strength, and this would be approximately true whether or not there is a center hole.

Gamma Columbium

Chemical Composition: (Per cent) C—0.34, Mn—0.45, Si—0.45, Cr—14.74, Si—0.010, P—0.013, Ni—25.06, Mo—4.32, Cb—1.88 (Allegheny Heat 34658)

Heat-Treatment: Same as for Cyclops 17W

The fourth series of tests consisted of four 10-in-diam wheels of "gamma columbium" material, $^7/_{16}$-in. thick, two without center holes and two with center holes, 1.364 in. diam. These four wheels were the same size as the third series.

The results of the bursting tests are given in Table 4. The

TABLE 4 PROPERTIES AND RESULTS OF BURSTING TESTS ON WHEELS OF GAMMA COLUMBIUM MATERIAL

Physical Properties

(Standard $^1/_4$-in-diam test specimen from rim)

Disk no.	Tensile strength, psi	Proportional limit, psi	Elongation, per cent	Reduction of area, per cent	Brinell no. Rim	Brinell no. Center
G1.......	130200	81500	18.0	32.7	255	241
G2.......	127000	81500	18.0	25.5	255	255
G3.......	128200	79500	15.0	23.2	255	255
G4.......	130200	85500	18.0	31.4	262	248

Stresses at Bursting Speed

Disk no.	Speed, rpm	Average stress, psi	Per cent of T.S.	Elastic stress, psi	Per cent of T.S.
	10-in-diam flat wheels without hole				
G2............	39600	104700	82.8	129500	102.0
G3............	40000	106700	83.2	132000	103.0
	10-in-diam flat wheels with 1.364-in-diam center hole				
G1............	36600	103000	79.2	222000	169.8
G4............	37200	106500	81.8	229000	175.8

Deformation in Disk, Per Cent

Disk no.	Rim diam	Rim thickness	Center thickness	Bore diam	Bore thickness
	10-in-diam flat wheels without hole				
G2............	1.07	—0.78	—5.74	...	...
G3............	2.45	—0.95	—7.75	...	..
	10-in-diam flat wheels with 1.364-in-diam center hole				
G1............	1.51	—0.66	...	5.46	—4.42
G4............	1.44	—0.62	...	7.58	—4.00

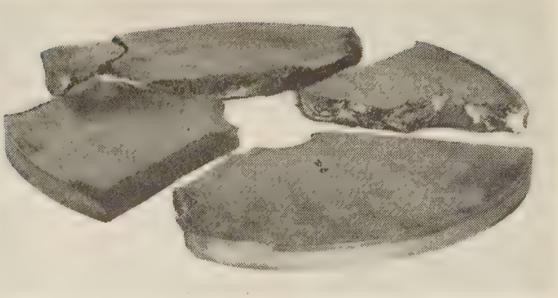

FIG. 10 GAMMA COLUMBIUM DISK G4, 10 IN. DIAM, WITH 1.364-IN-DIAM HOLE AFTER BURSTING AT 37,200 RPM

(Note that the wheel has quartered itself. The rupture in the foreground approximates 45 deg to plane of disk, except near the center hole.)

two solid wheels burst with average stresses 82.8 per cent and 83.2 per cent of the tensile strength. Had they remained elastic, the maximum stresses at bursting would have been 102.0 per cent and 103.0 per cent.

The two wheels with holes in them burst with average stresses 79.2 per cent and 81.8 per cent of tensile strength. If they had remained elastic, they would have burst with maximum stresses of 169.8 per cent and 175.8 per cent of tensile strength.

The behavior of these wheels was remarkably similar to the behavior of those in the third series, and the same remarks might be made about them.

Fig. 10 shows typical fragments, illustrating the manner in which the wheels of the three high-alloy series burst. The sequence of the several types of rupture is not so clear as for the single type illustrated in Fig. 5. Certain interesting speculations might be indulged in as to what type of stress or strain reached a limiting value and constituted itself the trigger to release the burst. It seems certain that a variety of limitations can be observed among the various types of break so that the most useful conclusion is that no one theory of failure can be relied on to represent all conditions of stress or strain.

TIMKEN "25-16-6"

Chemical Composition: (Per cent) C—0.15, Mn—1.14, P—0.010, S—0.022, Si—0.84, Cr—16.75, Ni—25.26, Mo—6.29, N_2—0.175 (Heat No. 0213)

Heat-Treatment: Same as for Cyclops 17W

The fifth series of tests consisted of four wheels of Timken 25-16-6 material, $^7/_{16}$ in. thick, all 10 in. diam, except T1 which was 8.517 in. diam. As in the third and fourth series, two wheels were solid and two had center holes 1.364 in. diam.

The results of the bursting test are given in Table 5. The two solid wheels burst with average stresses 71.6 per cent and 61.5 per cent of tensile strength. If they had remained elastic, the stresses at bursting speed would have been 88.6 per cent and 76.2 per cent of tensile strength.

The two wheels with holes in them burst with average stresses 75.2 per cent and 72.5 per cent of tensile strength. If they had remained elastic, the stresses at bursting would have been 156.9 per cent and 156.4 per cent of tensile strength.

The behavior of these wheels differed somewhat from that of all the other series. The average stresses at bursting speed were lower relative to the tensile strength than for any of the other materials, being 71.6 and 75.2 per cent, respectively, except for T4 which developed only 61.5 per cent.

It is noteworthy that the wheels with center holes actually ran to practically the same bursting speeds as the solid wheels. The average stress at bursting was actually a higher percentage of the tensile strength in the wheels with holes than in the solid wheels.

TABLE 5 PROPERTIES AND RESULTS OF BURSTING TESTS ON WHEELS OF TIMKEN "25-16-6" MATERIAL

PHYSICAL PROPERTIES

(Standard $^1/_4$-in-diam test specimen from rim)

Disk no.	Tensile strength, psi	Proportional limit, psi	Elongation, per cent	Reduction of area, per cent	Brinell No. Rim	Brinell No. Center
T1........	134500	87600	17.0	22.6	277	255
T2........	137500	95600	17.0	16.3	285	269
T3........	134500	93900	11.0	9.4	277	255
T4........	140500	93900	16.0	22.6	285	269

STRESSES AT BURSTING SPEED

Disk no.	Speed, rpm	Average stress, psi	Per cent of T.S.	Elastic stress, psi	Per cen of T.S.
	10-in-diam flat wheels without hole				
T3.............	38000	96300	71.6	119200	88.6
T4.............	36000	86400	61.5	107000	76.2
	10-in-diam flat wheels with 1.364-in-diam center hole				
T1[a]............	36200	101100	75.2	212000	156.9
T2.............	36000	99700	72.5	215000	156.4

DEFORMATION IN DISK, PER CENT

Disk no.	Rim diam	Rim thickness	Center thickness	Bore diam	Bore thickness
	10-in-diam flat wheels without hole				
T3.............	1.48	0.00	—2.98	...	...
T4.............	1.90	—0.23	—2.26	...	...
	10-in-diam flat wheels with 1.364-in-diam center hole				
T1[a]............	4.76	0.57	...	22.8	—1.60
T2.............	0.71	0.23	...	1.87	—0.74

[a] Wheel T1 was actually only 8.517 in. diam and burst at 42,000 rpm; 36,200 rpm would give the same average stress with a 10-in-diam wheel. The elastic stress is based on the actual original dimensions.

THE HIGH-ALLOY MATERIALS

As a lot, the materials of the third, fourth, and fifth series differ in certain characteristics of performance from those of the first two series. They are like the first two series in that all the bursts on flat wheels appear to originate at the center where both the elastic stress and the measured strain are maximum. However, the three high alloys show a change of center thickness or bore diameter around one third of the percentage elongation in the tensile test while the carbon-steel and nickel-chrome-molybdenum wheels show corresponding deformations equal to or greater than those attained in the tensile test.

If there is a quality that might be called "biaxial ductility," the wheels of the first two series might be said to possess it while those of the last three series are not so ductile under biaxial stress regardless of the tensile-test results.

However, in spite of this condition, these high-alloy materials develop average bursting stresses of about the same level relative to their tensile strengths as the very ductile carbon steel. But none of them equals the nickel-chrome-molybdenum material in developing an average bursting stress equal to or greater than the tensile strength.

The nickel-chrome-molybdenum steel has both biaxial ductility and a continuously increasing strength with deformation. It has no region of weakness such as the carbon steel displays when "the beam drops" in the tensile test.

TABLE 6 PERCENTAGE ELONGATION IN TENSILE TEST AND SPEED REDUCTION DUE TO CENTER HOLE OF VARIOUS MATERIALS

	Elongation in tensile test, per cent	Speed reduction due to center hole, per cent
Low-carbon steel..................	37.5	16 to 17
Ni-Cr-Mo steel....................	15.0	6 to 10
Gamma Columbium..............	15 to 18	6 to 8
Cyclops 17W......................	15 to 18	3 to 5
Timken 25-16-6..................	11 to 17	0 to 5

The lower ductility of the high alloys would suggest that these materials might be more sensitive to the elastic-stress concentration near the center hole shown in Fig. 1. However, the results, collected in Table 6, appear to show these materials to be even less sensitive than the more ductile materials.

The actual bursting speeds of the various wheels were influenced by some variation in tensile strength for which it would be proper to make some correction. However, it is impossible to escape the conclusion that the most ductile material at the top of the list, the carbon steel, loses most due to the hole, while the least ductile material at the bottom of the list is very little affected by the center hole.

It is to be noted that the two carbon-steel wheels with center holes did not enlarge so much in over-all diameter as did the solid wheels with no center hole, although the center holes in the low-carbon steel had the greatest percentage enlargements measured in any of the wheels. It may be that the rims had not yet stretched enough to be carrying their fair share of load while the holes were working at the limit of what the material was good for.

At the bottom of Table 6, the high-alloy materials may have had the elastic-stress concentrations to some degree neutralized by the cold work during heat-treatment. Thus these disks may initially have been in a state of internal compression with the rim in tension like the tire on a wagon wheel. This would give the outer portions of the disk a head start toward carrying a fair share of the bursting load in contrast to the handicap in the case of the rims of the low-carbon steel disks.

These tests do not include any comparisons with small and highly localized concentrations such as would be caused by keyways, bolt holes, pinholes, or balance holes. It is, however, evident that the manufacture of a high-speed wheel is a complex art which involves much more than any mathematical analysis of stress distribution. Furthermore, these tests do not reflect any evaluation of the high-temperature qualities of the various materials. That would be quite another story.

Factors of Safety and Criterions of Failure: It would seem that a series of tests to complete destruction should offer some useful data relative to factors of safety and criterions of failure. In considering these questions, it is necessary to decide on a point of view as to what constitutes a failure.

It is no doubt legitimate to call a bad case of malfunctioning due to unbalance, with consequent vibration, a failure. Similarly, a gradual loss of clearances with resulting rubbing, injury to packings, and consequent deterioration of efficiency might be called failure. But failure is a rather strong word; and if easily repairable malfunctioning is to be called failure, what word is left for destructive rupture?

Many articles are written in which the incidence of any plastic distortion with consequent permanent set, however small, is called failure. It would be more accurate to say that elastic theory has failed to apply than that any real failure has occurred in the machine.

However, it is certainly true that there are few useful structures either among machines or buildings that do not exceed the elastic limit in many regions of local overstress. Certainly this does not constitute failure. Indeed, its occurrence is usually very difficult to detect.

On the other hand, there is usually no doubt about the complete destruction accompanying actual rupture. Largely for this reason, but also because it seems a fundamentally sound viewpoint, the wheel bursts here reported have been discussed in terms of bursting speed rather than in terms of the speed or local maximum stress at which some measurable permanent set was first detected.

Certainly elastic stresses should be computed and design proportions should take account of them, but when the question of a true ultimate factor of safety is concerned, it would seem better judgment to compare operating conditions with actual rupture rather than with some half-way point of less critical significance.

Conclusions

The fact that the average bursting stress frequently equals or exceeds the tensile strength throughout the first series of tests shows that in some locations there must be maximum stresses considerably in excess of the tensile strength, since it is hard to believe that the leveling effect of the plastic yielding can have completely reduced all stresses to the average value. Thus it is necessary to conclude at least for the nickel-chrome-molybdenum steel, that the biaxial-stress distribution has some strengthening value which would tend to justify the maximum-strain theory of ultimate failure. In other words, looking at the center of the solid wheels, the presence of two equal stresses at right angles to each other enables each to attain a higher value than it could if deformation were not in part restrained by the presence of the other stress.

It might be possible to study the redistribution of stress during the plastic enlargement of the disk by noting the change of thickness along the radius. However, the fact that carefully duplicated bars show widely varying plastic behavior under stress suggests that such a detailed analysis would not be fruitful of any useful information.

The most important item of information given by these tests is that with ductile materials such as here tested the real factor of safety against bursting may be more closely estimated by comparing the average bursting stress with the ordinary tensile strength than by trying to go by the maximum elastic stress. Thus, the weakening effect of an inspection hole is nowhere near the 50 per cent that elastic theory suggests. In fact, with the type of material most likely to be used, the weakening effect is no more than in proportion to the material removed.

Acknowledgments

The author wishes to acknowledge the generous co-operation of K. D. McMahan of the General Electric Research Laboratory in Schenectady, E. E. Stoeckly of the Supercharger Engineering Division, and W. L. Badger, Thomson Laboratory, both of River Works of the General Electric Company, West Lynn, Mass. together with their respective associates.

Appendix

The formulas for elastic stress distribution in flat disks with and without central hole are collected herewith for convenient reference. Note that the centrifugal force per cubic inch of steel is

$$8N^2r$$

where r is the radius in inches and N the speed of rotation in thousands of revolutions per minute. (The even figure 8 happens to be correct for the density of steel. For other metals, this figure should be multiplied by the density relative to steel.)

Elastic-Stress Distribution in a Flat Steel Disk

The elastic distribution of the stresses in a flat disk due to centrifugal force is described by the following formulas:

Hoop stress at any radius r in a solid flat disk of diameter $2R$ is

$$N^2[(3 + n)R^2 - (3n + 1)r^2]$$

where n is Poisson's ratio.

The radial stress is

$$N^2[(3 + n)(R^2 - r^2)]$$

The maximum value of each occurs at the center and is

$$N^2[(3 + n)R^2]$$

Hoop stress at any radius r in a flat disk of diameter $2R_2$ with a central hole of diameter $2R_1$ is

$$N^2\left[(3+n)\left(R_2^2+R_1^2+\frac{R_2^2R_1^2}{r^2}\right)-(3n+1)r^2\right]$$

The maximum value occurs at $r=R_1$

$$2N^2[(3+n)R_2^2+(1-n)R_1^2]$$

The radial stress is

$$N^2\left[(3+n)\left(R_2^2+R_1^2-\frac{R_2^2R_1^2}{r^2}-r^2\right)\right]$$

Average Bursting Stress in a Flat Steel Disk

Regardless of the stress distribution throughout the volume of the disk, the average bursting stress may be computed by dividing the total centrifugal bursting force by the total area of cross section.

The average bursting stress in a flat disk of diameter $2R_2$ with a central hole of diameter $2R_1$ is

$$\frac{8}{3}N^2\frac{R_2^3-R_1^3}{R_2-R_1}$$

If there is no central hole the average bursting stress in a flat disk of diameter R is

$$\frac{8}{3}N^2R^2$$

The ratio of the average bursting stress in the solid flat disk to the maximum stress at the center under elastic conditions is

$$\frac{8}{3}N^2R^2/N^2[(3+n)R^2]=\frac{8}{3(3+n)}$$

For various values of n this ratio is as follows:

n	$\frac{8}{3(3+n)}$
0.25	0.820
0.30	0.808
0.33	0.800
0.40	0.784
0.50	0.762

Computation of Centrifugal Bursting Stress for Special Steel Shapes

In any particular wheel, the average centrifugal bursting stress may be computed as follows where t is the thickness in inches:

$$\frac{8N^2\int tr^2dr+\frac{N^2}{2\pi}\times\text{total radial bucket load at 1000 rpm}}{\text{Cross-sectional area of disk profile from center out}}$$

For a section of uniform thickness

$$8N^2\int_{R_1}^{R_2}tr^2dr=8N^2t\frac{R_2^3-R_1^3}{3}$$

For a section of conical profile whose thickness is t_1 at R_1 and t_2 at R_2

$$8N^2\int_{R_1}^{R_2}tr^2dr=8N^2\left[(t_2+AR_2)\frac{R_2^3-R_1^3}{3}-A\frac{R_2^4-R_1^4}{4}\right]$$

where $A=(t_1-t_2)/(R_2-R_1)$

For a section of hyperbolic profile defined by the relationship

$$\frac{t}{t_2}=\left(\frac{R_2}{r}\right)^k$$

$$8N^2\int_{R_1}^{R_2}tr^2dr=8N^2\frac{t_2R_2^k}{3-k}(R_2^{3-k}-R_1^{3-k})$$

In the last case the corresponding area of cross section between R_1 and R_2 is

$$\int_{R_1}^{R_2}tdr=\frac{t_2R_2^k}{1-k}(R_2^{1-k}-R_1^{1-k})$$

The area for the first two profiles is easily computed without formulation.

Discussion

Hans Dahlstrand.[2] A short time ago the writer's company made similar tests to those of the paper on disks machined from a large forging which was scrapped because of flaws in one end of the forging. These flaws consisted of inclusions and pin-point cavities which occasionally occur in forgings. Normally, flaws of this type will cause the forging to be scrapped if they are located at a critical point or if the flaws cover a large area. There are at present no reliable means by which the extent of such flaws below the finished surface may be examined; consequently, the decision of whether or not the forging is suitable for use rests largely upon the judgment of the engineer. It is clear that any cracks or seams cannot be tolerated, as during operations flaws of that nature are likely to increase until bursting occurs.

The purpose of the tests was to determine the extent to which the inclusions and round cavities would affect the ductility and the strength of the steel. Several disks from this forging, both from the area showing the flaws and from areas showing sound metal, were machined for the tests. While the report of tests has not been fully completed, it is sufficient to mention that flaws of the nature described do not materially affect the strength or the ductility of the steel. It is therefore concluded that forgings having such flaws may be used with safety without any particular danger. Care nevertheless must be exercised in the examination of the flaws before approving the use of such forgings, as it is reasonable to assume that these flaws may be of such nature and cover such large areas at vital points that the forgings could not be safely used.

R. V. Kleinschmidt.[3] Perhaps the most interesting data presented in this paper, and material which would warrant very careful study, are the behavior of the disks at speeds lower than the rupturing speed. The study of distortion gives considerable insight into the actual phenomena occurring. It would be interesting to have more consideration of the phenomena of distortion, and some comments as to the practical possibilities of prestressing wheels by overspeeding to stabilize the deformations before final finishing and assembly.

The findings on the importance of ductility in relation to stress concentration around a central hole are somewhat surprising until we realize that all of the materials tested were sufficiently ductile to permit the necessary relief of stresses without

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rupture. An extension of the tests to materials having 2 to 5 per cent elongation would probably present another story. Also, it must be remembered that these remarks apply to a central hole with uniform stress conditions around it. It is doubtful if similar conclusions can be drawn with respect to holes eccentrically placed. Possibly the author could comment upon experience with such holes in disks.

Another item which has not been mentioned and which must be considered is that of temperature effects. There are three interesting conditions which come to mind: (1) Uniform high temperature or low temperature, will change the strength and ductility relationships of the materials. Atmospheric tests alone may be misleading at other temperatures. (2) Stresses due to nonuniform temperature distribution. In this connection one possible type of nonuniform temperature distribution would appear, from the data given, to be very serious, namely, a cooled disk with a hot rim. (3) Temperatures varying rapidly with time as in heating up and cooling down. One of the major problems in gas-turbine development may be the possibility of changes of several hundred degrees in the temperature of the working fluid in a fraction of a second.

R. P. Kroon.[4] It is a commonly accepted idea that when a ductile material is under steady load, a local stress concentration is not too serious. It is assumed that there will be some plastic flow, after which the stresses will be distributed more evenly. This presupposes that the amount of local deformation will not be sufficient to cause rupture.

On this basis the notion that average stress instead of local peak stresses should be a criterion for disk failure seems sound enough.

We are confronted, however, with the results shown in Table 6, indicating that "the most ductile material loses most due to a hole."

It would certainly be interesting to establish whether this behavior is due to some internal-stress condition of the disks or whether the type of stress-strain curve of the material might play a role. A mathematical analysis might prove helpful in the latter case.

The high-temperature alloy disks with holes, examined by the author, have elongations of at least 15 per cent and fail at stresses in the order of 120,000 psi at room temperature.

In high-temperature service these materials would be loaded with stresses of perhaps 30,000 psi, and their elongation, with a reasonable service life, would be only a few per cent.

It would be extremely interesting to know whether under high temperature, with creep as the dominant factor, the comparison of disks with and without holes would be similar to that established at room temperature. Can the author throw any light on this phase of the disk problem?

J. A. Maul.[5] This paper has positively answered a question which has been an open point for discussion in the dynamometer engineering section of our company for several years, i.e., the possibility of taking advantage of the fact that a solid wheel theoretically has only one half the maximum stress of a corresponding wheel with a small central hole. Cracks at the center of a forging or minute openings between the grains may be the equivalent of a central hole.

It is important to know that we can run a solid wheel to higher speed than the corresponding shrunk-on shaft construction. However, the writer would be interested in knowing how much faster the solid wheel could be run. Evidently the limiting feature in the case of the solid wheel would be the ductile expansion of the outer surface of the wheel. In the case of a shrunk-on wheel, the limiting feature would be the expansion of the bore to the point where the wheel loosened on the shaft.

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D. J. McAdam, Jr.[6] The writer cannot agree that the evidence presented by the author gives a basis for conclusions as to criteria for "failure."

The combination of principal stresses affects all three strength indexes; yield stress, ultimate stress, and breaking stress.[7,8,9] In considering criteria for fracture, one should compare "true" breaking stresses under different stress systems, and allowance should be made (if possible) for differences in total plastic deformation under the two conditions. The author, however, compares the breaking stress, either average or "elastic," with the tensile strength of a specimen under unidirectional loading. The "true" breaking stress under unidirectional loading would be somewhat greater than the ultimate stress, and probably would be greater than the breaking stress under biaxial loading. The higher value under uniaxial loading, however, would be due to the greater total plastic deformation. The relative values of the two technical cohesion limits would be the resultant of two factors, total plastic deformation (ductility) and the stress system. As the ductility (measured by decrease in sectional area) was much less for biaxial than for uniaxial loading, the influence of the stress system cannot be determined from the experiments.

For a brittle metal, the technical cohesion limit probably would be greater under biaxial than under uniaxial loading. Such a relationship, however, would not necessarily mean that the criterion for fracture is the greatest principal strain. As illustrated in Fig. 17 of a paper by the writer,[7] the criterion for fracture possibly is a function of both the volume stress and the greatest principal stress; if not, the criterion must be expressed in terms of the three principal stresses.

Arthur McCutchan.[10] This paper is a straightforward account of tests made to verify engineering judgment regarding the significance of certain types of stress concentration. Such tests insure that results of stress analysis will be considered in their proper perspective.

The improvement in performance of the wheels made with a special contour giving a uniform stress throughout the wheel is noteworthy. An examination of the cross section of this special design, as shown in Fig. 3 of the paper, however, suggests that straight lines giving a trapezoidal shape would be simpler to produce than the double-curve bottle shape and should give practically identical results.

The means of fastening the solid disks onto the rotating device is evident from Fig. 4 of the paper, but how this was accomplished for the disks with center holes is not shown. It would seem that any clamping or expanding device applied at or around the hole might appreciably affect the results. It would be of interest to learn how the wheels with holes were attached in making these tests.

[6] Metallurgist, National Bureau of Standards, Washington, D. C.

[7] "The Technical Cohesive Strength and Yield Strength of Metals," by D. J. McAdam, Jr., tech. pub. 1414, Trans. A.I.M.E., vol. 150, 1942, pp. 311–357.

[8] "An Investigation of the Technical Cohesive Strength of Metals," by D. J. McAdam, Jr., and R. W. Mebs, A.I.M.E. tech. pub. 1615, Metals Technology, August, 1943.

[9] "The Technical Cohesive Strength and Other Mechanical Properties of Metals at Low Temperatures," by D. J. McAdam, Jr., and R. W. Mebs, American Society for Testing Materials, preprint no. 40, 1943.

[10] Engineer, Engineering Division of The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

While, as stated by the author, the evaluation of high-temperature qualities of the various materials is another story, it occurs to the writer that if spinning tests of this kind could be made at high temperatures they might furnish quite different information than conventional creep and rupture tests. For one thing, it is possible that the ductility of the high-alloy materials under biaxial stress might be more nearly like that of the carbon and low-alloy steels than found from tests at room temperature.

K. D. McMahan[11] and E. E. Stoeckley.[12] This paper confirms the intuitive beliefs of most designers of high-speed rotors who lacked actual tests to prove them. The work signals the approach of a new method of specifying materials for high-speed rotors. Fortunately, most "good rotor materials" give a higher factor of safety than classical calculations indicate. However, rotors from certain aluminum alloys, usually having low elongation in tensile tests, are known to burst at speeds equal to or below that at which the bore stress, as calculated by classical methods, equals the tensile strength. Similarly other alloys having not too different tensile characteristics behave like the turbine-rotor materials and burst at speeds corresponding very nearly to average stresses. Thus the difference in bursting speeds of similar specimens may vary almost by a factor of 2:1 for materials having the same apparent tensile strengths as determined by the usual tensile tests.

A discussion of this paper seems incomplete without mentioning the rather unique apparatus that makes these bursting tests possible. As early as 1939, the G. E. Research Laboratory, noting the work of Beams[13] and others, undertook to develop apparatus for whirling aluminum-alloy rotors weighing from 10 to 25 lb up to 50,000 rpm. Subsequently, in March, 1940, a steel rotor weighing 25 lb was burst at just under 40,000 rpm. Since the early work in the laboratory, the art has progressed beyond all early expectations, and there now seems to be no limitation to the size of rotors that may be tested by this method.

An adequate description of this test apparatus is a subject for a paper in itself. Plans have been made for such a paper when the time permits. It must suffice at present to state that the specimens are driven by a small air turbine while suspended on a small vertical piano-wire shaft in an evacuated explosion chamber.[14] The turbine mounts atop the chamber with a special seal to prevent leakage around the wire shaft.

A. Nadai.[15] The distribution of the internal stresses in a rotating steam- or gas-turbine disk which has been stressed beyond the plastic limit of the material is an attractive mechanical problem which, however, is not new. Several investigation reports have been published dealing with the analysis of this case and some related cases of biaxial stresses with a rotational symmetry in the plastic state, to which no references could be found in the paper. The writer believes that a quantitative analysis of plastic distributions of stress of this kind can give and has already given valuable information to the designers of turbine wheels, even if some of these computations have been based upon idealized conditions with regard to the flow, and despite the fact (mentioned in the paper) that duplicate tests made with bars of some of the alloy steels showed widely varying plastic behavior under stress. Whether the opinion expressed by the author "that such detailed analysis would not be fruitful of any useful information" is sufficiently justifiable or can be based on facts, must be left an open question.

Engineers are accustomed to base the design of steam- and gas-turbine disks in the elastic range of strains on the analytical methods which have been developed for such computations. It is believed that there cannot be advanced any serious reasons against attempts why these well-established methods could not be utilized with success and furnish reliable information to designers also when the plastic state has been reached in revolving disks. It may even occur that the distributions of stress in this latter state are simpler to analyze due to certain simplifications which can be introduced in the assumptions, because, for certain ductile steels (such was the case for low-carbon steel S.A.E. 1020), the tangential stresses change much less rapidly with the radial distance from the disk center, as in the elastic state. This alone probably explains several facts concerning fracture which were recorded in the author's paper in an empirical fashion.

In artillery arsenals, a related problem was investigated long ago, namely, the plastic distribution of the stresses in gun barrels which have been cold-worked and overstressed by internal hydrostatic pressure. In A. Stodola's book on steam and gas turbines, the case of an overspeeded turbine disk has been treated.[16] F. Laszlo[17] dealt with the same problem, and particularly with the stability of the equilibrium of rotating disks in the plastic state. He showed that their equilibrium may become unstable under certain conditions which depend upon the shape of the stress-strain curve of the material.

L. H. Donnell and the writer in a joint paper[18] investigated the stress distribution in rotating disks of equal thickness of ductile materials after the yield point had been reached in them. In this paper, two or three typical cases of flow were considered in detail, namely, "progressive" yielding and "complete" yielding for a material having a well-defined yield stress. In the former case, the outer portion of the disk is stressed elastically, while in the inner portion the stresses were assumed at the plastic limit. In the third case which was considered, it was assumed that the material had a stress-strain curve in which strain-hardening would become more pronounced. A case was worked out as an example in which the true stress was a linear function of the conventional permanent strains. This curve had a definite yield point and a finite slope. Such examples show that the distributions of the radial and tangential stresses can be obtained in rotating disks in the plastic state, at least in a few characteristic simple cases, in a quantitative manner. The means which are available for an analysis of the stresses in overstressed or overspeeded disks are precisely of the same or an analogous nature as those which are in use by designers in the elastic range of the strains. Stress-strain relations have been studied in the plastic range under biaxial or combined stress conditions in a satisfactory manner for moderate strains. (Such were the fairly small permanent strains which were observed in the author's overspeed tests.)

It may perhaps interest readers of the paper to note that some

[11] Research Laboratory, General Electric Company, Schenectady, N. Y.

[12] Supercharger Engineering Division, General Electric Company, Lynn, Mass.

[13] "High Speed Centrifuging," by J. W. Beams, *Reviews of Modern Physics*, vol. 10, 1938, pp. 245–263.

[14] "High-Altitude Hot-Gas Standards for Testing Turbosuperchargers and Engines," by E. E. Stoeckley, presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

[15] Consulting Mechanical Engineer, Westinghouse Research Laboratories, East Pittsburgh, Pa. Mem. A.S.M.E.

[16] "Steam and Gas Turbines," by A. Stodola, McGraw-Hill Book Co., Inc., New York, N. Y., 1927, vol. 2, p. 1080. Reference is also made to an article, "Contribution au Calcul des Disques," *Brown-Boveri Review*, vol. 6, 1919, pp. 260–263.

[17] "Geschleuderte umdrehungskörper im Gebiet bleibender Deformation," by F. Laszlo, *Zeitschrift für angewandte Mathematik und Mechanik*, vol. 5, 1925, pp. 281–293.

[18] "Stress Distribution in Rotating Disks of Ductile Material After the Yield Point Has Been Reached," by L. H. Donnell and A. Nadai, Trans. A.S.M.E., vol. 51, 1929, paper APM-51-16.

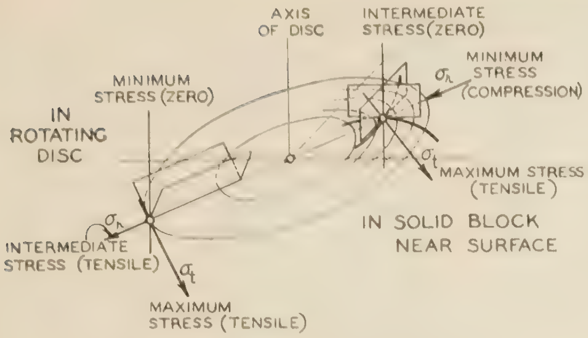

Fig. 11 Orientation of Slip Planes

of the author's observations, namely, those concerning the orientation of the flow figures which became visible on the flat sides of the low-carbon-steel disks, also conformed quite nicely with what was to be expected from the Mohr theory of the formation of the surfaces of slip in a ductile metal. It is characteristic that they appear usually in an orientation in which the two equivalent planes of slip in a point in a plastic metal intersect each other along the line of the intermediate principal stress. This is illustrated in Fig. 11 of this discussion, which refers to two characteristic cases of rotational symmetric plastic flow. On the right side of Fig. 11 is shown the orientation of the two slip planes in a point situated in the surface of a solid block near an impression in which high concentrated localized compression stresses are applied to the surface in the perpendicular directions to it. In this case one of the principal stresses is a high tensile stress (the peripheral stress σ_t) and the other is a compression stress (the radial stress σ_r). The direction of the intermediate stress (which in the surface is equal to zero) is therefore perpendicular to the free surface of the solid block. The two planes of slip must therefore be perpendicular to this plane and intersect the radii and concentric circles (shown in Fig. 11) under an angle of 45 deg. The flow figures should therefore consist of a series of logarithmic spirals, like the ones which are indicated in Fig. 12 of the discussion. Such flow lines are frequently observed around punched holes in heavy plates, Brinell impressions, etc., in mild steel.

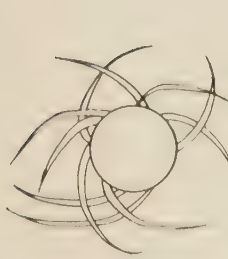

Fig. 12

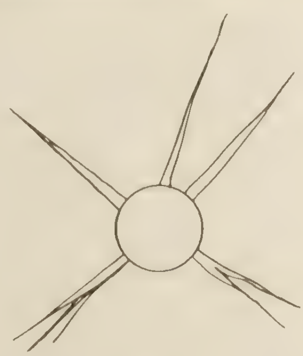

Fig. 13

Fig. 12 Surface of a Solid Block Near an Impression

Fig. 13 Rotating Disk

In the case of the rotating disk, on the other hand, both principal stresses are tensile stresses. The radial (σ_r) is always smaller than the tangential (σ_t), the third one (σ_a) is again zero. The intermediate stress acts now in the radial directions. Consequently, the flow lines should appear along the radii of the revolving disks. This was the case in the Figs. 6, 7, and 8 of the paper for the overspeeded disks of low-carbon steel. The flow lines in them appeared in the directions shown in Fig. 13 of this discussion.

The tests described in the paper should throw some light on an important question, to which attention should be called also through these remarks. Suppose that a small circular hole is present in a disk which is stressed uniformly in its plane in all directions. What is the nature of the disturbance in the distribution of the stresses caused through the presence of the hole? More precisely, what is the distribution of the radial and tangential components of stress σ_r, σ_t around the hole? In an elastic material one expresses the result of such computations by stating the ratio k of the peripheral stress at the boundary of the hole to the stress in large distances of it. This is known as the factor of stress concentration. In this case it is $k = 2$. In our paper,[18] it was shown that in a rotating disk having a small hole the tangential-stress components are nearly uniformly distributed over the entire disk if the material yields under a constant stress. Since the periphery of the hole is in a state of pure tension the factor of stress concentration for a perfectly plastic disk must have been reduced to unity. From these considerations one should expect that the factor k should essentially depend upon the shape of the stress-strain curve or on the law of strain-hardening, according to which $\sigma = f(\epsilon)$, where ϵ is the plastic strain. Not much has yet become known quantitatively in this respect about values of k, the difficulty being that with the increasing stresses the dimensions of the plate near a hole must also be changing. In one case the writer has computed an exact expression for the factor of stress concentration in a plastic disk having a hole, which was uniformly stressed in its plane.[19] The strains were assumed to be small and for the material in the plastic state a stress-strain curve was assumed

$$\sigma = \sigma_0 \epsilon^m$$

or a power function (σ stress, ϵ permanent strain, σ_0, m material constants, $0 < m < 1$). Factor k for this strain-hardening law was found to equal

$$k = \left(\frac{2}{1 + 3m}\right)^{\frac{2m}{1 - 3m}}$$

which gives for

Exponent m	= 1	2/3	1/2	1/3	1/4	1/5	0
Factor of stress concentration k	= 2	1.72	1.56	1.40	1.31	1.25	1

As long as the maximum value of the peripheral stresses should occur in the boundary of the hole and not as it could in some special cases also occur, in the interior of the disk, one would expect that the material should break at the hole in pure tension. It would, however, not be permissible to base the design on the value of the tensile strength or of the true fracture stresses which were observed in ordinary tensile tests run with standard round bars, because necking in a plate around a hole or in a wide flat bar under tension must be restricted, compared to the usual mode of necking in the standard round-bar tensile test. This points to the need of studying fracture conditions in the ductile metals under special conditions which cannot be reproduced through the standard tensile test.

E. M. Phillips.[20] The author's tests indicate the previously accepted theory of safety as dependent upon the maximum elas-

[19] "On the Creep of Solids at Elevated Temperatures," by A. Nadai, *Journal of Applied Physics*, vol. 8, 1937, p. 430.

[20] Turbine Engineering Division, General Electric Company, (Lynn) River Works, Lynn, Mass. Mem. A.S.M.E.

TABLE 7 BURSTING TESTS ON SHAPED TURBINE-WHEEL DISKS

Material	Location	Physical Test Results: Tensile stress, psi	Elongation, per cent	Reduction, per cent	Bursting Test Results: Bursting speed, rpm	Average tangential stress, psi	Ratio (7)/(3)	Ratio tensile strength (at center)/(at rim)
(1)	(2)	(3)	(4)	(5)	(6)	(7)		
Gamma cb.	Rim	147000	8	32.1	42000	79600	0.542	0.835
	Center	123000	4	6.9			0.648	
Gamma cb.	Rim	131100	10	23.1	46800	98800	0.743	0.936
	Center	124700	8	11.6			0.793	
Gamma cb.	Rim	122400	10	21.0	38700	67500	0.552	1.00
	Center	122400	5	18.2			0.552	
Gamma cb.	Rim	157400	11	17.5	43000	83400	0.530	0.90
	Center	141900	6	4.7			0.588	
Gamma cb.	Rim	149400	8	19.0	42000	79600	0.533	0.80
	Center	119300	2	3.1			0.667	
Gamma cb.	Rim	148600	8	14.5	41000	75700	0.510	0.835
	Center	124000	1	0.81			0.610	0.884 Avg
17W	Rim	138000	15	20.4	43000	81400	0.590	
17W	Rim	133700	15	40.4	49000	105500	0.791	
17W	Rim	142100	16	32.0	48600	104000	0.731	

TABLE 8 COMPARISON OF TEST RESULTS

Material	Ratio average tangential stress to tensile strength: E. L. Robinson	Tests Table 7	Ratio	Based on tests at
Gamma cb.	0.812 avg	0.510 to 0.743	0.628 to 0.915	Rim
		0.552 to 0.793	0.68 to 0.977	Center
17W	0.793 avg	0.590 to 0.791	0.745 to 0.998	Rim

tic stress to be in error. The evidence presented shows rather that safety is dependent either upon the average stress or upon the ductile properties at the point of maximum elongation or reduction in area.

These tests were made upon smooth wheel disks. Disks having points of stress concentration may be expected to show less favorable performance. Table 7 of this discussion shows fracture results on nine shaped turbine-wheel disks $9^9/_{16}$ in. diam, bolted to a shaft hub through three reamed holes $^7/_{16}$ in. diam on a radius of $2^3/_4$ in. These holes are rounded by $^1/_8$ in. radius at the disk faces. In addition, there is a male rabbet $1^9/_{16}$ in. diam joined to the disk face by a radius of $^5/_{32}$ in. Breaks in these disks pass through the attaching holes and are probably influenced by the concentrations at the holes and at the rabbet. The table also includes tensile-test properties taken from broken sections of the disk. The test elongation values may have been influenced by the cold work in bursting.

It is interesting to note that the tensile strength at the center varies from 80 per cent to 100 per cent of that at the rim, the average value being 88 per cent.

In comparison, Table 8 herewith indicates a reduction due to stress concentration as high as 32 per cent when based upon center properties and 37 per cent when based upon rim properties.

Arthur M. G. Moody.[21] Mr. Robinson's valuable paper supplies us with a good deal of quantitative information in a field where we have been largely limited to conjecture. It also raises some questions, one of which the writer would like to discuss.

Mr. Robinson mentions that the word "failure" is not always clearly defined. It is often taken to mean the plastic yielding of some part of a machine as a result of the presence of a stress in excess of the elastic limit. He states that this is often not detrimental to the performance of a machine, and he therefore turns our attention to the bursting speed of a disk rather than the speed at which the elastic limit is reached.

Now it is probably perfectly true that designers, in dealing with stresses, do not consciously subscribe to the view that failure refers to the elastic limit. Nonetheless, it is probably also true that the rules by which designers are guided have this philosophy implicit in them.

Most disk stress criteria consist of a comparison between a stress (maximum or average) calculated on the basis of elastic theory, and some property found from a single tensile test, such as the yield point. But Mr. Robinson has made it clear that bursting speeds have practically no connection with elastic stresses. Should we not, therefore, have other criteria?

As a matter of fact, a good deal can be said against permitting extensive yielding in a disk, so that perhaps our design criteria should, after all, be kept on a basis similar to that now in general use. One important change might, however, be made, namely, the use of an effective elastic limit making allowance for the stiffening effect of the biaxial stress condition which exists in the disk. Such an elastic limit would be based on the simple tensile test but would be corrected, perhaps empirically, according to the degree of elongation and reduction of area.

It would be most interesting if Mr. Robinson could find the opportunity to conduct further tests relating to the incidence and progress of plastic yielding.

Author's Closure

The author concurs with Mr. Dahlstrand's remarks. However, the bursting test may show the general soundness of the material, but may not reveal a weakness in the matter of fatigue resistance.

Commander Kleinschmidt inquires about the distortion preceding rupture. The wheels were run at successively higher speeds, and, after the first appearance of plastic set, records were kept of dimensional changes. These records were rather voluminous, and the author feared that the paper might get too long for publication had they been included. Suffice it to say here that the different materials behaved quite differently.

As regards his inquiry about prestressing by overspeeding before assembly, the author has always felt that just as much good or even more would be done by such overstressing as occurs in ordinary service as by deliberate overspeeding in advance of assembly. Under actual service conditions, no more prestressing than necessary is applied. Overspeeding in advance is attended by the remote possibility of aggravating some hidden flaw or weakness that would not otherwise be serious. This is only a statistical possibility. If in any particular case the material is presumed to be perfectly sound, prestressing should be beneficial. Undoubtedly, this procedure helps keep a shrunk-on wheel tight on the shaft. However, evidence is accumulating that resistance to fatigue is improved by initial compression and reduced by initial tension. Bearing this in mind, the bore of the wheel would be

[21] Chief Research Engineer, DeLaval Steam Turbine Company, Trenton, N. J. Jun. A.S.M.E.

improved in fatigue strength by the localized initial compression induced by overspeeding; whereas the outer regions of the wheel would be very slightly weakened by the small initial tension. Thus overspeeding a wheel does nothing to improve the fatigue strength of the part of a wheel which may be subject to vibration but improves the part that does not need it. On the other hand, the dovetail connections of a fully assembled turbine bucket wheel might well be improved in fatigue strength by overspeeding the finished and bucketed wheel to a speed which would result in some residual compression in the local regions of maximum stress.

Commander Kleinschmidt also inquires about holes eccentrically placed. Such holes are very likely to be located in a region where the radial and tangential stresses are not greatly different in size. Under such conditions, the behavior should be much the same. Of course, if the radial and tangential stresses differ greatly, the behavior might be expected to differ. On the other hand, the author believes that a more important consideration may be the fact that such holes are likely to be of small size and that there may be a worst size for regions of stress concentration.

Mr. Phillips has kindly contributed in Table 7 the results of bursting tests on actual shaped wheels containing small-size eccentrically located holes, such as referred to by Commander Kleinschmidt. Table 8 shows that such wheels when made of the gamma columbium material develop average bursting stresses 0.628 to 0.915 of the author's average figure. (Rim physical properties are used because all the author's comparisons are with rim tests.) In the case of the Cyclops 17W material, the actual shaped wheels containing eccentric holes were 0.745 to 0.998 as strong as the author's wheels. In both cases there is evidence of a reduction of strength due to the local stress concentration, but this reduction is not as bad as one might be led to believe on the basis of elastic theory.

Commander Kleinschmidt asks three questions covering the importance of temperature effects. These are certainly very important considerations upon which the author can only speculate as follows, at the present time:

1 High temperature is likely to be accompanied by a loss of strength and an increase of ductility while low temperature would have the opposite result. There are variations from these relationships, and certain unexpected effects connected with the nature of the ductility. Certainly the atmospheric tests must not be assumed to give conclusions valid for other temperatures.

2 There are mitigating circumstances associated with the increase of stress at the center of a wheel with a hot rim so that this condition is, perhaps, not quite so serious as it might seem to be. The colder and more highly stressed center is usually stronger than the rim by virtue of its lower temperature. Furthermore, the volume of stressed material is usually very much larger toward the center, with the result that a larger thermal strain in a rim of small cross section cannot cause a very excessive opposite strain in the very large cross section of the central web or hub.

3 Certainly the effect of rapidly varying temperatures is a very real problem. In this matter, the author can only suggest that much similar effects exist at a lower temperature level in steam turbines using lower-alloy materials. It is to be hoped that the newer alloys will behave equally well at the higher levels.

The author shares with Mr. Kroon the desire for some rationalization of the paradoxical results given in Table 6. The author believes that both influences suggested by Mr. Kroon were present. In retrospect, it is only possible to speculate on the likelihood that initial internal stresses would go a long way toward explaining what happened. As regards Mr. Kroon's question about high-temperature behavior, the author is at present unable to give any answer.

In connection with Mr. Maul's problems, it is important, of course, to consider the difference in the physical properties of the material available for a solid rotor, as compared with what may be obtained in separate shrunk-on wheels.

The author has a good deal of sympathy with Dr. McAdam's disagreement as to criteria for "failure." The crux of the question seems to be the definition of "failure," but the point of view is just as important also. If one is investigating the ultimate nature of a material, certainly, "true" breaking stresses should be determined as Dr. McAdam suggests, allowing "(if possible) for differences in total plastic deformation." In the first draft of his paper the author started to do this. However, upon reflection, it occurred to him that plastic deformations are never, and by the nature of the materials cannot with certainty, be available to a designer. Therefore having the designer's viewpoint, it seemed better to base the figures on dimensions available to the designer, as is done in the tensile test under unidirectional loading. The artificiality of the results must be admitted, but there is a corresponding directness of utility to a designer that rests for its validity only on the systematic outcome of the results. The author regrets that complete data on the progressive deformation of the wheels were omitted in the interest of brevity, as Dr. McAdam's analysis of the "true" stresses, taking account of plastic deformation, would be valuable to have.

The author agrees with Mr. McCutchan that the trapezoidal cross section should be practically identical in result with the bottle shape. In fact, the weakest point in this design was the generous fillet uniting the very modest rim with the web. This detail was not generous enough.

How to float the disks with center hole while rotating at 40,000 rpm proved to be quite a real problem by itself and various devices were tried. Suffice it to say that the most successful device was shaped like a tulip blossom and about that size, with the wheel seated just below the petal tips and the pistil of the flower serving as drive shaft. The outward centrifugal pressure of the petals at the small central radius was practically inconsequential but sufficed to transmit the small torque required for acceleration.

The author agrees as to the importance of securing similar information at high temperature in order to find out how to apply conventional creep and rupture tests.

The author is completely indebted to Messrs. McMahan and Stoeckly for carrying out all his bursting tests in their high-speed stands and looks forward to the time when those most useful testing machines may be adequately described. Their remarks in passing about the different behavior of certain aluminum alloys should be noted.

Mr. Moody recalls a totally different implication in the use of the term "failure" from that suggested by Dr. McAdam. Designers must certainly always have in mind that their machines may be put out of business by a very little permanent deformation of an amount which may not suffice to permit bursting or stresses anywhere near bursting. Doubtless this is a legitimate use of the word "failure," but the author regrets the practice of basing a "factor of safety" on the attainment of the elastic limit, because usually no safety is in jeopardy when this occurs even though the type of failure mentioned may be imminent.

The author wishes to express his appreciation of Dr. Nadai's valuable and illuminating discussion. He makes a good case in favor of attempting to predict behavior after plastic action has set in with sufficient accuracy to be useful to designers. The author would regret having the skepticism expressed in his quoted remark taken to mean that a sound and useful analysis would not be welcome. Certainly it would be desirable if a satisfactory analysis could be agreed upon which would improve upon the simple average stress suggested by the author. However, such a setup for design purposes would have to be accompanied by data on the appropriate strength values. It is plain from both

Dr. McAdam's remarks and from Dr. Nadai's also that ordinary tensile-test results could not be used. The author believes that Dr. Nadai's last paragraph expresses opinions pretty much in agreement with his own and also with those of Dr. McAdam. The more common use of high-speed test stands may, in future, make it possible to determine the bursting strength of a test coupon having the size and shape of a baker's cookie instead of a stick of candy.

In answering one of Commander Kleinschmidt's questions, the author made use of some of the data contributed by Mr. Phillips. There is another lesson to be learned from these figures. Mr. Phillips gives physical tests from both rim and center of the gamma columbium wheels, showing the center usually to have lower tensile strength anywhere from 80 to 100 per cent of the rim tensile strength. All the tensile tests cited by the author were based upon coupons taken from the rims before the wheels were made. Since all the bursts except those of the wheels of uniform stress design apparently originated near the center or at the edge of the center hole, this variation of material strength may go a long way to explain the deviation from a uniform relationship between average bursting stress and ordinary tensile strength. The author does not, of course, claim that this relationship is other than fortuitous with so artificial a property as the conventional tensile test. It may, however, be nonetheless quite a real and useful relationship.

Fluid Flow Through Two Orifices in Series—II

By MILTON C. STUART[1] and D. ROBERT YARNALL[2]

This is the second of a series of papers on the characteristics of fluid flow through two orifices in series. The first paper (1)[3] presented a theoretical analysis of the flow of water and steam through two orifices in series over a wide range of pressures and temperatures. This paper presents the results of experimental investigations on this subject using water at an initial pressure of 100 psia and over a range of temperatures from room temperature to saturation temperature. The principal data sought were the pressures established in the intermediate chamber between the two orifices, and especially the variations of this intermediate pressure with reference to the temperature of water supplied and the type of orifice. One of the most significant facts revealed by these researches is that under certain conditions of flow the fluid assumes a metastable state which greatly influences the characteristics of the process. Finally, a device responsive to the pressure variations obtained between two orifices in series constitutes a control element, useful, among other things, to control flow in response to the state of the fluid supplied.

Nomenclature

The following nomenclature is used in the paper:

M/A = mass rate of flow, lb per sec per sq ft
U = velocity, fps
V = specific volume, cu ft per lb
H = enthalpy, Btu per lb
P = absolute pressure, psf
p = absolute pressure, psi
F = temperature,

Subscript 1 refers to entrance to first orifice
Subscript x refers to intermediate chamber
Subscript 2 refers to downstream section following second orifice
Subscript s refers to constant entropy process
"Chamber pressure," (p_x), is absolute pressure in intermediate chamber between two orifices
"Cold water" is water which has a temperature lower than saturation temperature corresponding to lowest pressure attained by water during flow
"Compressed liquid," see "subsaturated" liquid
"Critical pressure" is lowest pressure which will exist for flow of expanding fluid in converging portion of nozzle
"Discharge pressure," (P_2), is pressure of liquid after passage through second orifice
"Flashing" is process of partial evaporation of liquid caused by adiabatic reduction of pressure
"Hot water" is water which has a temperature higher than saturation temperature corresponding to lowest pressure attained by water during flow
"Initial temperature" (t_1) is temperature of fluid at entrance to first orifice
"Initial pressure" (p_1) is absolute total pressure of liquid at entrance to first orifice
"Metastable" refers to a fluid condition in which pressure of fluid is less than saturation pressure of liquid corresponding to existing temperature, or, alternately, a fluid condition in which temperature of fluid is higher than saturation temperature of liquid at existing pressure
"Orifice" is a converging or parallel-sided passage provided for flow
"Quality" is percentage of vapor present in a liquid-vapor mixture
"Saturated liquid" refers to water having a temperature equal to temperature listed in Steam Tables corresponding to existing pressure
"Saturation pressure" refers to pressure listed in Steam Tables corresponding to existing temperature of water
"Saturation temperature" refers to temperature of water as listed in Steam Tables corresponding to existing pressure
"Stable" refers to a fluid condition in which pressure of fluid is equal to or greater than saturation pressure, or alternately, a fluid condition in which temperature of fluid is equal to or less than saturation temperature
"Subpressure" is reduction of pressure of metastable water below saturation pressure
"Subsaturated liquid" (compressed liquid) is a liquid having a temperature less than saturation temperature, or alternately, a liquid having a pressure higher than saturation pressure
"Superheated liquid," same as "metastable liquid"
"Superpressure" is excess of pressure of stable water over saturation pressure
"Supertemperature" is excess of temperature of metastable water over saturation temperature
"Total pressure" is measured absolute pressure of a fluid at any point
Vapor is used in this paper to designate vapor phase of water
Water is used in this paper to designate liquid phase of water

Introduction

In a previous paper (1) the authors presented a theoretical analysis of the flow of saturated and subsaturated water through two orifices in series over a wide range of pressures and temperatures. Since the presentation of that paper an extensive experimental investigation has been conducted in the high-pressure laboratory (2) of the Yarnall-Waring Company to determine the characteristics of flow of hot water through various types of orifices in series. The most significant facts revealed by these researches are that in many types of flow the fluid is in a metastable state and that in the metastable state the characteristics of the

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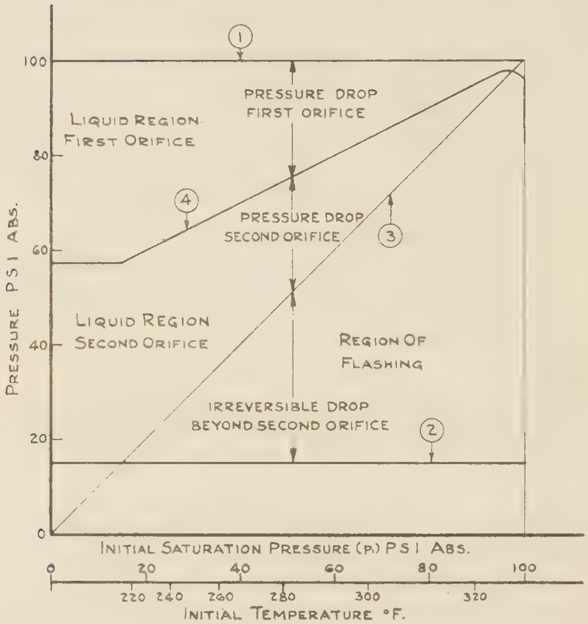
[3] Numbers in parentheses refer to the Bibliography at the end of the paper.
Contributed by the Research Committee on Fluid Meters and the Power Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.
Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

flow of water through orifices differ greatly from the characteristics under stable conditions. This paper presents the experimentally determined characteristics and a thermodynamic analysis of the flow of saturated and subsaturated water through orifices in series with special consideration of the existing metastable state.

FIG. 1 ARRANGEMENT OF TWO ORIFICES IN SERIES



1 Constant initial total pressure
2 Constant discharge-region pressure
3 Initial saturation pressure
4 Intermediate pressure

FIG. 2 PRESSURE DIAGRAM FOR IDEAL STABLE FLOW OF WATER THROUGH TWO ORIFICES IN SERIES

FLOW OF HOT WATER UNDER STABLE CONDITIONS THROUGH TWO ORIFICES IN SERIES

In the paper presented by the authors in 1936 (1) on fluid flow through two orifices in series a thermodynamically stable condition of the fluid was predicated. It was assumed that when the pressure decreased to a point as low as the saturation pressure corresponding to the temperature of the liquid, vapor immediately started to form and the temperature of the resulting mixture of liquid and vapor was subsequently that corresponding to the saturation pressure. Following are some of the conclusions of the former paper which pertain to the present investigations:

(*a*) The flow of subsaturated water under stable conditions through an orifice or nozzle is characterized by a critical-pressure phenomenon just as is the flow of a gas or vapor. For subsaturated water this critical pressure is practically equal to the saturation pressure of the entering liquid.

(*b*) Because of this critical-pressure phenomenon, the rate of flow of hot water under stable condition through orifices is much lower than the rate of flow of cold water. In orifice flow, the energy available to cause flow is limited by the amount of energy rendered available by the change in state of the liquid from the initial pressure to the critical pressure.

(*c*) In flow of hot water under stable conditions through two identical sharp-edged or rounded-entrance orifices in series, critical pressure existed at the outlet of the second orifice and the intermediate pressure in the chamber between the two orifices automatically adjusted itself to be midway between the initial pressure at the entrance to the first orifice and the critical pressure. Flashing did not occur in the intermediate chamber until the saturation pressure of the liquid closely approached the initial pressure.

Pressures existing throughout the system for flow under stable conditions through two orifices in series, Fig. 1, are shown graphically in the pressure diagram, Fig. 2. In this diagram the abscissa is the initial saturation pressure of the water entering the first of two identical orifices with an initial total pressure 100 psia. The corresponding initial temperatures are also marked on the abscissa. Ordinates represent pressures existing throughout the system for flow under conditions of the various initial temperatures (and corresponding initial saturation pressures).

Line 1 represents the constant initial pressure; line 2 represents the constant pressure in the discharge region beyond the second orifice; line 3 represents the initial saturation pressure and is also the pressure which, under stable conditions, must necessarily exist at the throat of the second orifice. Line 4 represents the variation of the pressure in the intermediate chamber between the two orifices. It may be noted that for initial temperatures below 213 F the intermediate pressure is always 57.5 psi, being midway between the initial pressure, p_1, of 100 psi, line 1, and the discharge pressure, p_2, of 15 psi, line 2. For any initial temperature above 213 F the intermediate pressure increases linearly with initial saturation pressure and is midway between initial total pressure and initial saturation pressure. This feature, it was pointed out in the former paper, constitutes the factor in the utilization of two orifices in series for controlling the flow of hot water. For initial temperatures very near the saturation temperature, the critical pressure is only slightly less than the saturation pressure with a correspondingly slight effect on the intermediate pressure.

In Fig. 2 are also indicated the liquid region of the first orifice, the liquid region of the second orifice, the region of flashing which occurs after the second orifice, and the corresponding pressure drops in the first orifice, in the second orifice, and beyond the second orifice.

It should be recalled that the foregoing analysis of flow under stable conditions applies equally well to sharp-edged and rounded-entrance orifices.

DESCRIPTION OF APPARATUS AND EXPERIMENTAL METHODS

The apparatus used in this investigation, Fig. 3, consists essentially of a continuous hot-water supply leading to two identical orifices arranged in series and a means of measuring the following quantities: Pressure of water at the entrance to the first orifice; temperature of water at the entrance to the first orifice; intermediate pressure; intermediate temperature; downstream pressure beyond the second orifice; and the rate of flow.

Hot water and steam were obtained from blowoff and steam connections, respectively, of a gas-fired high-pressure boiler (2) supplied with distilled water. Initial temperature control of water entering the apparatus was obtained by maintaining the boiler pressure considerably above the desired pressure and by proportioning flow from a direct boiler connection and a parallel connection through a water-cooled coil. Valves with by-passes served further to throttle the water to the pressure maintained in

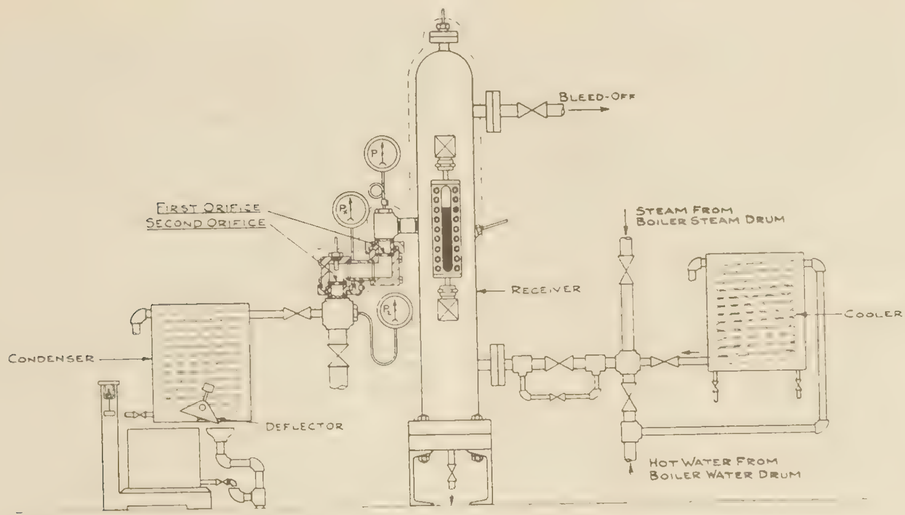


FIG. 3 ARRANGEMENT OF APPARATUS

the receiver. The column-type receiver was provided with water glass, thermometer well, upper and lower bleedoff connections, and an outlet leading to the orifices under investigation. Pressure gages served to indicate the initial, intermediate, and discharge pressures. The orifices, clamped between gaskets in union-type fittings, were purposely offset to eliminate velocity effects of the first orifice upon the second orifice and to return the enthalpy of the fluid in the intermediate chamber to its initial value.

The intermediate chamber between the orifices was provided with a thermometer well and a pressure gage. On the downstream side of the orifices a connection was provided through the gate valve to the water-cooled condenser which served to cool the discharged fluid preparatory to weighing. A deflector provided means of diverting the flow from the drain connection to the weighing tank on a platform scale, so that it was possible to measure accurately by a stop watch the rate of flow over a period of time.

The operating procedure consisted of establishing steady flow through the orifices at the desired initial temperature and pressure. The initial pressure was maintained at 100 psia for all experiments. The initial temperature was varied from 70 F to the saturation temperature of 327.8 F. Discharge pressure was maintained at approximately 15 psia.

Flow determinations were based upon time intervals ranging from 1 to 4 min. The stop watch was checked against a watch of known accuracy. The platform scale was sensitive to $^1/_2$ oz, approximately equivalent to 0.1 per cent of the average quantity of water weighed. At the beginning and end of each interval, observations were taken of initial pressure, initial temperature, intermediate-chamber pressure and temperature, and discharge pressure.

Pressure gages were of the standard Bourdon-tube type and were frequently checked by a dead-weight tester during the course of the investigation. Temperature readings were obtained by means of accurately calibrated mercury-in-glass thermometers inserted in mercury-filled thermometer wells.

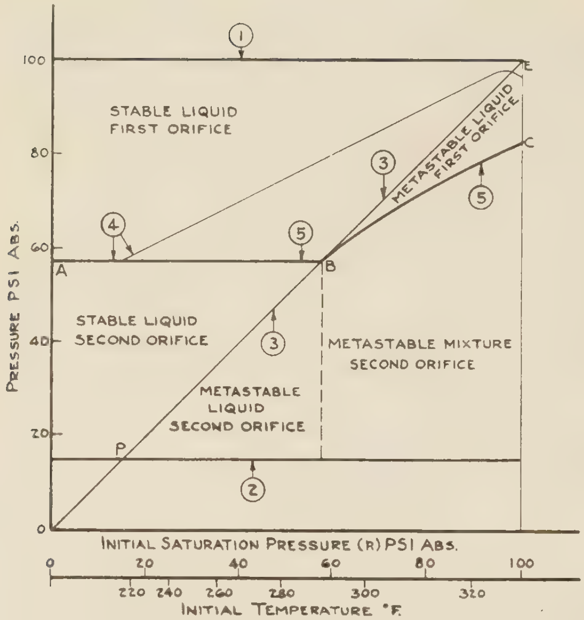
The Metastable Condition

Before presenting the results of this investigation, a word of explanation is given concerning stable and metastable conditions of water. By definition, "stable" refers to the condition in which the pressure of the liquid, or of the mixture of liquid and vapor, is not less than the saturation pressure corresponding to the existing temperature, or, conversely, that the temperature of the liquid or the mixture is never higher than the saturation temperature corresponding to the existing pressure. If a condition exists where the pressure of a liquid, or of a liquid-and-vapor mixture, is less than the saturation pressure corresponding to the existing temperature, the condition is said to be "metastable." The most important characteristic of the metastable state is the delay in an anticipated phase change. Metastable states are known to exist in many circumstances where a change of phase is about to occur (3). Examples are the cooling of liquid water below the freezing temperature and the cooling of steam below the condensation temperature; the latter condition occurs in the well-known supersaturation phenomenon which exists in steam flow through nozzles.

The metastable state of a liquid may be attained in several ways, all of which involve the phenomenon that bubbles of vapor do not form in the mass of liquid when temperatures exceed the saturation temperature corresponding to the existing pressure of the liquid, or, alternately stated, when pressures become lower than the saturation pressure corresponding to the existing temperature of the liquid. Some of the means of attaining a metastable state in a liquid are by the careful heating of water at constant pressure; by the sudden release of the pressure of a quiescent liquid; and by the flow of a liquid without turbulence through an orifice or nozzle to a discharge pressure less than the saturation pressure of the liquid. The retardation of the formation of bubbles has been attributed to the time necessary for the mass and heat transfers which are required for a bubble formation.

A more basic explanation of the metastable state depends upon the phenomenon of surface tension and the role of surface energy in delaying the initiation of very small steam bubbles in a mass of pure liquid (3). It is supposed that the formation of the initially minute bubbles is accelerated by impurities in the liquid, gases, or solids in solution, turbulence, ionization, or any other factor which will assist in creating an initially minute curved surface or nucleus about which the bubble may form.

In the flow of water through orifices it is known that the attainment of metastable states is favored by the purity of the water and by the smoothness of the passage, and also by a minimum of turbulence and friction. The existence of metastable states in the flow of water through orifices has been established by a number of investigators (4, 5, 7, 8).



1 Constant initial total pressure
2 Constant discharge-region pressure
3 Initial saturation pressure
4 Intermediate pressure for stable condition
5 Intermediate pressure for metastable condition

Fig. 4 Pressure Diagram for Flow Through Two Sharp-Edged Orifices in Series

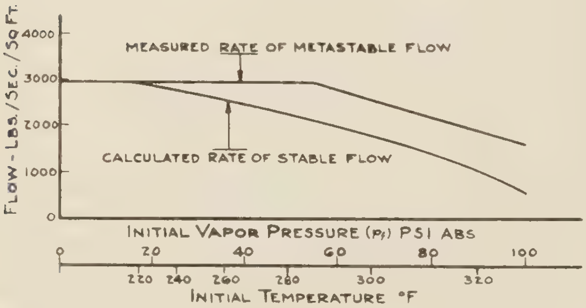


Fig. 5 Rates of Stable and Metastable Flow; Sharp-Edged Orifices

Flow of Hot Water Under Metastable Conditions Through Two Sharp-Edged Orifices in Series

The characteristics of flow of hot water through orifices in series will vary greatly with the type of orifice used. The sharp-edged orifice was selected as a standard for determining the basic phenomena of the process. Two sharp-edged orifices $^1/_8$ in. diam, $^1/_8$ in. thick, and chamfered at 45 deg to a working edge of 0.020 in. were arranged in series in the apparatus shown in Fig. 3. The essential data yielded by a run consisted of the initial pressure, the initial temperature, the intermediate pressure, and the measured rate of flow through the orifices.

Results for runs over the entire range of initial temperatures are presented in Figs. 4 and 5. In the pressure diagram, Fig. 4, the abscissa is the initial saturation pressure. The measured temperatures of the inlet water are also marked along the abscissa. Ordinates represent pressures existing throughout the system for various inlet temperatures. As in Fig. 2, line 1 represents the constant initial pressure of 100 psi, line 2 represents the constant discharge-region pressure of 15 psi, and line 3 is the initial saturation pressure. The measured intermediate pressures are shown by line 5. Line 4, which represents intermediate pressures for flow under stable conditions, is redrawn from Fig. 2 for purposes of comparing characteristics. In Fig. 5 are plotted the rates of flow as measured and, for comparison, the rates as calculated for ideal stable fluid flow. The most significant results of this investigation, as disclosed by an examination of Figs. 4 and 5, are (*a*) that the intermediate-chamber pressure, as calculated for stable fluid flow, line 4, is much higher than the measured intermediate pressure, line 5, and (*b*) that the calculated rate of stable fluid flow is much lower than the actual measured rate of flow. These striking differences in the measured characteristics and the similar characteristics, as calculated for stable fluid flow, can be accounted for completely and uniquely by recognizing the existence of the metastable state of the fluid in certain regions during the passage of the hot water through the orifices.

In stable fluid flow, the lowest pressure which can exist at the end of the second orifice is the saturation pressure, as shown by line 3 in Fig. 4. In actual flow, under certain conditions, the water continues through an orifice as a liquid in the metastable condition, that is, at pressures lower than the saturation pressure, without flashing, and with temperatures higher than saturation temperatures. In order definitely to establish the regions in which metastable conditions exist, a thermodynamic analysis is made in the following paragraphs.

Thermodynamic Analysis of Metastable Flow

It may be established quite directly from the pressure diagram in Fig. 4 that the metastable condition must exist in the second orifice for initial temperatures of water at least up to the point *B* because equal pressure drops and equal flow in each orifice can be accounted for only by the maintenance of a liquid condition in the second orifice along the entire line *A-B*. If flashing occurred in the second orifice, as would be required for the stable condition, a greater pressure drop would be required in the second orifice than in the first orifice. However, in the region from *B* to *C* it is not immediately evident to what extent metastable conditions exist, since a greater pressure drop exists in the second orifice than in the first. In order to investigate the existence of the metastable state, a complete thermodynamic analysis was made of the flow of initially saturated water at 100 psi and 327.8 F by assuming in turn that the fluid was stable in both orifices, metastable in the first orifice only, and finally metastable in both orifices. By comparing the results of this analysis with the data obtained in the experiment in regard to intermediate pressure and rate of flow, it was definitely established that the metastable condition existed in the first orifice in the region bounded by *EBC* in Fig. 4. The metastable condition also existed in the second orifice in the entire region below the line *PBC*.

The results of this thermodynamic analysis of the flow of saturated water through two sharp-edged orifices in series under both stable and metastable conditions are shown graphically in Fig. 6. The intermediate pressure is the measured value of 84 psia. The calculations for stable conditions were made on the basis of constant entropy during flow. Calculations for flow through the first orifice were made for quality, specific volume, change of enthalpy, velocity, and mass rate of flow per unit area. All calculations were by usual methods, except that calculations for the change of enthalpy were made by numerical integration of $\int VdP$ (ref. 1). The state in the intermediate chamber was known because in the throttling process H_1 equals H_x.

Calculations for flow through the second orifice, assuming constant entropy and stable expansion, were made in the same manner as for the first orifice. These results, plotted in Fig. 6, disclosed the usual critical pressures and the corresponding possible maximum rates of flow through each orifice. The maximum

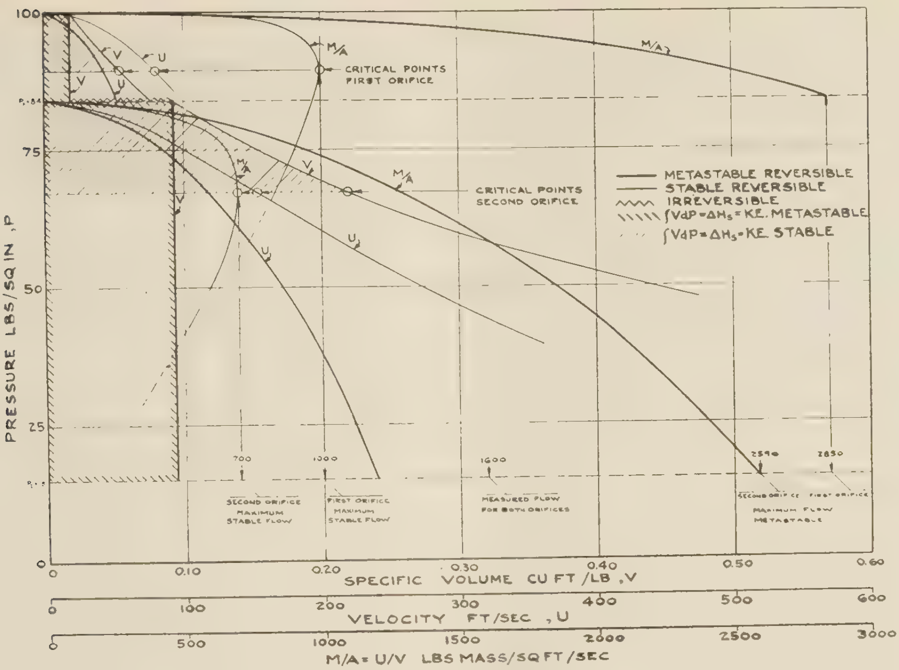


Fig. 6 Thermodynamic Analysis of Flow of Saturated Water Through Two Sharp-Edged Orifices in Series

calculated rates of flow obtainable under stable conditions were each much lower than the actual measured rates of flow. This phenomenon of actual rates of flow in excess of ideal rates of flow is characteristic of the metastable state.

Calculations were next performed assuming the existence of metastable conditions. For these calculations it was assumed that the fluid passed through the first orifice as a liquid with no change in volume from the initial volume of the saturated liquid. Further assumptions were that in the intermediate chamber stable conditions were restored, flashing occurred, and the enthalpy, H_x, returned to the original enthalpy, H_1. For the metastable condition in the second orifice it was assumed that the fluid, consisting of liquid and vapor bubbles, passed completely through the second orifice with no change in volume, no further evaporation, and no decrease in temperature. Calculations of enthalpy change, velocity, and mass rate of flow were thus made for metastable conditions in each orifice. The calculated rates of flow for metastable conditions were 2850 lb per sec per sq ft for the first orifice, and 2590 lb per sec per sq ft for the second orifice. These calculated rates of flow exceeded the measured rate of flow for each orifice, which was 1600 lb per sec per sq ft. The flow coefficients for the first and second orifice were 0.562 and 0.618, respectively. These coefficients, although not equal, were both acceptable for fluid flow through sharp-edged orifices. This circumstance, contrasted with the similar circumstance for previously assumed stable condition, established the condition of the flow as metastable or nearly metastable, in both orifices.

Similar thermodynamic analyses showed that, as previously anticipated in Fig. 4, the metastable condition existed in the first orifice in the region bounded by *EBC* and in the second orifice in the entire region below the line *PBC* which extends completely down to the discharge pressure. Even with some vapor present at the entrance to the second orifice, further evaporation was suppressed by the persistence of the metastable state.

It is noted that in the second orifice the liquid-vapor mixture attained a supertemperature of 102.4 F, which is the excess of

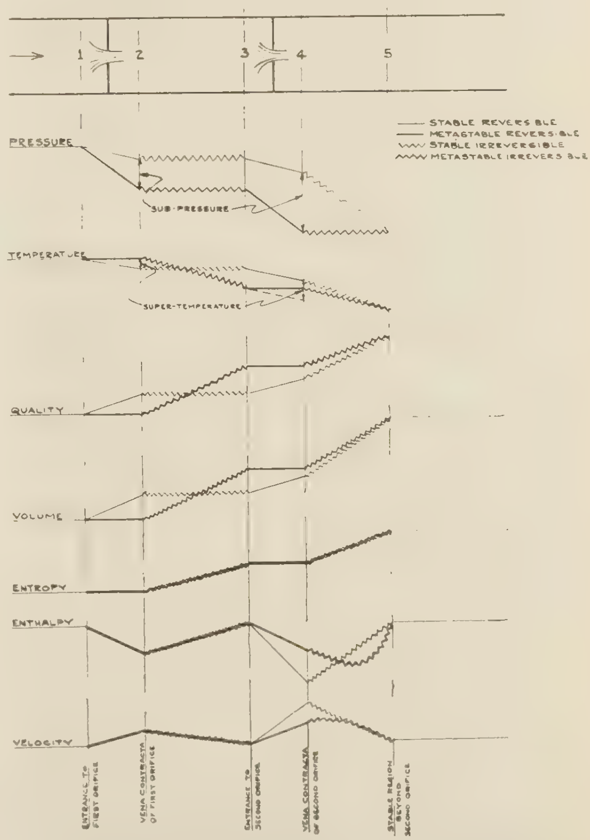


Fig. 7 Changes in Thermodynamic Properties for Flow of Saturated Water Through Two Orifices in Series

the actual temperature of 315.4 F over the saturation temperature of 213.0 F.

Qualitative Changes in Properties of Fluid During Flow

Comparisons of progressive changes in the properties of the fluid as it passes through the two orifices in series under stable and metastable flow conditions are portrayed graphically, but not to scale, in Fig. 7. Pressure, temperature, quality, volume, entropy, enthalpy, and velocity changes of the fluid throughout the system for stable and metastable conditions of initially saturated water are indicated at five significant reference points, viz., 1, entrance to first orifice, 2, vena contracta of first orifice, 3, entrance to second orifice, 4, vena contracta of second orifice, and 5, stable discharge region beyond the second orifice. It should be noted that in the region from orifice entrance to vena contracta the metastable state persists, and that immediately beyond the vena contracta there is an irreversible turbulent return of the fluid to a stable condition.

Physical Appearance of Jet of Hot Water Discharging From Sharp-Edged Orifice Into Atmosphere

In order further to verify the existence of the metastable state during flow, photographs were taken of jets of hot water discharging vertically downward from a sharp-edged $^1/_8$-in-diam orifice into the atmosphere. The initial pressure of the water was 100 psia, and temperatures were varied from 155 F to saturation temperature of 327.8 F. The apparatus used to produce the jet is shown in Fig. 8. The views, Fig. 9, are arranged progressively with increasing initial temperatures from 155 F to 327.8 F.

With cool water supplied, the jet was clear and of constant area and shape to a distance of 10 in. or more from the orifice. This form of jet persisted with increasing temperature but vapor was emitted from the surface of the stream as the temperature approached the boiling point. Fig. 9(*a*) for a temperature of 155 F, is characteristic of the appearance of the jet at all temperatures below 212 F. At 212 F there was no indication of boiling within the observed length of jet, but vapor emission from the surface of the stream was more pronounced. This vapor emitted from the jet was hot to the touch, and at temperatures somewhat in excess of 212 F the vapor emission was of noticeable intensity although the jet remained clear, an indication that there was no boiling in the interior of the stream.

With increase in temperature to about 225 F, Fig. 9(*b*), the vapor emission increased and the jet started to stream into filaments at a point about 12 in. from the orifice. Further increase in temperature caused a greater degree of streaming and spreading of the jet closer to the orifice, as shown in Fig. 9(*c*), for a temperature of 260 F.

The spreading of the stream assumed the conical form at a temperature of 275 F, indicated in Fig. 9(*d*). As the temperature increased, a transition point was reached at about 280 F where the streaming previously referred to changed to droplet formation. In contrast to the rather smooth and uniformly increasing streaming action, droplet formation proceeded explosively from a clear jet at a point 2 or 3 in. beyond the end of the orifice. Fig. 9(*e*) shows the form of the jet at a temperature of 285 F, slightly above this transition point, in contrast to Fig. 9(*d*) taken at a temperature just below the transition point. At a temperature of 300 F, Fig. 9(*f*), the average distance from the throat section of the orifice to the point of jet expansion was about 1 in. The discharge cooled rapidly in the region beyond the point of jet expansion and was relatively cool to the touch beyond the region of full expansion. The point of jet expansion did not remain at a fixed distance from the orifice, but moved up and down the jet over a range of possibly $^3/_4$ in. with a crackling explosive sound.

Figs. 9(*g*) and 9(*h*) show the decreasing length of clear jet at temperatures of 310 F and 322 F, respectively. The transition point where marked jet expansion occurred was taken to be indicative of transition from the metastable to the stable state. If the region beyond the transition point assumed a conical form, it was presumed that the fluid consisted of a continuous liquid phase containing bubbles of vapor. If at the transition point the expansion was marked and explosive, it was presumed that the subsequent fluid consisted of a continuous vapor phase with entrained drops of liquid.

In the case of saturated water supplied to the orifice, Fig. 9(*i*), an explosive transition was definite, the change occurring quite near the orifice.

With steam flow, Fig. 9(*j*), the issuing jet assumed approximate conical form within a very short distance from the orifice. Because the jet was almost invisible for initially dry saturated steam a trace of moisture was introduced to outline the jet. The form of the jet did not appear to be visibly affected by this moisture addition.

These observations verify the existence of the metastable condition of hot water during flow through, and somewhat beyond, the sharp-edged orifice. The distance beyond the orifice marked by the existence of the metastable state appears to be an inverse function of the temperature of the water. For the jet of water at a temperature of 225 F (a supertemperature of 12 deg) the jet persists for a distance of 12 in. For the jet of water at a temperature of 322 F (a supertemperature of 107 deg) the metastable jet persists only for a fraction of an inch. Thus the time of existence of metastability is an inverse function of the supertemperature.

Flow Characteristics of Rounded-Entrance Orifices

The maintenance of the metastable condition in both sharp-edged orifices down to the vena contracta is attributed to the almost completely reversible (frictionless and nonturbulent) flow produced by the sharp-edged orifice. Anything which produces friction or turbulence will prevent the maintenance of the metastable state during flow. A rounded-entrance orifice introduces a much larger surface of contact than the sharp-edged orifice, and to this extent may produce conditions departing from the metastable. Investigations were conducted with various pairs of identical rounded-entrance orifices in series. The radius of entrance was $^1/_8$ in. and the minimum diameter about $^1/_8$ in. The lengths

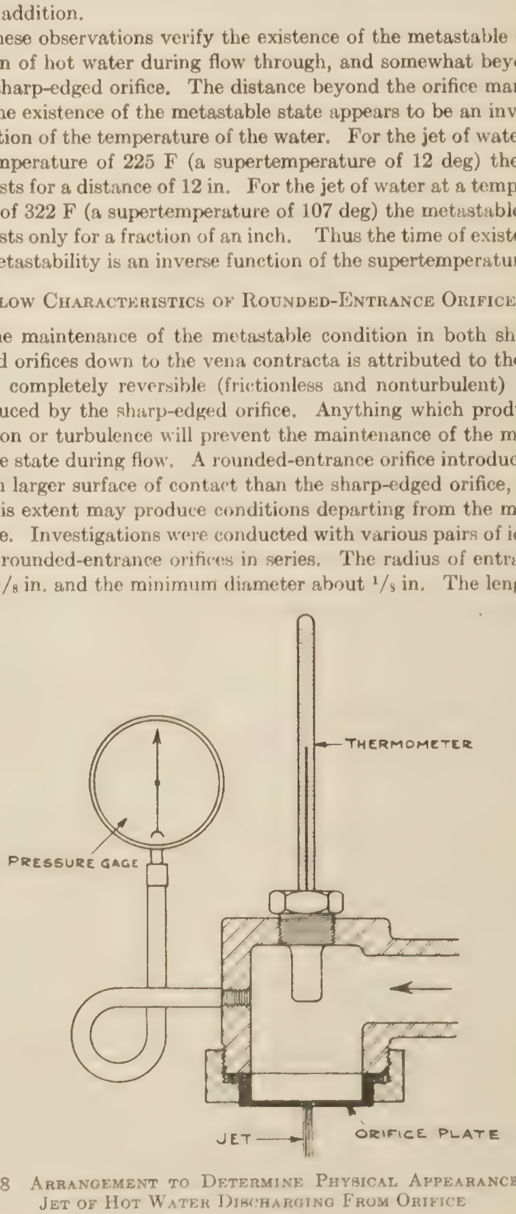
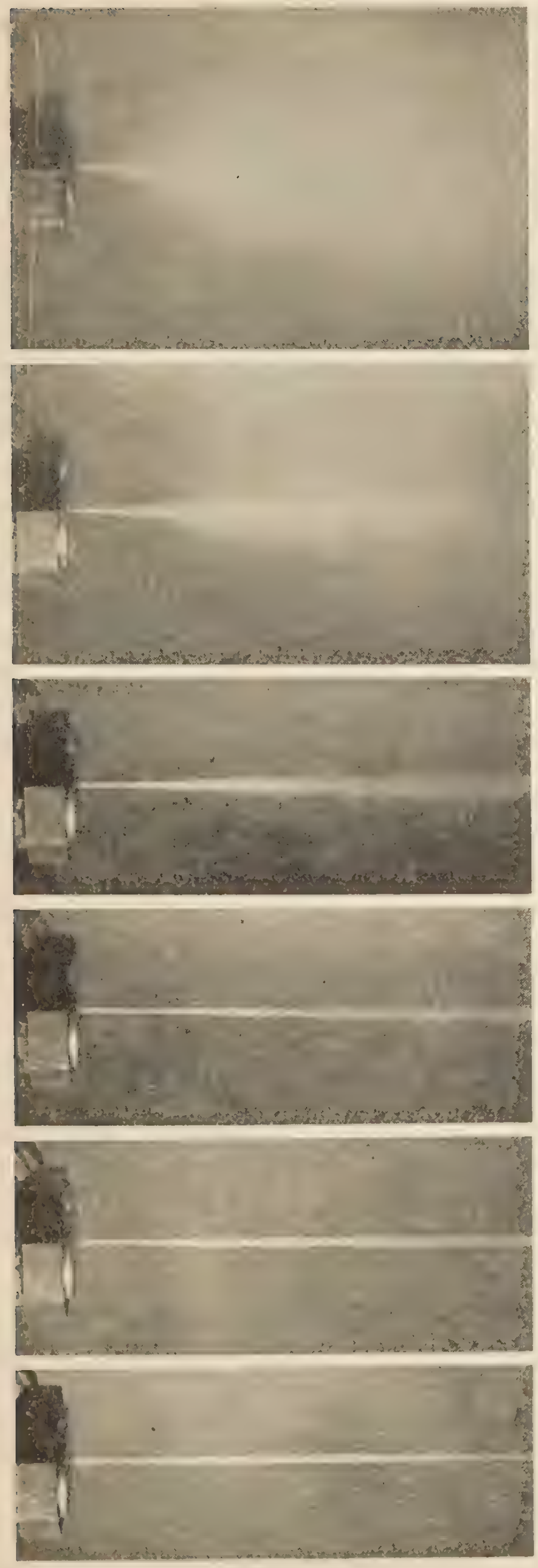


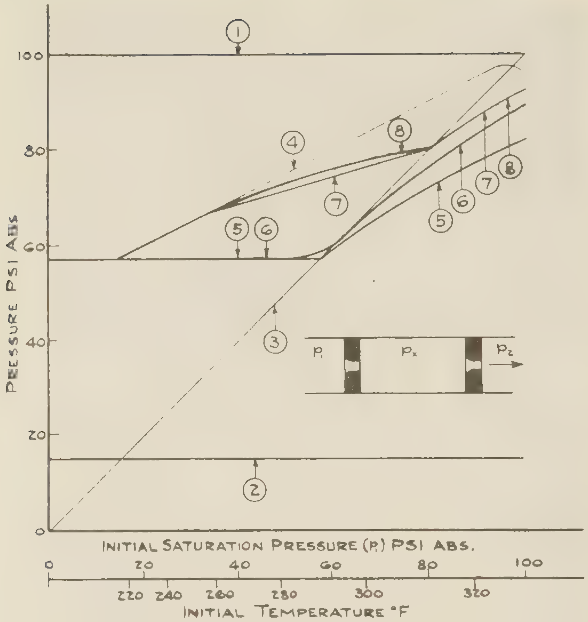
Fig. 8 Arrangement to Determine Physical Appearance of Jet of Hot Water Discharging From Orifice



(a) Initial temperature 155 F (b) Initial temperature [illegible] F (c) Initial temperature 260 F (d) Initial temperature 275 F (e) Initial temperature 285 F (f) Initial temperature 300 F

(g) Initi[illegible] (h) Initial [illegible] 322 F (i) Initial temperature 327.8 F saturated (j) Steam 327.8 F

Fig. 9 Jet of Hot Water Discharging Through a Sharp-Edged Orifice From Initial Pressures of 100 Psia Into [illegible]



1 Constant initial total pressure
2 Constant discharge-region pressure
3 Initial saturation pressure
4 Stable flow intermediate pressure
5 Metastable flow intermediate pressure for sharp-edged orifices
6 Metastable flow intermediate pressure for rounded-entrance orifices, 1 diam long
7 Metastable flow intermediate pressure for rounded-entrance orifices, 5 diam long
8 Metastable flow intermediate pressure for rounded-entrance orifices, 8 diam long

Fig. 10 Pressure Diagram for Flow of Hot Water Through Rounded-Entrance Orifices in Series

including the straight parallel-sided portion and the rounded entrance were 1, 5, and 8 diam in each pair of experimental orifices. The results with these orifices, plotted as a pressure diagram in Fig. 10, show that the intermediate pressures attained with rounded-entrance orifices were between those for the ideal stable fluid flow, line 4, and the complete metastable fluid flow of the sharp-edged orifices, line 5. Line 6, for the 1-diam rounded-entrance orifice, shows an intermediate pressure similar to that for the sharp-edged orifice. It should be noted, however, that the intermediate pressure increases gradually before the saturation pressure reaches intermediate pressure. This, together with the observed reduction in flow, is indication that the point marking the start of the transformation from metastable liquid to the stable mixture had moved upstream and into the second orifice with increasing saturation pressure. This movement of the transformation point with respect to saturation pressure is illustrated in the views shown in Fig. 9. For flow through longer rounded-entrance orifices the state of the fluid approached a stable condition at the end of the orifice. This confirms the theory for stable flow as presented in the former paper by the authors (1) and agrees with the results of Benjamin and Miller in their paper on the flow of flashing mixtures of water and steam through pipes (6).

Flow Characteristics of Short Tubes

Experiments were made with the orifices in the form of standard short tubes with sharp-edged entrance, as shown in Fig. 11. A characteristic of a short tube, well known in hydraulics, is that a vena contracta forms in the tube and upon expansion causes the water to fill the tube at the exit. This fact introduces distinctive characteristics in the flow of hot water through short tubes. As shown in Fig. 11, the intermediate pressures for short tubes are somewhat similar to those for rounded-entrance orifices but differ in the important aspect that intermediate pressures obtained were higher than for orifices with rounded entrances and even higher than for stable flow with sharp-edged orifices.

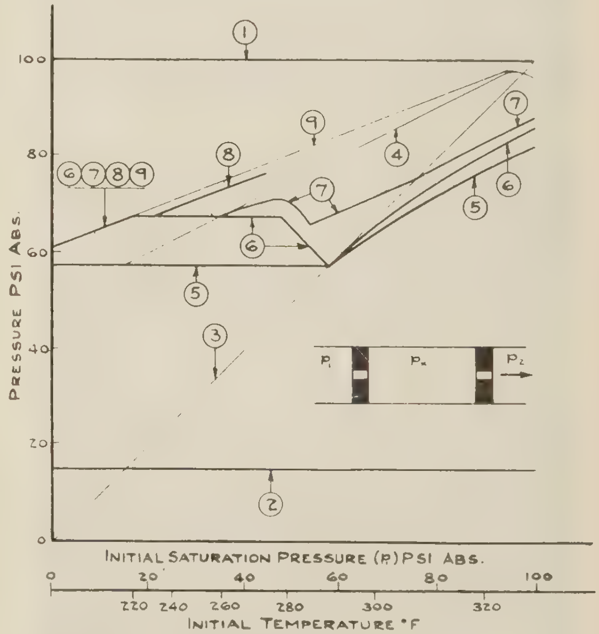
All of the data presented are based upon flow through two identical orifices in series. The combination of various types and differing sizes of orifices makes it possible to obtain desired intermediate-chamber pressure characteristics.

Control of Flow of Hot Water by Means of Orifices in Series

The characteristics of fluid flow through two orifices in series, as presented in this paper, may serve as a basis for the design of a control element. Such an element consists of an arrangement of two orifices in series, in which the state of the fluid supplied causes variations in pressure between the two orifices, these pressures being in turn communicated to a pressure-sensitive member. Fig. 12 shows the practical form of a two-orifice control element developed for automatic condensate drainage.

The piston valve, which constitutes the only moving part, fulfills the combined requirements of the series-control orifices, pressure-sensitive element, and drainage valve. The annular clearance between the disk on the piston valve and the wall of the cylinder forms the first orifice, the space in the cylinder above the disk constitutes the intermediate chamber, and a small rounded-entrance orifice at the top of the piston valve communicating with the discharge through the center of the valve constitutes the second orifice.

In order to obtain the required area relationship between the two orifices, the cylinder wall is tapered and the cylinder is made vertically adjustable to permit variation of the clearance between disk and cylinder wall.



1 Constant initial total pressure
2 Constant discharge-region pressure
3 Initial saturation pressure
4 Intermediate pressure for thin-plate orifices, stable state
5 Intermediate pressure for thin-plate orifices, metastable state
6 Intermediate pressure for short tube 1 diam long, metastable state
7 Intermediate pressure for short tube 4 diam long, metastable state
8 Intermediate pressure for short tube 8 diam long, metastable state
9 Intermediate pressure for short tube 1 diam long and over, stable state

Fig. 11 Pressure Diagram for Flow of Hot Water Through Short Tubes in Series

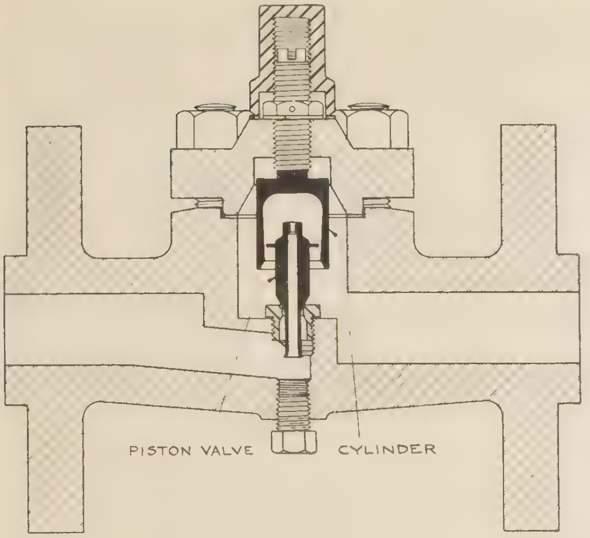


FIG. 12 PRACTICAL FORM OF CONTROL ELEMENT FOR CONDENSATE DRAINAGE

On proper adjustment, the relatively high intermediate pressure acting on the upper surface of the piston for steam and very hot condensate flow is sufficient to close the valve. With cooler condensate the correspondingly lower intermediate pressure permits the valve to lift and discharge the condensate.

The device is operative in like manner over a wide range of initial pressures without further adjustment. As suggested earlier in the paper, desired characteristics for the variation of the intermediate pressure with initial temperature may be obtained by suitable choice of types of orifices and by relative sizes of the orifices. Further, the existence of the metastable state, as disclosed in this paper, makes it possible to obtain a far wider range of control conditions than would be possible if the fluid remained in the stable state throughout the process.

ACKNOWLEDGMENTS

The authors acknowledge with appreciation the work of John F. McKee, engineer associated with the Yarnall-Waring Company, in the original development of the control element of fluid flow through two orifices in series.

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8 "Discharge Capacity of Traps," by A. E. Kittredge and E. S. Dougherty, *Combustion*, vol. 6, Sept., 1934, pp. 114–119.

Discussion

E. BROWN.[4] Study of the metastable state of water will probably help toward a better understanding of pitting in water turbines of the pressure type. As the water passes from the scroll case to the draft tube its pressure is lowered, and high vacuum may occur in the draft tube near the discharge tips of the blades. It is probable that in certain localized zones the vacuum is much higher than the general average vacuum across the whole section of the draft tube. In such zones the pressure may fall below the saturation pressure corresponding to the temperature of the water, because the drop of pressure is fairly rapid. The time element is probably an important factor in determining the rate at which small pockets or bubbles of water vapor, or of gases held in solution, are formed. The rapid collapse of such bubbles or voids may cause local pressures of great intensity which destroy the metal with which the water comes in contact. This was first demonstrated by the late Sir Charles Parsons in connection with the corrosion or pitting of ships' propellers. Whether this theory, or the alternative one of chemical action, be accepted, the fundamental conception is the formation of bubbles of released gases or water vapor. In water turbines such gases or vapor tend to be reabsorbed as the pressure in the water rises during its passage to the tailrace. The gradual formation of such voids, and their subsequent gradual disappearance, has been proved by photographs, and the writer has found experimental evidence of distinct differences between the gas content of water drawn from the draft tube of a small turbine from a zone where erosion was occurring, and a zone where there was no erosion. In both cases the gas content was less than that of water at the intake.

The writer has tested many models of water turbines during the past 20 years in connection with water-power developments. His experiences of corrosion or pitting have been varied, and in many cases seemingly contradictory. He has long held the view however that more intensive pitting may develop in a large turbine than in a small model even when they are operated on the same head. Under such conditions, the velocity of the water will be the same in both cases. But if the model is to, say, one-twelfth scale, the water in the large turbine will take 12 times as long as the water in the model to flow between corresponding points above and below the runner, and the harmful bubbles will probably be created at a different rate. Questions of heat transfer and surface tension are involved, and impurities in water may have some definite influence. The problem of relating the behavior of models to that of water turbines of greatly increased size is fraught with many difficulties. The work of the authors, apart from its evident value in such processes as those discussed in their paper, should contribute toward a clearer scientific understanding of some of the troubles of the engineer engaged in the development of water power.

G. A. HORNE.[5] This paper suggests some interesting points to the refrigerating engineer. In the refrigeration cycle, the liquid refrigerant is generally assumed to be in a stable condition throughout.

The writer has two-stage ammonia compression systems in

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[5] Consulting Engineer, Merchants Refrigerating Company, New York, N. Y. Mem. A.S.M.E.

which all the liquid passes from the condenser through Venturi meters for measurement. To insure any change of state from liquid to vapor at the throat of the meter, the liquid is sufficiently reduced below saturation temperature by means of a cooling coil placed between condenser and meter.

In these two-stage compression systems all of the liquid is passed through an expansion valve into an intermediate cooler at the intermediate pressure between the two stages of the compressor. In this operation, it is generally assumed that the refrigerant remains in a stable condition so that the amount of vapor formed is the anticipated quantity in accordance with the pressure drop. However, the liquid which accumulates in the intermediate cooler is in a quiescent state and is maintained at a certain level while the vapor mixes with the balance of the vapor from the low-pressure cylinder, thence to the high-pressure cylinder.

The liquid from the bottom of the intermediate receiver passes through an expansion valve into the low-temperature evaporator or brine cooler. A certain residual amount remains in the receiver to serve as a trap.

The question which is prompted by the authors' work on fluid flow is therefore: Under the operation just described, would a metastable condition of the liquid refrigerant be apt to exist in the intermediate receiver?

Obviously, if this were the case, the efficiency of the system would be lowered and would result in an increase in the horsepower to drive the compressors.

The authors refer to certain means of attaining a metastable state in a liquid. A few years ago the writer was interested to observe an example of this phenomenon in determining the freezing point of grapefruit juice. It had generally been assumed in the fruit trade that oranges and grapefruit should be carried in cold storage at temperatures varying from 34 to 38 F. It had been found that lower temperatures, as low as 30 F were of considerable advantage in lengthening the storage period and improving the quality of the products. Although certain authorities showed the freezing point of the juice to be around 29 F, we decided to check it.

A 1-in. test tube was filled with juice, inserted through a cork into a bottle, and the bottle immersed in a large vessel containing brine at —5 F. Through a cork in the test tube an accurate thermometer was placed. It was expected that crystals would appear when the temperature of the juice reached 28 or 29 F. However, nothing happened while the temperature gradually dropped to 19 F. At that point the writer stirred the juice a little and immediately the entire contents became a mass of grapefruit ice, the temperature rose to 28.6 F and stayed there for some time.

It seems probable that the greater subcooling of water and water solutions, as compared to other liquids, is partly due to the fact that the maximum density of water is several degrees above the freezing point. As the authors point out, the metastable state depends upon surface tension. The ice crystals must form at the surface instead of at the bottom.

J. H. Keenan.[6] This paper is an important contribution to our knowledge of the flow of liquids in the neighborhood of the saturation state. The literature on this subject, as indicated by the Bibliography included in the paper, is indeed limited.

As pointed out by the authors, the expansion of a liquid across its saturation line, like the flow of a vapor across its saturation line, results in metastable states. Various investigators from Wilson to Yellott[7] have attempted with a degree of success to determine the extent to which the expansion of a vapor may be extended beyond the saturation line before the first evidence of condensation appears.

This paper sheds some light on the analogous problem in the liquid phase, namely, to determine the extent to which the expansion of the liquid may be extended beyond the saturation line before the first evidence of vaporization appears. The photographs of Fig. 9 suggest that expansion to atmospheric pressure may occur at a temperature as high as 285 F without vaporization. This corresponds to an expansion of about 40 psi below the saturation pressure.

These illustrations, however, are not conclusive in this regard, because they are of such small scale that it would be impossible to detect the presence of small bubbles of vapor where the stream first attains atmospheric pressure. Somewhat better evidence appears in curve 6 of Fig. 10, which indicates that for rounded-entrance nozzles no evidence of vaporization appears for expansion to atmospheric pressure until the temperature exceeds 280 F. In a rounded-entrance nozzle, the pressure in the exit plane of the nozzle approximates the pressure in the exhaust space for a radius of rounding equal, as in this case, to the diameter of the nozzle. It appears, therefore, that expansion to approximately atmospheric pressure occurred in the downstream rounded-entrance nozzle without vaporization.

The rough agreement between the indications of the views in Fig. 9 and curve 6 in Fig. 10 does not support the statement of the authors that "a rounded-entrance orifice introduces a much larger surface of contact than the sharp-edged orifice, and to this extent may produce conditions departing from the metastable." The statement may be true, nevertheless.

In this connection, it should be noted that curve 5 in Fig. 10, which represents the performance of sharp-edged orifices, offers no such conclusive evidence as curve 6. In flow through a sharp-edged orifice the pressure in the plane of the orifice does not even approximate the pressure in the exhaust space but exceeds it by a large fraction of the total pressure drop. It would be entirely possible, therefore, to have considerable vaporization in the stream at pressures in excess of the exhaust pressure without any effect on the flow through the orifice. Thus the calculated curve 4 is relevant to the rounded-entrance orifice but not to the sharp-edged orifice.

The curves in Fig. 11, for flow through short tubes, show many curious characteristics. Perhaps the authors could suggest some explanation of the reversal in the effect of rising temperature on the intermediate pressure which appears in curves 6 and 7.

If the heavy lines in Figs. 4, 10, and 11 represent experimental values, as they appear to, that fact is somewhat obscured by the label "metastable flow." Labeling of curves should be factual rather than interpretive.

It is a temptation not to be resisted to point out that the so-called metastable expansion assumed in Fig. 6 for the second orifice is an adiabatic process of decreasing entropy.

In the absence of experimental points, some indication should be given of the reproducibility and the precision of the data.

L. K. Spink.[8] Referring to the diagrammatic arrangement of the apparatus, some of the pressure gages are shown connected with loops or syphons. The gage registering Px is connected with a self-draining connection. Bourdon-tube gages not only zero differently but also calibrate slightly differently when the tube is filled with a material of different density. The magnitude of the discrepancy is not sufficient to affect the reliability of this work, but is mentioned as a point of general interest.

The dimensions of the conduit between the two orifices are not

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[7] For references see "Thermodynamics," Bibliography (3) of the paper.

[8] The Foxboro Company, Foxboro, Mass.

given but appear sufficient to reduce the velocity of approach to a negligible value. Only in that case would the location of the taps and the shape of the tap openings be of no importance. This low velocity of approach would also minimize the effect of the disturbed flow condition set up by the two right-angle turns in the conduit.

Although it is natural to expect a variation in the discharge coefficients of nearly identical orifices as small as $^1/_8$ in., the value 0.562 appears abnormally low. The orifice thickness is slightly in excess of that generally accepted for thin-plate orifices for flow measurement; but this, and the small diameter, would tend to increase the coefficient. If both values were true discharge coefficients, an intermediate pressure midway between the initial pressure and the discharge pressure, as shown by line *A-B* in Fig. 4, would be difficult to explain. The exact significance of the coefficient of the second orifice is obscured by the assumption that the vapor bubbles undergo a change in pressure without any change in volume.

Presumably the authors have a reason for assuming that the vapor does not expand in passing through the second orifice, and have additional data or a theory to explain the apparent inconsistency of the coefficients. Further elaboration on these points would be appreciated.

P. W. Swain.[9] This paper, like its predecessor, published in 1936, interests the writer chiefly because of the directness of its application to commercial power equipment.

In recent years American engineers have advanced notably in their ability and will to analyze phenomena in terms of the abstractions of mechanics and thermodynamics. Of this trend, this paper is a good example. However, it would seem that too many of the engineer researchers and analysts have been happy to stop with a published report and analysis. This the writer could never understand, for it would seem that the whole life of the true engineer is so wrapped up in the building and operation of machines that abstractions, even experiment and experimental data, could never satisfy him, except as a means to some practical end.

[9] Editor, *Power*, New York, N. Y. Mem. A.S.M.E.

The condensate drainer, pictured in Fig. 12, gives everyday meaning to the research and analysis of the authors. The writer sincerely hopes that it evidences an increasing interest of scientifically minded engineers in practical applications. It is possible that this interest in application largely explains the immense productivity of American engineering as compared, for example, with that of prewar France, a country gifted with many fine and highly trained technical minds, yet singularly sterile in engineering results.

Authors' Closure

The discussions pertain chiefly to metastable phenomena rather than directly to the characteristics of flow through series orifices. Most significant are the remarks of Professor Brown in regard to the influence of the metastable state in promoting cavitation in water turbines, and the instances cited by Mr. Horne where the possibility of metastable phenomena are encountered in refrigeration.

The authors accept the criticisms of both Professor Keenan and Mr. Spink that the assumption of a truly constant volume process in the second orifice is undoubtedly too dogmatic. An assumption more nearly in agreement with the facts is that the vapor bubbles in the mixture increase in volume adiabatically, with perhaps a slight increment due to evaporation from the metastable liquid portion of the mixture.

The reversal in the effect of the rising temperature on the intermediate pressure for flow through short tubes, referred to by Professor Keenan, is attributed to the limitation of the minimum pressures which may exist in the vena contracta as within the short tubes. When a short tube is flowing full at the exit, the pressure at the vena contracta within the tube does not become less than the saturation pressure.

Mr. Swain correctly surmises that it was the embodiment of a scientific principle in a practical commercial device which prompted the authors to undertake these experiments, intended both to explain the operation of the device and to reveal further basic information on the principles involved.

The Combustion of Barley Anthracite

By ALLEN J. JOHNSON,[1] PHILADELPHIA, PA.

Because of its availability, excellent combustion qualities, and low cost, "barley" or No. 3 buckwheat anthracite is one of the most economical of all industrial fuels. After giving the analyses and physical properties of this fuel, the author discusses the effect of utilizing barley upon boiler performance. Details of the equipment available for burning this fuel, either by hand or stoker firing, are outlined, as well as practical suggestions for actual operation of the various types.

Preparation, Quality, and Characteristics

The 1942 production of "barley" or No. 3 buckwheat amounted to 5,800,000 tons which represented 10 per cent of the total production of anthracite. Interest in its industrial use centers around the combined facts that it is an excellent fuel for both hand and stoker firing, and that it is one of the most economical of all industrial fuels. At the present time, barley is one of the few fuels, if not actually the only one, of which there is a substantial surplus at the mines with adequate transportation facilities.

Adequate Supply Available. Because of the greatly increased demand for the larger or domestic sizes of coal without a corresponding industrial absorption, the production of barley now exceeds consumption by several thousand tons a week, thus creating substantial stock piles, available without fear of interruption to all plants having suitable utilization equipment. Sizing specifications, analyses, and physical properties of barley anthracite are given in Tables 1 and 2, respectively.

Quality of Barley. Like many other valuable commodities, barley originates as a by-product prepared from the breakage of the larger sizes of anthracite. With this accidental beginning, all resemblance to a by-product ceases, as the modern concentrator tables and other equipment upon which barley is cleaned are capable of preparing it to a high quality with any reasonable specifications of size and ash content. In fact, the fineness of the coal particles actually facilitates the removal of impurities. However, because of its small size and the consequent limitation of use to plants having specific combustion equipment, the price of barley is low, as shown in Table 3.

Effect of Coal Size and Quality Upon Boiler Performance

Several characteristics of coal affect the efficiency and capacity of boilers. However, in comparing the properties of coals, the terms efficiency, capacity, and economy are likely to be confused. Increased efficiency from a quality fuel may not overbalance its increased cost in one plant; while in another, increased capacity may be of sufficient importance to justify a sacrifice of efficiency and even of economy. The following factors should thus be carefully considered.

When to Use Barley. Barley anthracite is the smallest and cheapest size commonly used in all but the largest (traveling-grate) plants. Many plants now using rice (No. 2 buckwheat), or even No. 1 buckwheat would effect substantial savings by changing to this size, with the important advantage of assurance of supply.

Even where excessive loads in existing equipment preclude the continuous use of barley, substantial economies may be effected by burning it during off-peak seasons, as for example during mild weather in heating plants.

In general the following are requisite to the use of barley:

(*a*) Adequate forced draft or natural draft from unusually high stacks (preferably 300 ft or more as in large hotels or office buildings).

TABLE 1 STANDARD SIZING SPECIFICATIONS OF BARLEY ANTHRACITE AT BREAKER[a]

Through round-mesh screen in.	3/16
Over round-mesh screen, in.	3/32
Oversize maximum, per cent	10
Undersize maximum, per cent	20
Undersize minimum, per cent	10

[a] A breaker is the preparation plant at the mines.

TABLE 2 ANALYSES AND PHYSICAL PROPERTIES OF BARLEY ANTHRACITE

	Approximate ranges	Average of 41 breakers[a]
Proximate analyses, per cent:		
Volatile matter	3.0 to 8.0	4.4
Fixed carbon	77 to 86	82.1
Ash	9.5 to 17.5	13.6
Sulphur	0.5 to 1.2	0.7
Fixed carbon dry mineral matter, free basis, per cent	92 to 98	...
Heat value Btu, dry basis	12000 to 13600	12890
Moisture (as received), per cent	5 to 7	7.2[b]
Softening temperature of ash, deg F	2400 to 3000	2800
Approximate ignition temperature, deg F	525 to 775	...
Weight and volume:		
Specific gravity	1.45 to 1.75	...
Pounds per cu ft	50 to 60	...
Cu ft occupied by 1 ton	33 to 40	...
Tendency to spontaneous combustion	None	None
Grates recommended:		
Proper type	Stationary or dumping	..
Max size of air opening	3/16 to 3/32	...
Free air space per cent	4 to 10	...
Combustion rate (lb per sq ft per hr):		
Hand-fired	15 to 20	.
Hand or "semi" stokers	18 to 23	.
Traveling-grate stokers	20 to 40	..
Availability:		
Tons mined in 1942	5,800,000	..
Percentage of total breaker production	10.0	..
Approximate f.o.b. mines: Cost (present O.P.A. ceiling)	$2.50	...

[a] Based upon U. S. Bureau of Mines Report R. I. 3283, 1935, analyzing 268 samples from 41 breakers mining 50 per cent of all anthracite.
[b] Measured at breaker.

TABLE 3 FREIGHT AND APPROXIMATE COST DATA ON BARLEY ANTHRACITE

	Typical f.o.b. mines cost	Freight rate[a]	Total carload cost on railroad sidings per net ton[b]
Harrisburg, Pa.	$2.50	$1.45	$3.95
Lancaster, Pa.	2.50	1.54	4.04
Philadelphia, Pa.	2.50	2.05	4.55
Wilmington, Del.	2.50	2.23	4.73
Albany, N. Y.	2.50	2.27	4.77
New York, N. Y.[b]	2.50	2.42	4.92
Rochester, N. Y.	2.50	2.50	5.00
Baltimore, Md.	2.50	2.50	5.00
Washington, D. C.	2.50	2.50	5.00
Buffalo, N. Y.	2.50	2.61	5.11
New Haven, Conn.	2.50	3.08	5.58
Springfield, Mass.	2.50	3.18	5.68
Boston, Mass.	2.50	3.18	5.68

[a] Calculated to net-ton basis, and also includes 4 cents per net ton Federal tax.
[b] Add 8 to 15 cents per ton for points outside harbor limits.

[1] Mechanical Engineer, Anthracite Industries, Inc. Mem. A.S.M.E.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

(*b*) Stationary or dumping grates having a mesh of not over $^3/_{16}$ in.

(*c*) Semi- or hand stokers, with grates as described in (*b*).

(*d*) Traveling-grate stokers.

(*e*) Proper arch and furnace design when stokers or semi-stokers are used.

(*f*) Properly proportioned grate areas.

The use of straight barley anthracite upon underfeed stokers is not often recommended, unless the load is considerably below boiler and stoker rating, because of reduced effective grate areas available. However, mixtures of barley and bituminous are in use very successfully upon underfeed stokers.

Effect of Btu Value. The heat value or Btu content of coal is of course a direct measure of its value. In addition, since freight rates are on a per-ton basis, it costs more, per pound of useful combustible, to transport low Btu fuel. In considering this point, it is noteworthy that the further the plant is located from the mines the more significant coal quality becomes, because of the greater effect of freight rate.

Boiler ratings are also in direct proportion to Btu because, for any given number of pounds of coal per hour, fewer pounds of combustible will be consumed, as the percentage of inert material or ash increases. In plant design, however, this point can be overcome to permit the use of coal with practically any Btu content by adding grate surface. In this connection, the additional cost of the grates is usually insignificant, as compared with fuel savings over the life of the equipment.

Boiler efficiency is not always proportionately affected by heat value especially if the plant has been designed with the particular coal in mind. It is entirely possible to obtain "relatively" high efficiencies with low-grade coals even though the design problems are materially increased.

Effect of Ash. In anthracite, heat value is inversely proportional to ash content, Fig. 1. The effects of high ash upon economy, ratings, and efficiency are therefore as described for low heat content.

With respect to efficiency, Fig. 2 shows that, in a representative test over the entire range of ash contents, there was a drop of only 2 per cent in over-all boiler efficiency. Thus if the plant has been designed with sufficient grate surface to compensate for the lower outputs of high-ash coal it should operate satisfactorily, although never at the ratings obtainable with better fuel.

In all cases, the cost of handling and removing ash should be added to the price of coal plus the freight rate, in making cost comparisons. The best system consists of placing all fireroom labor, fuel and ash costs, and handling charges on a basis of cost per million Btu. Direct comparisons may then be made, which will place costs on a basis of the heat actually received rather than the relatively meaningless price of Btu at the mines.

Effect of Undersize. Fig. 3, based upon tests by L. A. Stenger (1),[2] shows the rate of air flow through coals having different percentages of dust content. The shape of this curve illustrates the well-known fact that uniformity of sizing is one of the most important points in connection with the combustion of barley. With more than 20 per cent undersize, the resistance of the fuel bed to the flow of air rises so rapidly as to change completely all combustion characteristics. It is thus significant that the Anthracite Institute Committee on Standards of Sizing selected 20 per cent as the maximum allowable undersize for barley. Conversely, however, 10 per cent was set as a minimum limit of undersize as it was not considered practical to prepare coal to finer standards.

With anthracite, the effect of dust is severalfold. It is difficult to maintain a uniform fuel bed because of a segregation of

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

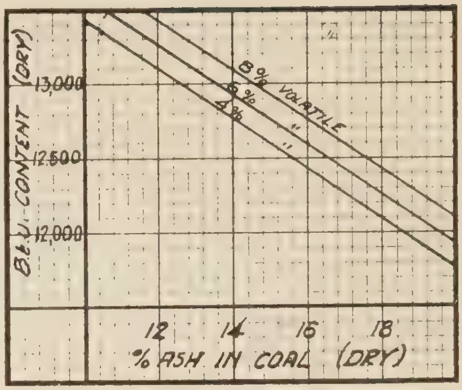

Fig. 1 Btu Versus Ash Content for Anthracite

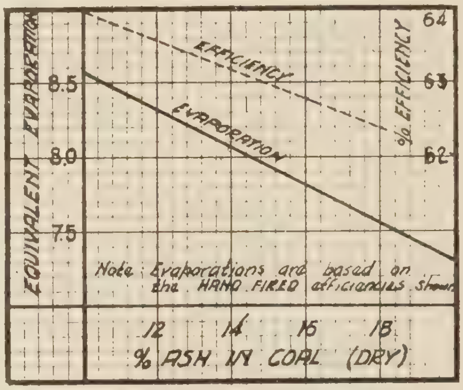

Fig. 2 Effect of Ash on Performance

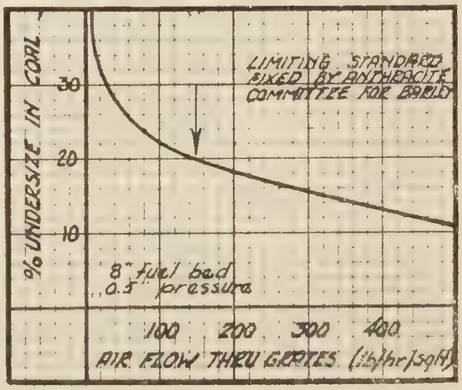

Fig. 3 Reduction of Air Flow to Undersize

fines and the consequent blowing of holes; and the fines impede the flow of air to increase any tendency toward clinker and reduce combustion rates. In addition, fly ash and siftings are usually in proportion to the dust content.

In analyzing undersize content, one additional point is of particular importance, i.e., the character of the fines is as important as the percentage. Thus if a given percentage of undersize is almost all dust, serious difficulties may ensue; if, however, a large percentage of the fines is just slightly under the minimum screen size, considerably better results may be expected. It is thus suggested that sizing tests be reported on the following basis:

(*a*) Per cent larger than No. 3 buckwheat (over $^3/_{16}$ in. screen)
(*b*) Per cent of No. 3 buckwheat through ($^3/_{16}$ in. over $^3/_{32}$ in.)
(*c*) Per cent of No. 4 buckwheat through ($^3/_{32}$ in. over $^3/_{64}$ in.)
(*d*) Per cent smaller than No. 4 buckwheat (through $^3/_{64}$ in.)

This method of analysis will show whether the undersize is really dust, or whether it is merely a mixture of No. 3 and No. 4 buckwheats, a much more desirable condition.

Effect of Moisture. As will be seen in Fig. 4, the boiler efficiency, calculated mathematically, is in direct relation to the moisture content of coal in that each 1 per cent of increase in

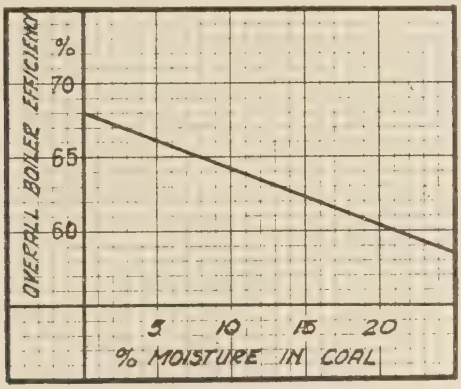

Fig. 4 Effect of Moisture on Efficiency

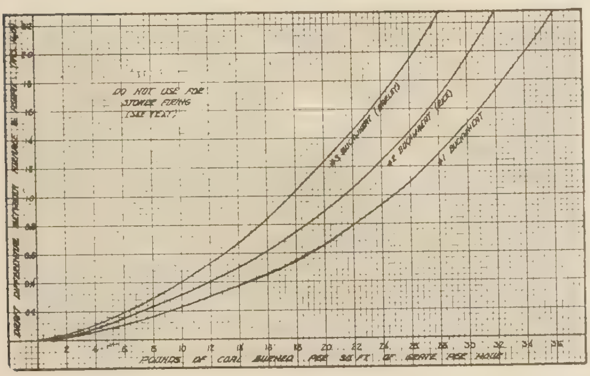

Fig. 5 Draft Required at Different Combustion Rates When Burning Commercial Sizes of Anthracite

moisture content will result in 0.4 per cent decrease in boiler efficiency. In actual practice, however, there is a reason to doubt the infallibility of this rule. A certain percentage of moisture (probably as high as 4 per cent) is credited by many engineers with being useful in tempering the fire to produce better combustion conditions. In many plants jets of steam are introduced into the ashpit to supply this moisture artificially. Properly controlled, they are believed helpful in reducing clinker formation. In addition Stenger (1), in reporting the effect of dust, remarked, "wetting dust-bearing coal practically doubles the air flow." It is thus probable that if Fig. 4 could be based upon actual tests instead of upon calculations it would be a flat curve, or even rise to show a maximum efficiency at about 4 per cent moisture. However, beyond 12 to 14 per cent moisture, much difficulty may be expected in obtaining either capacity, efficiency, or even ignition.

In a car or in a rectangular pile 8 ft deep, 1 in. of rain could produce only 1 per cent of moisture even if none drained off or evaporated. Furthermore, with anthracite, moisture is merely a surface condition which may be readily drained and dried. It is thus unlikely to become a significant factor especially if the storage pile is properly drained.

Effect of Volatile. Generally speaking, high fixed-carbon coals, such as anthracite, are the most efficient because the problem of a proper supply of secondary air is simplified. Within the anthracite industry, the entire range between the lowest- and highest-volatile coals mined is so small that this is a negligible factor, except when comparing anthracite with other fuels.

Furnace Volume. One other practical advantage of the relatively low volatile content of anthracite is the greatly increased permissible heat release per cubic foot of furnace volume. This results from the fact that, with anthracite, combustion is much more nearly complete at the surface of the fuel bed than with bituminous coal. However, it is the concensus of leading engineers that proper mixing of gases, turbulence, secondary combustion, and slag or fly-ash formation are directly and materially affected by furnace and arch design.

Air Requirements. The theoretical weight of air for the complete combustion of any fuel is practically a constant at an average of 7.65 lb of air per 10,000 Btu. With 12,900-Btu (dry) barley, 9.85 lb of air are thus required per lb of coal. With entering-air temperatures of 90 F, 13.85 cu ft equals 1 lb. Thus 137 cu ft of air will be required theoretically for the combustion of each pound at maximum CO_2 of 20.1 per cent (no excess air). Table 4 shows similar air requirements for other CO_2 values.

TABLE 4 AIR REQUIREMENTS FOR BARLEY ANTHRACITE

CO_2 in furnace, per cent	Excess air, per cent	Air required, lb[a]	Air at 90 deg F, cu ft[a]
6	225	32.0	445
8	150	24.6	342
10	100	19.7	274
12	65	16.2	226
14	43	14.1	196
16	25	12.3	172

[a] Total cubic feet per pound of coal burned.

Air and Draft Pressure. With barley as with other sizes, the combustion rate per square foot of grate varies in accordance with the differential pressure between the ashpit and the furnace. These rates, as shown in Fig. 5, will apply irrespective of the source of air supply (i.e., on either forced, induced, or natural draft, or any combination). Thus natural draft may be used for barley where it is adequate to produce the necessary combustion rate. However, where natural draft is used, it will produce such a high suction above the fire that extreme care should be exercised to keep all brickwork and openings extremely tight. For this reason many operators prefer a "balancing" forced-draft fan even though sufficient natural draft is available.

With both hand-firing and semistokers, it is necessary (on forced draft) to provide sufficient maximum air pressure in the ashpit to burn the coal properly when the fuel bed is at maximum thickness just before cleaning. A static pressure of $2^1/_2$ in. of water is suitable for fans when delivering the rated quantity of air for barley anthracite.

Air pressures, as shown in Fig. 5, will not apply with traveling-grate equipment because the continuous discharge of ash permits materially thinner fuel beds with correspondingly lower air requirements. The zoning of air on traveling grates also permits the admission of air in steps proportional to the actual rate of combustion. On traveling grates, maximum air pressures do not usually exceed $1^1/_2$ to 2 in. of water.

(As the exact air pressure required depends entirely upon such factors as fuel-bed thickness and rating, these rules refer to the maximum pressure that should be available to the operator rather than the pressures to be actually used.)

Furnace Draft. Where forced draft is available, it is customary to balance it against the pull of the stack or induced draft. However, in such cases, the furnace draft should not be below 0.05 in. of water since lower pressures will result in excessive gas and heat penetration into the brickwork with greatly increased maintenance costs. Conversely, drafts in excess of about 0.10 in. of water can be considered to carry an unnecessary amount of heat to the stack.

Furnace drafts of about 0.05 in. of water should be maintained under the nose of the rear arch of traveling- or chain-grate-stoker installations for best results.

Mechanical draft regulators, designed to hold furnace drafts between these limits, are very desirable. In their absence visual furnace draft gages are a "must" in every plant.

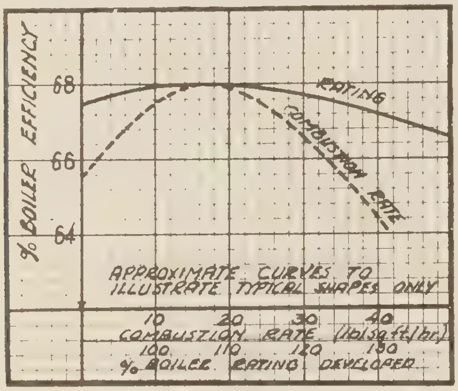

FIG. 6 BOILER EFFICIENCY VERSUS RATE AND OUTPUT

Proper Boiler Ratings. In many plants it is common practice to attempt to carry as much of the load as possible on a relatively few boilers, even though idle boilers are available. This is particularly noticeable in plants that have just recently converted from oil. From Fig. 6, it will be seen that this does not always result in the most efficient or economical operation. While boiler installations differ materially, they all have characteristics similar to those shown in the curves.

Peak test efficiencies usually occur somewhere between 100 and 150 per cent of rating on boilers without heat-recovery equipment, however, in actual daily operation, the best operating efficiency will be at a somewhat higher rating because of greater stability in the furnace and fuel bed. The addition of heat-recovery equipment may make the most economical point at some higher rating, and the efficiency will not fall off as rapidly with increased rating. The problem of whether or not to operate a sufficient number of boilers so that each will be as nearly as possible at its peak operating efficiency is a rather complex one requiring careful study for each individual case. This involves the shape of the daily load curve, maintenance and operating labor costs because of having extra equipment in service, etc. For example, in a plant with a flat load curve or a long-sustained peak, it may be advisable to operate a sufficient number of boilers to hold each boiler close to its maximum efficiency, whereas in a plant where the load is of shorter duration, the banking loss during light-load periods may offset any possible increase in efficiency. In some cases, operation of additional boilers requires additional operating labor, which nullifies the fuel savings possible.

On the other hand, attempts to operate barley fires at excessive rates of combustion frequently necessitate cleaning the fires before they are completely burned out with resultant high ashpit losses. The additional skill required for high ratings, and the strain on labor due to too frequent attention and cleaning periods are further factors.

In brief, the question of the number of boilers to be used is one of sufficient importance to warrant careful study, as based upon individual plant conditions.

Equipment Available for Burning Barley

In general three types of equipment are employed for burning barley:

1 Pinhole grates or grates with narrow slots (such as herringbone).

2 Semistokers with inclined grates.

3 Traveling-grate stokers.

A selection between these is usually made on a basis of size of installation and degree of automatic operation desired, as balanced against relative equipment costs. Approximate costs are shown in Table 5.

TABLE 5 APPROXIMATE COST OF BARLEY EQUIPMENT[a]

Pinhole grates (dumping), per sq ft	\$10 to \$12
Semistokers, per developed hp[b]	\$12.50 to \$13
Traveling-grate stokers per developed hp	\$20 to \$26

[a] All prices f.o.b. factories, not including installation.
[b] Includes forced-draft equipment.

Hand-Fired Barley Grates

Description and Design Data. Hand-fired grates for barley should be of the "pinhole" or slotted type. The latter are commonly known as herringbone grates when the adjacent rows of slots are set at an angle. A choice between the two types seems largely to be a matter of manufacturer and plant preference. The minimum practical limitation of size is about 10 sq ft; the maximum is a fuel-bed length of about 10 to 12 ft. Dumping-grate weights will usually be between 60 and 80 lb per sq ft.

Both dumping and stationary grates are available, but a substantial percentage of all grates now being sold is of the dumping type. Grates that dump to the center have the advantage of keeping hot ash away from the front of the ashpit and the rear ash closer to the point of removal.

The size of the pinholes ranges from $^3/_{16}$ to $^5/_{64}$ in., depending upon the manufacturer. They are flared or tapered underneath to make them self-cleaning and to secure a modified Venturi effect. The distance between individual pinholes is usually about $^3/_4$ in.; and the over-all metal thickness of the grates is also usually about the same.

Important points of design are the percentage of free air space and the uniformity of air delivery. If the free air space through the grates is less than 4 per cent, they are likely to restrict the flow of air unduly, and thus necessitate excessive velocities; if on the other hand the free air space exceeds 8 to 10 per cent of the grate area, the velocity will be so low for the required air delivery that much of the desired "jet" or blast effect will be lost.

Uniformity of air delivery is unusually important because of the "jet" type of operation. Either dead or unusually active areas of only a few inches will invariably produce unequal air distribution, resulting in ridges or patches of less or more active combustion with corresponding difficulty due to spotty fires.

Method of Hand-Firing. The primary rule in firing any size of anthracite is "let it alone." This applies with full force to barley. In addition, it should be fired lightly and often. It is also good practice to clean only one half of a boiler at a time, skipping every other door, going back to them only after the row has been completed. By this method, the output will be more nearly continuous, the CO_2 higher, and the gases more completely consumed.

In the case of all barley equipment, the drafts should be ad-

justed so that the surface of the fuel bed just begins to float without either being dormant or "boiling" too violently.

While the frequency of cleaning the fires is necessarily a function of design, boiler load, and quality of fuel, cleaning not oftener than once per 8-hr shift is recommended as being easier both on labor and equipment.

Spreading Method of Firing. In the spreading method, sometimes called "the alternate" method, a small amount of coal is fired at one time and spread evenly over the fuel bed from the front to the rear. The firing is done alternately through the fire doors, so that the entire fire is not blanketed with green coal. Hand-firing practice leans more and more to spread-firing.

The Bureau of Mines recommends burning coal rapidly and at high temperatures in order to secure the best available economy. Fire small quantities of coal at short intervals, so that thin places do not burn through and admit large excess of air. The quantity depends upon the grate area and draft. Firing and cleaning should be so co-ordinated as to produce a total fuel-bed depth of not over 6 to 8 in. just prior to cleaning.

Cleaning Hand Fires. Cleaning of the fire is necessitated by the fact that clinker and coarse ash will not pass through the grates. The intervals between cleanings depend upon the proportion of ash in the coal, the character of ash, and the type of the grate. If the coal contains much ash, or ash that is fusible, the fires have to be cleaned often; if light fires are carried, less clinker forms, and under such conditions the fire can often be run through a day shift without cleaning. The cleaning of the fires should be done thoroughly. All the clinker and ash should be removed, so that they cannot fuse to the side wall. They should be removed in such a way as to waste very little combustible. There are two methods of cleaning hand-fired furnaces.

In the side method one side of the fire is cleaned at a time. The good coal is scraped and pushed from one side to the other. The clinkers may have to be removed from the grates by the slice bar. When they have been loosened and broken up, they are dumped into the ashpit, or on stationary grates, and scraped out of the furnace with the hoe. The fireman should gather the clinker on the front part of the grate before pulling it out into the wheelbarrow, as this saves him from exposure to the heat. After the one side is cleaned, the burning coal from the other is moved and scraped to the clean side.

At this point, it is advisable first to spread a thin layer of coal on the grates so that the full heat of the live fire will not rest directly on the grates. A few shovelfuls of fresh coal are then added, in order to have enough burning coal to cover the entire grate when the cleaning is done. This adding of coal is important, especially when the cleaning must be done with the load on the boiler. The clinkers are then removed from the second half of the grate. In cases where the load is heavy, a time interval is recommended between cleaning the two halves of the grate, in order to allow the new fire to build up to its load-carrying capacity.

When cleaning is started, there should be so much burning coal in the furnace that enough will be left to start a hot fire quickly, when the cleaning is completed. If a light fire is carried, it may be necessary when starting to clean to put some fresh coal on the side to be cleaned last. During cleaning the damper should be partly closed. A fireman, after becoming familiar with the side method, should be able to clean a 200-hp boiler furnace in 10 to 12 min.

In the front-to-rear method of cleaning, the burning coal is pushed with the hoe against the bridge wall. It is usually preferable to clean one half of the grate at a time. The clinker is loosened and pulled out of the furnace and the burning coal is spread evenly over the bare grates. If the front-to-rear method must be used while the load is on the boiler, the side method should be employed after the day's run is over, so as to prevent

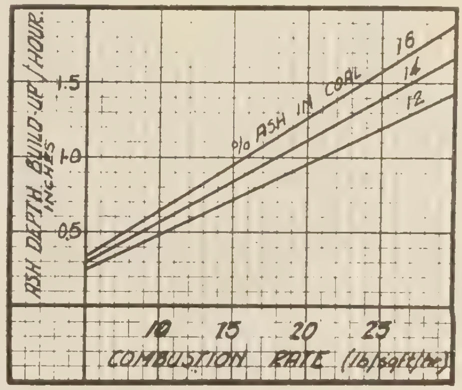

Fig. 7 Ash Accumulated Between Cleanings

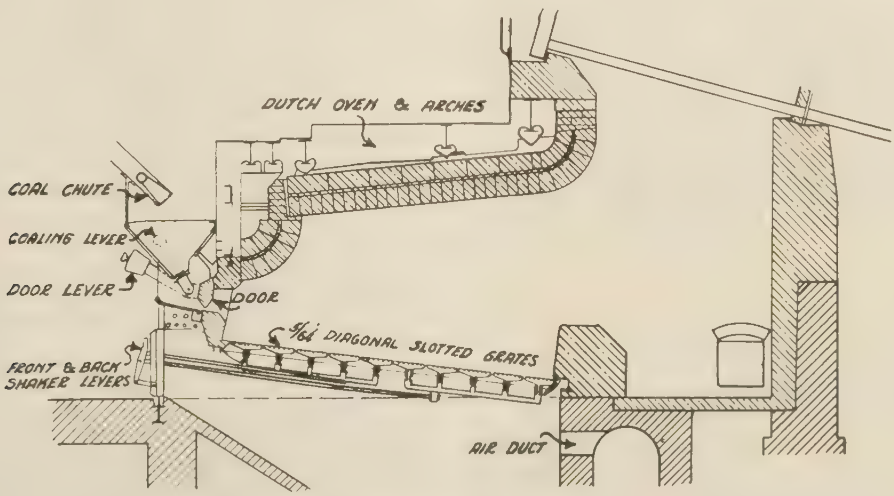

Fig. 8 Spreader or Semistoker
(Courtesy of the McClave Company, Allentown, Pa)

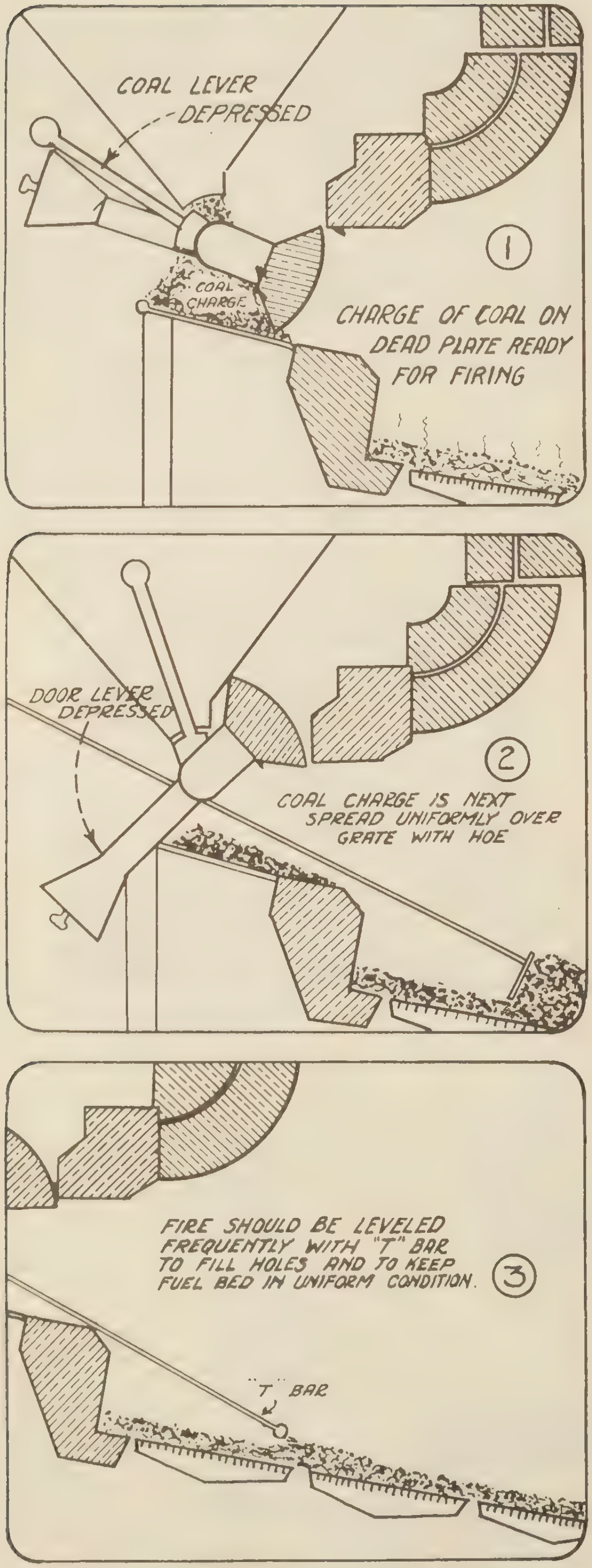

Fig. 9 Proper Method of Operating Hand Stoker

the large accumulation of thick and hard clinker at the bridge wall.

Some firemen have the habit of pulling the clinkers out of the furnace without scraping and pushing the burning coal against the bridge wall or to one side. This really is not a method of cleaning the fire. They run a slice bar under the clinker to lift it to the surface of the fuel. Then they take a hoe and pull the large pieces out. The small pieces are not easily detected and are left in the fire. These fuse in a few minutes, due to the high temperature near the surface of the fuel bed, and to the air deficiency under the clinker. Thus more masses of clinker are formed which are usually worse than those previously removed. This habit should be discouraged.

The cleaning of a banked fire should be done about 2 hr before steam is needed. It is advisable to bank the fire at the front of the grate near one of the doors. This exposes the clinkers which can be pulled out.

As cleaning intervals are largely determined by the ash accumulation, Fig. 7 shows the depth of pure ash produced in each hour under various conditions.

Operating Results. With good firing and cleaning, combustible in the ash will be as low as 12 to 18 per cent; and CO_2 of from 10 to 12 per cent can be carried continuously without danger of CO.

Light-Load Firing. As has been stated 6 to 8 in. is a proper "full-load" fuel-bed depth for hand-fired barley. However, in periods of light loads, the attention required may be considerably decreased by firing considerably heavier.

Hand or Semistokers

A type of semiautomatic equipment especially suitable for barley coal is the semistoker, such as is shown in Fig. 8. These stokers consist of a suitably inclined dumping grate usually of the slotted type having openings approximately $^5/_{64}$ in. wide, and a total air space of approximately 8 per cent.

By means of an ingenious arrangement of gates, coal may be dropped onto a convenient hearth at the upper edge of the inclined grate. It is then spread over the live coals manually by means of a long-handled spreader bar.

While not a stoker in a sense of being motor-driven, this equipment does have the advantage of hopper storage of coal with manual attention reduced to the actual operation of spreading the coal, cleaning the fire, and dumping the ash or clinker. The amount of manual attention involved is thus comparable or even better than the removal of clinker from the familiar bituminous underfeed stoker. Installations of this nature have been made in outstanding schools, office buildings, and industrial plants with perfect success, and extremely satisfactory operation.

Sizes Available. Semi-stokers are available in a full range of sizes up to about 12 ft long and 14 ft wide, with capacities from 100 to 600 developed hp. Approximately 7 to 10 ft of setting height is required for their installation.

Operating Characteristics. The operating characteristics of the semistoker are similar to those on dumping grates. As has been stated, labor is less, and, because of the type of firing, operators feel that more uniform better conditioned fuel beds can be carried. Somewhat higher loads can thus be carried, as are reflected in the ratings given in Table 2. Typical operating methods for a semistoker are shown in Fig. 9.

Chain- and Traveling-Grate Stokers

The ultimate in the mechanical-firing of barley and the No. 4 buckwheat sizes of anthracite is through the use of moving grates of the endless or "belt" type. There are two familiar versions of these, i.e., chain-grate stokers, and traveling-grate stokers. The latter type is shown in Fig. 10.

Chain-Grate Stokers. In principle, the chain of a chain grate

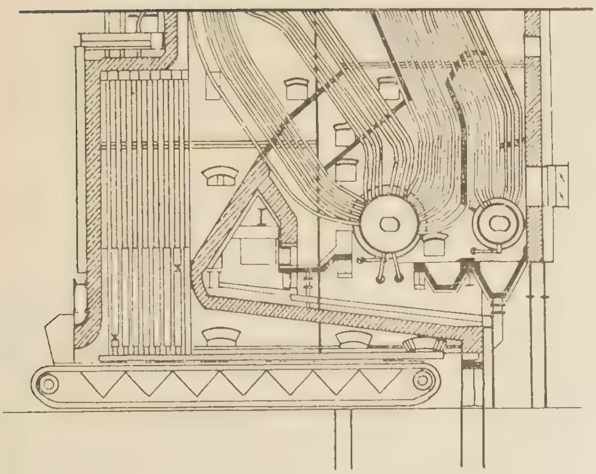

FIG. 10 MODERN TRAVELING GRATE WITH ARCHES FOR ANTHRACITE

consists of a series of individual links made up into the form of an endless chain. The edges of the links are usually serrated to provide from 7½ to 10 per cent free air space for the passage of combustion air.

Traveling-Grate Stokers. The traveling-grate stoker is similar to the chain-grate stoker, the difference being that in traveling grates the grate bars consist of small individual castings or keys carried forward in endless-belt fashion on chains but not being in themselves a part of the chain. The free-air openings are usually about the same as on the true chain-grate type.

A characteristic advantage of the traveling grate is the ability to replace and add burned keys without disassembling the chain.

As all other points of construction and operation are practically identical, the remainder of this section applies equally to chain and traveling grates, even though for the sake of simplicity the term "traveling grate" is used.

An outstanding advantage of these types of stokers lies in the fact that the air can be carefully regulated and zoned as combustion proceeds. As the coal burns progressively from its ignition at the front of the grate to ash at the rear, from three to seven (or even more) partitioned wind boxes are employed to divide the air supply into zones. In extremely large installations, the rear wind box is also subdivided longitudinally to equalize the burning-out of the tail end of the fire.

Size and Design Data. Traveling-grate stokers are manufactured in a wide range of sizes. The largest stoker of this type so far built is located at the Calco Chemical Company, Bound Brook, N. J. This stoker is 24 ft wide × 28 ft long with 672 sq ft of grate. Another outstanding traveling grate is at the Cedar Street Plant of the Pennsylvania Power and Light Company in Harrisburg, which is 23 ft wide × 25½ ft long with 586 sq ft of grate.

It has been authoritatively stated that, with No. 3 buckwheat (barley) of 12,900 Btu, it is possible to design and build a stoker to deliver 300,000 lb of steam per hr continuously. This is given as the present practical limit of size for stokers on anthracite.

At the other extreme, traveling grates are manufactured as small as 3 ft wide × 7 or 8 ft in length, but it is not usually economically feasible to install them for less than 100 developed hp.

The development of the traveling grate has seen many changes with material improvement in both flexibility and combustion. Grate speeds, originally 15 fph, now range from 40 to as high as 100 fph. Fuel-bed thicknesses have decreased from 6 to 8 in. to a present maximum of 4 to 4½ in. (with barley). Vastly improved arch design has stepped up combustion rates from 25 lb to a present practice of 35 to 40 lb, and on extremely large jobs where provision for the re-use of siftings and fly ash has been made, 45 to 50 lb per sq ft have been reported.

Performance Characteristics. With old-style front-arch furnaces, combustible in the ash of 25 to 30 per cent is considered good practice on traveling grates. With the new-type rear arches, the combustible in the ash will be as low as 10 per cent. CO_2 of 12 to 14 per cent can be consistently carried without CO.

CONCLUSIONS

In conclusion, with adherence to a very few fundamental rules, satisfactory economical performance may be expected from barley on several classes of equipment:

1 The coal should be as free from fine undersize or dust as practical.

2 Low ash content is preferable unless the plant has been specifically designed for high-ash fuel.

3 The fundamental rule of firing anthracite is "let it alone."

4 A selection of hand-firing, semi-hand-firing, or fully automatic stokers is available on a basis of initial equipment cost, cost of operating labor, capacity which can be developed in a given boiler, and maintenance cost.

5 Regardless of the firing method, anthracite is free from all smoke and soot and not subject to spontaneous combustion.

ACKNOWLEDGMENT

For assistance in preparing the material for this paper the author is grateful to C. H. Frick, Plant Betterment Engineer, Pennsylvania Power & Light Company, Allentown, Pa.; Roy Johnson, Carbondale Grate Bar Company, Scranton, Pa.; J. H. Kerrick, Fuel Engineer, Philadelphia & Reading Coal & Iron Company, Philadelphia, Pa.; Christian Schillinger, President, Coxe Stoker Engineering Company, Hazleton, Pa.; Charles Warg, Chief Engineer, The McClave Company, Allentown, Pa.; Harvey Rettew, Chief Mechanical Engineer, Philadelphia School Board; John Van Brunt, Vice-President, Combustion Engineering Company; Otto de Lorenzi, Combustion Engineering Company; A. C. Fieldner, Chief, Fuels Service, U. S. Bureau of Mines; Chas. Schantz, Weston Dodson & Company.

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Discussion

A. A. BATO.[3] In the present fuel shortage, the author has rendered a most valuable service in directing attention to barley anthracite, a fuel available in large quantities, yet neglected even in cases in which it could be used with little or no difficulty. Every bit of information in connection with the use of this fuel is welcome, because, in the present emergency, plant managements cannot experiment with new fuels, unless they get full and

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reliable data enabling them to use such fuels readily in an efficient manner. On the other hand, due warning should be given concerning cases in which the fuel cannot be used without creating disturbances in plant operation.

The writer can state from actual experience that the use of steam jets in the ashpit to avoid clinkering, concerning which the author says that it is believed to be helpful, is actually of great help, easy to install, and comparatively economical, as the amount of steam used does not exceed 2 to 3 per cent of the steam generated, even in extreme cases.

The importance of proper grates has been somewhat overemphasized. There are two points involved; (*a*) excessive sifting of coal into the ashpit, and (*b*) the danger of holes being blown into the fuel bed in the case of too large air openings. As to item (*a*), it must be remembered that on hand-fired grates the coal does not rest on the grate but on the ashes of the coal burned previously, except immediately after fire-cleaning. Accordingly, if grates with small openings are not readily available, the fireman may either cover the grates after fire-cleaning with a thin layer of ashes or preferably small-sized clinkers, or use a small quantity of larger-sized coal for this purpose. As for item (*b*), the danger of holes being blown into the fuel bed is somewhat more serious, but prevails only during the first half hour or so after fire-cleaning, and can be minimized with a little care and frequent checking of the fire. This settles also the question of the grates of various shapes claimed to direct the jets of air at various angles, crosscurrent, etc., into the fuel bed, usually illustrated with "trained arrows." As a matter of fact, the direction, shape, etc., of the air jets reaching the live coal are determined by the ashes immediately under the latter.

The statement concerning furnace draft, "drafts in excess of about 0.10 in. of water can be considered to carry an unnecessary amount of heat to the stack," can be accepted only for cases in which the boiler setting is not tight, or in which other features of the design permit the infiltration of outside air into the furnace or the passes. If the setting is practically airtight, the stack loss depends upon the amount of combustion air only, independent of whether the draft was produced by low forced-draft pressure and high furnace draft, or vice versa. In the case of a heating plant serving a very tall building, the height of the chimney is likely to be determined not by the draft requirements but by the height of the building, resulting in a very high furnace draft. In such a case, it might be more economical to provide the boiler setting with a steel casing to prevent air infiltration, or to install a barometeric damper to reduce the stack draft and thus to save on the power used to produce forced draft.

The low volatile content of barley anthracite, especially when combined with a high moisture content, results in a characteristic of slow and difficult ignition, as compared with the flashy, high-volatile soft coal. This may cause some difficulty, as in the case of traveling-grate stokers, where the ignition of the coal may proceed more slowly than the velocity of the grate, in which event the fire "runs away" from the hopper gate. When this occurs, the grate must be stopped for a short time or slowed down, resulting in reduced performance. Such a condition may be due to faulty arch design, etc., and is likely to prevail when the stoker designer had to adapt the stoker to a narrow and long boiler. It is not likely to happen to a wide and short grate traveling at a slower speed for the same performance.

Slow ignition naturally causes difficulties in picking up sudden peak loads; consequently the real field for the use of barley coal is in plants having uniform loads.

As to mixing barley with bituminous coals, the following procedure should be adopted to avoid difficulties: Suppose the minimum volatile content at which the equipment can operate was found to be, let us say, 22 per cent, this being the volatile content of the coal used in the past. Suppose also, that barley coal of 5 per cent volatile content and bituminous coal of 35 per cent volatile are available. In such a case, the proper mixture will be about 42 per cent anthracite to 58 per cent bituminous coal, giving a volatile content of 22 per cent. In addition to the volatile content, the texture of the coal may have an influence upon ignition characteristics.

The information concerning the importance of the undersized coal is especially interesting. The paper is quite comprehensive, covering the field in a full and practical manner.

J. F. Barkley.[4] For some months, the Bureau of Mines, in co-operation with the War Department, and Athracite Industries, Inc., has been experimenting with the use of mixtures of barley anthracite and slack-size bituminous coal on underfeed stokers. Successful results have been obtained at several plants now regularly using a mixture; trials at further plants are proceeding.

With the eastern caking coals on single-retort stokers, the addition of some 15 to 20 per cent of barley is usually sufficient to open up the fuel bed and give less trouble from caking. At one plant, where the bituminous coal was high in ash, the air-pressure drop through the fuel bed lessened with the increase of barley, making it possible to carry higher ratings than with straight bituminous slack. At this plant, as much as 80 per cent barley gave satisfactory operation and carried the required load. At another plant, 50 per cent of barley was decided upon as the best ratio. At this plant, the number of times the fireman resorted to hand-working the fuel bed was reduced as the percentage of anthracite increased.

The effect of the anthracite varies with the amount of ash in the bituminous coal with which it is mixed. For low-ash bituminous coals, lower percentages of anthracite give better results, the higher percentages giving less flexibility on load changes and lower ratings. For high-ash bituminous coals, higher percentages of anthracite can be used satisfactorily in obtaining similar loads.

The combustible in the ash and refuse from single-retort stokers varied from about 18 per cent for straight bituminous up to 35 for an 80 per cent anthracite mixture. Combustible in the flue ash also increased with the percentage of anthracite.

As a broad general statement, the efficiencies obtained on single-retort stokers with various percentages of anthracite were about the same as with straight bituminous, the gain in better fuel-bed conditions due to the anthracite being offset to some extent by the increase in combustible in the ash and refuse.

The amount of smoke produced lessened as the percentage of anthracite increased.

Author's Closure

Both Mr. Bato's and Mr. Barkley's discussions are appreciated as being valuable additions to this paper. With reference to Mr. Bato's comment with regard to the relatively slow ignition of anthracite, we believe that this can be largely offset by modern arch design to a point which will permit a sufficiently high degree of flexibility on chain-grate stokers for all practical purposes.

The writer is familiar with several installations in dyehouses and other plants having wide fluctuations in load, and in no instance has the lack of flexibility been cited as a disadvantage where arch and furnace design was at all modern.

Mr. Barkley's comments relative to the use of mixtures of barley and bituminous are particularly important since in view of current shortages of soft coal, this subject is receiving very serious attention from many plant owners at this time.

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[4] Chief, Division of Solid Fuels Utilization for War, Bureau of Mines, Washington, D. C. Mem. A.S.M.E. This discussion is published by permission of the Director, Bureau of Mines, U. S. Department of the Interior.

New Combustion-Control Methods for All Standard Fuels

By ROBERT REED,[1] TULSA, OKLA.

As a result of extensive study and field experience with combustion problems, the author offers suggestions for increasing the accuracy of combustion control in all types of furnaces and with all types of standard fuels. This is accomplished by overcoming the existing limitations of flue-gas-analyzing methods either by recorded CO_2 concentrations, or by Orsat analysis. Curves are given which provide a simple means for determining "excess air by CO_2" and "excess air by O_2," by the use of which excess-air control can be quickly adjusted for any change in fuel to obtain unimpaired combustion. The importance of determining the presence of aldehydes in flue gases is recognized and a positive method for their indication has been developed in the form of a simple apparatus which is described. This check on combustion may save considerable fuel, and a great deal of mechanical difficulty with furnaces.

It is the purpose of this paper to offer some new and perhaps revolutionary suggestions for obtaining genuinely accurate control of combustion in all types of furnaces and with all types of standard fuels. It is needless to say that very precise standards for such control have already been set up and within their limitations these standards have real value for the engineer.

The fact that existing control standards have limitations which are universally recognized justified further thought toward arriving at means of eliminating the limitations as far as possible. We have done considerable research on this problem and offer the results thereof.

It is to be recognized that in a furnace fired with fuel which has fixed characteristics as to analysis and heating value such suggestions as we make will have less bearing than in the case of a furnace fired with two or more different types of fuels. We find, however, that furnaces fired with but one fuel are comparatively rare except in the Gulf Coast area where an abundance of cheap natural gas relegates auxiliary fuels to emergency status. Many such furnaces operate for years without change of fuel.

In the average installation where two, three, or even more types of fuel must be used to maintain operating conditions, or where the analysis of a given type of fuel is subject to change without notice (such as is the case in all refineries), the engineer is confronted with a serious problem in maintaining control of combustion through all the fuel changes.

Limitations of Flue-Gas-Analyzing Methods

Again, as we all know, control is taken through either recorded concentration of CO_2 in the flue gases or by Orsat analysis. If the first is the case, the engineer must estimate the amount of excess air as represented by a given CO_2 indication based upon the ultimate CO_2 available from the fuel being fired. If two or more fuels are being fired in combination in the same furnace the recorded CO_2 will have almost no value at all unless the precise volume of each fuel is known, and it would be necessary to resort to Orsat analysis to find the true condition of excess air.

Orsat analysis, where time and trained technicians are available, is far more satisfactory; but even here we find much room for improvement. The principal fault with such procedure lies in the fact that the Orsat reveals conditions existing at the time the sample was taken and these are subject to change many times during the course of a day. Other sources of error lie in faulty technique with the Orsat; dilution of the sample by air and exhausted or contaminated reagents, or leveling water in the burette.

We hasten to say that through long association we have the greatest respect for the Orsat flue-gas-analyzing apparatus but we must recognize its limitations. It will, if properly operated, indicate the concentrations of CO_2, O_2, and CO in the sample being tested, but beyond that it is no longer valuable.

The engineer may wonder just what further information on the analysis of flue gas might be needed. In order to answer this question, the author will review a little of the theory of combustion and cite some experiences in the field pertaining to this matter.

Combustion Theory

It seems to us that combustion theory is just that, as we have encountered conditions which seem to refute most of it; but after having made such a qualifying statement we will say combustion of hydrocarbons proceeds as a race between hydroxylation and thermal decomposition according to the type of fuel being burned and the manner in which it is burned. A clear flame indicates hydroxylation and a luminous flame indicates thermal decomposition, but neither condition denies or makes impossible the presence of the other. In both cases the end products of complete combustion are CO_2 and H_2O.

In the case of thermal decomposition which produces a luminous flame, the luminosity comes from the presence of glowing free carbon which would, in complete combustion, burn to CO_2 in the normal manner. If this free carbon should, however, be chilled to a temperature below its kindling point before combustion is complete, it will appear as soot. It will also appear as soot if sufficient O_2 is not immediately available.

In the case of thermal decomposition in the course of hydroxylation caused by either a lack of turbulence to force mixture with O_2, lack of O_2, or chilling to a temperature slightly below the kindling point of the intermediate compound, aldehydes will appear in the flue gas. At times free H_2 in quite appreciable quantity has been found in flue-gas samples, but such conditions are rare.

We would point out, therefore, that if the true condition of combustion is to be found, means of indicating the presence of soot, aldehydes, and free H_2 might be necessary in addition to the classical CO_2, O_2, and CO. Soot, if present in appreciable quantity, may be seen emerging from the stack. The presence of free H_2 is rare, so for all practical purposes we need only consider the aldehydes as important.

Some fuels, such as blast-furnace gas, burn with a clear flame

[1] Engineer, John Zink Company.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

in which no hydroxylation is involved because the nature of the burning of such combustible elements precludes possibility of luminosity in the flame. With such a fuel the Orsat is entirely sufficient. Coke also falls within this classification as long as its volatile content is very low.

Combustion Experiences in the Field

One of the author's most unusual experiences in connection with combustion problems related to the operation of an HRT boiler in a relatively small, but exceedingly well-managed plant in the Southwest where only one very closely controlled fuel was used.

In this particular plant the management had spent some $5000 improving conditions in the boiler room. New burners had been purchased; the furnaces had been rebuilt; steam leaks had been hunted down relentlessly, and in general no means of obtaining efficient operation of the boilers and the boiler plant had been overlooked. Hourly Orsat samples were taken from the breeching of the boilers and at no time was the excess-air factor as indicated by the Orsat analysis allowed to exceed 10 per cent.

Any engineer would say that with such operation the management could reasonably expect to make steam at a very low cost. Such was not the case, however, for immediately upon starting up the reconditioned plant the fuel costs rose 10 per cent for the same steam production. For 6 months the plant operated in this manner, despite frantic efforts on the part of the plant personnel and various other engineers.

The hourly Orsat checks showed, with minor variations, as follows: CO_2, 10.7 per cent; O_2, 2.0 per cent; CO, 0.0 per cent.

This analysis indicates an excess-air factor of 10 per cent when using a 0.55-specific-gravity natural gas which was the condition set as a standard. Since no CO was indicated, it was felt that satisfactory results were being obtained and the 10 per cent increase in fuel cost was due to some other factor in the plant.

It was at this point that the author entered the picture, merely as a person who had heard of an interesting condition and wanted to examine it firsthand. We found the burner making a beautiful clear fire with no haze. We found the furnace draft was being held at 0.02 in. w.c. and we checked the flue-gas condition with our own Orsat to get identically the same analysis as had been obtained by the plant personnel for some hours previous to the time of our visit. We noted the flue-gas analysis checked with the fuel H/C ratio by weight. We were prepared to agree that the trouble with the plant lay in some other condition, until we went to the back end of the furnace to look through a peephole at the burner in order the better to admire the beautiful fire it was making.

Remember that furnace draft of 0.02 in. at the burner level. The peephole was considerably above the level of the burner so a pressure existed at the peephole level to cause the furnace gases to pour out into the boiler room. The furnace gases were heavy with the odor of formaldehyde and at once we had a clue to the 10 per cent increase in fuel cost.

We suggested increasing the furnace draft and despite some opposition from the plant management, the draft was raised to 0.08 in. w.c. to see what would happen. The load on the boilers was very constant and the fuel flow remained exactly as it had been before the draft was increased; yet the boilers popped within a minute of the time the draft was increased.

The next hourly Orsat sample showed as follows: CO_2, 11.4 per cent; O_2, 2.0 per cent; CO, 0.0 per cent.

This, while it reflected a decidedly improved condition in the furnace, did not check with the H/C ratio of the fuel. A union in the flue-gas sampling line was found to be leaking and, after being tightened, the flue gas checked as follows: CO_2, 11.7 per cent; O_2, 0.0 per cent; CO, 0.0 per cent.

This analysis checked with the H/C ratio of the fuel and was presumed to be correct. The furnace draft was raised again until a true indication of 10 per cent excess air was obtained, and the fuel costs instantly dropped to a level which would justify the installation of the new equipment.

In this case tremendous fuel loss was suffered for more than 6 months despite very careful control according to accepted methods. The errors in this case were the inability of the Orsat to show the presence of aldehydes in the flue gas, and the fact that the leak in the flue-gas sampling line allowed sufficient O_2 to appear in the sample to check the sample against the H/C ratio-by-weight of the fuel.

Another incident, which is very much in point concerns the operation of a large boiler in an Eastern refinery. Fuels being burned were refinery gas and oil. Close control was being maintained according to accepted standards, yet this boiler operated for weeks in such a manner that numerous heat balances refused to check by about 10 per cent. No smoke appeared at the stack and the Orsat showed no CO in the flue gas.

A corps of engineers set about trying to discover the difficulty and failed completely to do so until a most fortunate mistake occurred. A fireman was instructed to raise the draft on an adjoining boiler but misunderstood his orders. He raised the draft on the boiler being checked, and the increased draft produced some startling results.

One of the results was an almost instant increase of about 10 per cent in the production of steam with the same fuel flow, and the other result was that the heat balance checked very closely.

In order to determine just what conditions existed in the furnace with the original furnace draft, the damper was returned to its original setting. Steam production fell off 10 per cent as had been the case before. Samples of flue gas were sent to a laboratory for analysis which revealed the presence of free H_2 along with other unburned hydrocarbons.

In this case, because two fuels were being burned in combination, it is evident that checking the H/C ratio of the fuel against the flue-gas analysis had been neglected and leakage of air into the samples was not detected. The leakage of air into the samples caused O_2 to show in the analysis while in the furnace there was actually a deficiency of O_2 for complete combustion.

Checking CO by Orsat Unnecessary

The author's experience in the field has been almost entirely with oil- and gas-fired furnaces of all types, forms, and sizes. In the course of a number of years of work with such furnaces, we are compelled in all honesty to admit that we have never found CO in detectable quantities with Orsat analysis of flue gas. We do not question its presence in very minute quantity but, without resorting to very painstaking analysis by the iodine-pentoxide method (which is far too exacting and expensive for ordinary use) we have found checking for CO a formality which has an almost inevitable answer of "zero per cent."

In order to avoid confusion, it must be said that, in the burning of coal or coke where carbon is burned directly and alone, following the evolution of such volatile elements as may be present in the fuel, CO in very appreciable quantities may by found. Our remarks do not, then, apply for the burning of such fuels.

We have discussed the issue of CO with many engineers in the field and the consensus is that the CO reagent in the Orsat unless coke or coal is being burned, is more or less useless. Many say frankly that they never use it in ordinary checking. We

recognize this statement as being highly revolutionary but we submit it is concurred in by a majority of operating engineers in the field. Even where coal is being burned, combustion deficiency is not fully indicated by the CO reagent since the volatile elements burn partly by hydroxylation and unburned substances other than CO can, therefore, readily appear in the flue gases.

Controversial as our statements have been, we hope it will be agreed that steps to correct for Orsat deficiencies should be taken in the operation of a majority of the furnaces in America, in order that high efficiencies may be maintained and precious fuel saved.

It has been indicated that by far the greater portion of the deficiency may be taken care of by obtaining suitable means for indicating the presence of aldehydes in the flue gases. It has also been pointed out that there is a very definite relationship between flue-gas analysis and the H/C ratio by weight of the fuel being burned. If the Orsat analysis does not check with this ratio, something is wrong in the operation of the furnace or the method of sampling.

Simple Means for Determining Excess-Air Factor of Fuels

In the course of poring over data on flue-gas analysis, we have also come across a most singular fact which, up to this time, seems to have escaped the notice of the profession at large. This concerns the fact that a certain concentration of O_2 in the flue gases indicates almost exactly the same excess-air factor for any fuel from natural gas to coke within the limits of accuracy of analysis and observation. This also is applicable for firing any of the fuels in the range previously noted in combinations of two or more.

It is our observation that the O_2-excess-air relationship is independent of the CO_2 concentration of the flue gases which would change with variations in the H/C ratios by weight of the fuels being burned. The CO_2 concentration of the flue gas is therefore relegated to the provision of a means of checking the flue-gas analysis against the H/C ratio of the fuel or fuels.

We have plotted two sets of curves on "Excess Air by CO_2" and "Excess Air by O_2" for consideration and use in the field. These curves, Figs. 1 and 2, are based upon data provided by various excess-air charts now in common use. These charts are very accurate as is proved by the fact that the data, shown on the curves as plotted, check calculation of excess air based on the N_2 content of the flue gases. This is true for the entire H/C ranges covered by the curves.

In the use of the curves it is to be presumed that there is no dilution or contamination of the flue-gas sample by air which would confuse results and make the sample useless.

As an example, consider the case of a power-plant boiler being fired with natural gas when, because of an emergency, it is necessary quickly to shift the firing to 75 per cent oil and 25 per cent gas. The use of oil would make it necessary to raise the excess-air factor from, say, 10 to 15 per cent. If excess-air control is being taken on the basis of the CO_2 concentration of the flue gases, only a mathematical wizard could tell the operator in anything like a reasonable time what CO_2 indication he should have and hold tq maintain efficiency.

It is here that the "Excess Air by O_2" curves become genuinely valuable as a time and efficiency saver. The chief engineer could instruct his operator to hold the O_2 concentration of the flue gases at 2.9 per cent and let the CO_2 go where it will. If it is necessary later to go 100 per cent on oil, the operator may continue to disregard the CO_2 indicator and hold the O_2 at the same 2.9 per cent to remain with very close limits of the 15 per cent excess-air factor. The same conditions would prevail if the fuel

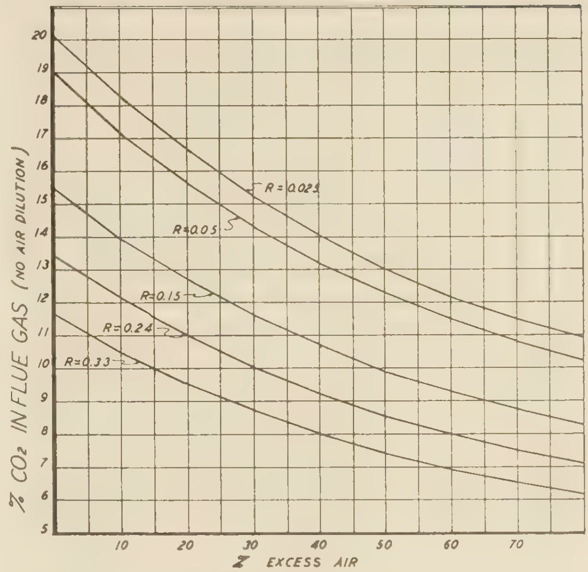

Fig. 1 Excess Air by CO_2

R = H/C Ratio by Weight

Coke (mean)	R = 0.025
Anthracite coal (mean)	R = 0.025
Bituminous coal (mean)	R = 0.05
Oil (mean)	R = 0.15
Refinery gas (mean)	R = 0.24
Natural gas (mean)	R = 0.33

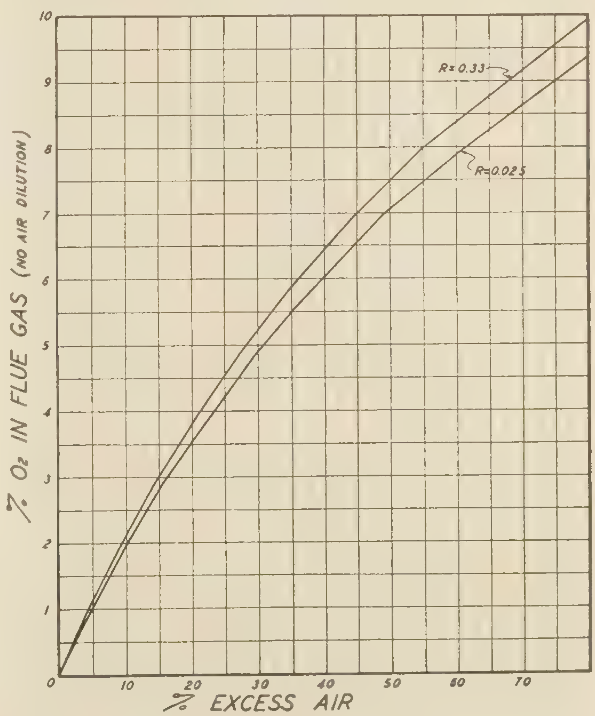

Fig. 2 Excess Air by O_2

R = H/C Ratio by Weight

Coke	R = 0.025
Anthracite coal (mean)	R = 0.025
Bituminous coal (mean)	R = 0.05
Oil (mean)	R = 0.15
Refinery gas (mean)	R = 0.24
Natural gas (0.55 sp. gr.)	R = 0.33

change was from oil to coal or any other combination or combinations necessary to maintain operation.

The O_2 observation may be taken either by Orsat or by any one of a number of continuous recorders which are available. That these recorders are reliable is proved by the fact that they are being used in the control of very delicate catalytic operations in the refining industry with excellent results.

If CO is present in the flue gas, the amount of air necessary to burn it must be subtracted from the indicated excess air to arrive at a true condition. Fuels bearing high concentrations of N_2 will require additional calculation to arrive at true excess-air conditions.

The curves may also be used to check the flue-gas analysis against the H/C ratio by weight of the fuel. As an example, consider a furnace being fired with oil having an H/C ratio of 0.15. If the excess-air factor has been set at 30 per cent, the Orsat analysis should be as follows: CO_2, 11.75 per cent; O_2, 5.07 per cent; CO, 0.0 per cent. If the analysis is at variance something is wrong either by dilution of the sample, lack of sufficient air for combustion, poor mixture of fuel and air, or some other more obscure error. Checking should continue until the flue-gas analysis checks the conditions which should exist with the fuel being burned.

Determining the Presence of Aldehydes in Flue Gases

In considering the presence of aldehydes in the flue gases, we have pointed out that their presence should be indicated by failure of the Orsat analysis to check with the H/C ratio of the fuel being burned. At times, however, because of air dilution of the sample or for some other reasons, this check may become unreliable. Further and more positive means of indicating the presence of the aldehydes becomes necessary.

In collaboration with J. A. Hart and L. N. Hollis, chemists of a major oil company, we have arrived at what we think is a simple and most effective means of indicating the presence of aldehydes in flue gases through the use of a standard reagent in connection with some very simple apparatus. Fig. 3 is a more or less self-explanatory sketch of the apparatus. The reagent is prepared as follows:

Dissolve 0.2 g of certified basic fuchsin in 10 cc of a freshly prepared cold saturated aqueous solution of SO_2. Allow this solution to stand for several hours until all pink color disappears and the solution becomes either colorless or pale yellow. Dilute with distilled water to 200 cc and preserve in a tightly stoppered dark bottle. The reagent must be protected from both light and air. Particular attention is to be directed to the fact that acid fuchsin may not be used. The solution thus prepared is a standard aldehyde indicator.

About 15 cc of the reagent is poured into the flask of the test apparatus which is then connected to a tube leading to the source of flue gas. Before the connection is made, the tube must first have been thoroughly purged of air.

Flue gas is then caused to bubble through the reagent for not less than 2 min by means of the aspirator. The apparatus may then be disconnected from the sampling tube and allowed to stand for 2 min more. Any pink, red, purple or blue discoloration of the reagent indicates the presence of aldehydes, and the operator will know that all the fuel being introduced into the furnace is not being consumed. He may then take steps to find the trouble and correct it.

We think that since the test is so simple and easy to run, it would be well to check the flue gases thus at hourly intervals or (in boiler plants) at the time boiler-water samples are taken.

It should, perhaps, be pointed out that incomplete combustion may exist despite the presence of ample O_2 to burn all the fuel because of failure of the burners to provide a thorough mixture

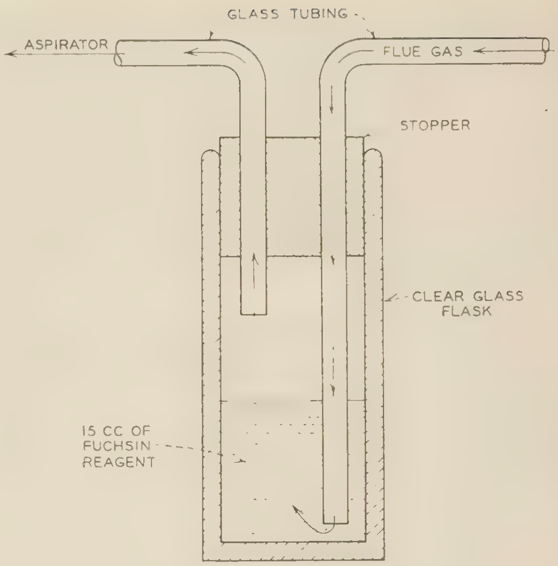

Fig. 3 Apparatus for Fuchsin Aldehyde Indicator as Suggested for Checking Flue Gas

of air and fuel. This failure may be due to plugging, clogging, or obstruction of portions of the burner. A radical change in fuel characteristics, which may not be noted by the plant personnel, may also result in incomplete combustion. The aldehyde test is really a double check which may save a great deal of fuel and mechanical difficulty with the furnace.

Perhaps the greatest source of error in flue-gas analysis lies in dilution of the samples by air. Chinks, cracks, and holes in the furnace, breeching, and stack are chief offenders, while leaks in sampling tubes come in for their share of condemnation. No pains should be spared to see that such sources of dilution are closed.

The best place from which to take a sample is from the furnace proper at a point where combustion has been completed some distance away and also at a point where minimum draft or subatmospheric pressure exists.

Such a point will be found near the arch of the furnace or, in the case of a boiler or generator, in or near the first pass. If the sample is to be taken from a zone of high temperature, it is necessary to use a water-cooled sampling tube to avoid loss of O_2 from the sample in oxidation of the tube metal. Means of obtaining an average sample should be considered important.

Discussion

A. A. Bato.[2] So far we have heard mostly of the presence of hydrocarbons only in the unburned flue gases, but this paper reports other cases involving aldehydes, that is, carbohydrates.

The importance of checking the flue-gas analysis against the H/C ratio, or, to be more exact, against the ratio of $H_2 - \frac{H_2}{8}$ to C, cannot be too strongly emphasized. In a previous paper,[3] the writer showed mathematically that a flue gas may contain CO and unburned H_2 in some form in such proportions that a check of the CO_2 and O_2 content against the H/C ratio may indicate the absence of CO, and therefore be misleading. A. H.

[2] Consulting Engineer, East Orange, N. J. Mem. A.S.M.E.

[3] "Flue Gas Computations," by A. A. Bato, *Mechanical Engineering*, vol. 48, 1926, pp. 330–336.

Senner[4] of the Bureau of Agricultural Engineering published the results of some tests made on domestic oil burners with a Bureau of Mines analyzer which involved such cases. Out of eight tests, about six check more or less closely in the manner mentioned, with the gases containing considerable quantities of hydrocarbons, also CO up to 5.1 per cent, with 1.2 per cent as the average.

They also indicate that the gases should be tested for CO with the Orsat, especially when burning light oils, notwithstanding the opinions of the many engineers quoted by the author as saying that the CO reagent in the Orsat is more or less useless. Their opinions may have had their origins in the assumption that, where there is no smoke in an oil- or gas-fired furnace, there is no CO either, consequently, it is enough to set the burner for smokeless combustion and test the gases for CO_2 and eventually for O_2 only. Senner's tests show heat balances with "unaccounted-for" losses up to 15 per cent, which, however, were explained with the aid of the complete gas analysis.

Concerning the effect of leaks in the gas-sampling pipe line, it must be emphasized that it can be misleading in one respect only, i.e., it may give the result that there is more excess air present in the gases than in reality. The analysis will be the same if some air got into the gases through the burner, crack in the brickwork, checkerwork, or any other opening, or a leak in the sampling pipe, and it will be impossible to tell the difference, unless there is some other instrument or control device connected to the furnace indicating the amount of excess air. It is impossible, however, that a gas analysis which does not check with the H/C ratio of the fuel with a leaking sampling tube should check when the leak was removed, as described by the author in connection with the first case mentioned in the paper. An analysis will check or not check with the H/C ratio independently of the amount of excess air, whichever way this air got into the sample, through a leak in the sampling tube, or with the combustion air into the furnace. Accordingly, there must have been an error in the second analysis of the first case given by the author which can be demonstrated mathematically as follows: This analysis gave 11.4 per cent CO_2, 2.0 per cent O_2, and 0.0 per cent CO, consequently, the N_2 content was 86.6 per cent. After the analysis was taken, a leak was discovered and removed in the gas-sampling pipe line, with the result that the gas analysis changed to 11.7 per cent CO_2, 0.0 per cent O_2, and 0.0 per cent CO. In the following it will be proved that this change could not have been the result of the stopping of the leak.

Before the leak was stopped 100 cc of gas contained 11.4 cc of CO_2, 2 cc of O_2, and 86.6 cc of N_2. If as a result of stopping the leak 2 cc of O_2 were removed from the sample, a corresponding amount of N_2 equaling 7.75 cc also was removed, leaving a total of 90.25 cc, while the amount of CO_2 remained the same, that is, 11.4 cc. The latter amount, however, equals 12.6 per cent of the 90.25 cc, and not 11.7 per cent as the analysis taken after the removal of the leakage showed, consequently, there must have been some other cause than the leakage for the 2 per cent O_2 in the second analysis. It seems to be certain, however, that the second analysis, made immediately before the discovery of the leak, was the erroneous one, whatever the cause might have been, as it did not check with the H/C ratio of the fuel, while the others did. These two analyses indicate an H/C ratio of 0.33, which in turn checks with the specific gravity of the fuel gas given as 0.55 of the air, indicating methane, more than 99 per cent pure. The H/C ratio corresponding to the second analysis would have been 0.28, or in order to check with the 0.33 value the O_2 content should have been 0.4 per cent.

A more thorough mathematical evaluation of the results may lead to an explanation of the first gas analysis combined with the 10 per cent loss in the following manner:

It is possible that aldehydes, for instance formaldehyde, is present in the gases, and yet the analysis checks with the H/C ratio, provided some unburned hydrogen is also present. The presence of formaldehyde decreases the percentage of CO_2 in the gases, while that of hydrogen increases it, if the carbon that formed the methane with this hydrogen burns to CO_2. The quantities of formaldehyde and hydrogen may be present in such proportion that they balance each other, with the result that the gas analysis checks with the H/C ratio. To this ratio corresponds a ratio of the two quantities of methane, of the one quantity that was hydroxylated to formaldehyde and water vapor, to the other quantity that was dissociated to hydrogen and carbon, with the latter burned to CO_2. A simple calculation shows that if for 100 parts of methane decomposed with the carbon burned to CO_2 and the hydrogen escaping unburned, there are 35.3 parts of methane hydroxylated to formaldehyde and water vapor, with the latter condensed, both without any excess air, the resulting gas mixture will have a CO_2 content of 11.7 per cent, which is the same as when methane is burned to CO_2 and water vapor without excess air, that is, it will check with the H/C ratio. Evidently, any amount of this mixture can be added to the products of a complete combustion of methane, with or without excess air, and the analysis of the gases will check with the H/C ratio as if there were no hydrogen and formaldehyde in the mixture. This requirement, that for every 100 parts of decomposed methane there shall be 35.3 parts of methane hydroxylated, can be put down in the form of one equation.

Another requirement, furnishing another equation, is that the hydrogen and the formaldehyde present in the proportion just described shall represent a loss of 10 per cent. The solution of the two equations shows that this requirement is fulfilled if 11.4 per cent of the methane burned is decomposed and 4.05 per cent hydroxylated. The loss through decomposition is equal to the heat value of the escaped hydrogen, in this case 7.46 per cent, while the loss due to hydroxylation is equal to the heat value of the escaped formaldehyde, in this case 2.54 per cent of the heat value of the total methane burned.

Under these conditions, the hydrogen content of the flue gases will be 2.76 per cent, which is not impossible, as for instance Senner,[4] in his tests mentioned, found 2.2 per cent. At the same time the formaldehyde content will be 0.515 per cent by volume.

This explains the results of the first test with the exception that it admits decomposition, while the flame is described as not "luminous," that is, indicating decomposition, but "clear." As both decomposition and hydroxylation might have taken place, this point may or may not be a sure indication of the phenomena which actually occurred.

Figs. 1 and 2 of the paper are slight but handy modifications and extensions of Figs. 2 and 3 in Ostwald's "Contributions to Graphic Combustion Engineering."[5] These curves can be used to check the accuracy of the analysis in the Orsat as to the presence or absence of CO, but not the accuracy of the sample-taking from the point of view of possible admixture of air.

The otherwise well-known fuchsin test for aldehydes comes certainly very handy in connection with flue-gas analysis, as it is very simple; still it must be emphasized that it is only qualitative and not quantitative, and that even if used in combination with the Orsat it does not settle the question of hydrocarbons and carbohydrates other than aldehydes. Although the author

[4] "Domestic Oil Burners," by A. H. Senner, *Mechanical Engineering*, vol. 58, 1936, p. 705.

[5] "Beiträge zur graphischen Feuerungstechnik," by Walter Ostwald, Otto Spamer, Leipzig, Germany, 1920. An instance in American literature, Heating, Ventilating, Air-Conditioning Guide, 1942, p. 162.

mentions the presence of aldehydes in connection with gaseous and liquid fuels only, it would be worth while to investigate the flue gases of solid fuels as well. The importance of volatile matter in solid fuels appears if we consider that such matter may be contained in a coal to the extent of 35 per cent by weight only and yet represent 41 per cent of the heat value at the same time. Thus, elusive as the hydrocarbons and carbohydrates are when flue gases are analyzed, they may represent considerable losses. That carbohydrates may be present in the products of combustion of solid fuels in some special cases is illustrated in the case of Highland Scotch whiskey, which gets its special flavor due to the fact that the malt used in preparing it is dried in kilns fired with peat. All this is also a warning that flue gases of anthracite and some other "smokeless" fuels should be tested for hydrocarbons.

Altogether, the paper represents a very good contribution to our knowledge of flue gases and their analysis and is excellent proof of the faculty of observation and resourcefulness of the author.

Author's Closure

It was the intention of the author to call the attention of the power industry to the fact that existing methods of combustion control were inadequate. The inadequacy exists through human error as much as in any other manner it is true but this in no wise mitigates the urgent need to save every possible heat unit in the fuel being burned.

It is probable that through weight of sheer numbers of Orsat tests made by him, the author may consider himself quite well trained in Orsat technique, yet as has been pointed out in the discussion it is entirely possible that Orsat data as presented in the paper were incorrect.

Mr. Bato has pointed out evidence of error in the first field test as reported. The author agrees with Mr. Bato with reservations.

If increased draft should cause air to enter the flue-gas passage at a point where the temperature was above the kindling point of such unburned products as might exist, the complete or partial burning of these products might be expected to change the Orsat data obtained. It is also entirely possible for the incomplete combustion to occur in such manner as to cause the Orsat data to check the H/C ratio of the fuel. If such a condition existed before the draft increase it is entirely possible for such burning as might result through leakage after the draft increase to upset the Orsat-H/C relationship.

The question of quantitative as well as qualitative analysis of flue gases for aldehydes has been raised. It is the author's thought that such analysis is hardly justified for two reasons. The first is that if combustion exists in a marked shortage of O_2, products other than aldehydes may easily appear in the flue gases along with the aldehydes, therefore quantitative analysis of the aldehydes alone would not necessarily indicate the true fuel loss. The second reason is that if aldehydes are found in the flue gases the operator should be concerned with means for burning his fuel completely rather than with the fuel loss due to their presence.

Particular care should be taken to check for the presence of aldehydes in the flue gases coming from a furnace in which haze, flame, or terminal combustion exists in the immediate vicinity of cold surfaces such as the pass from the furnace into the boiler tubes, etc. Chilling of the gases in contact with the relatively cold tubes is a very potent source of incomplete combustion. So-called "cold" furnaces such as those in water-wall furnaces, oil-processing furnaces, steel and cast-iron boiler furnaces, or any furnace in which heat is transferred with extreme rapidity from the flame directly to the absorbing surfaces should be checked closely.

Investigation of Blade Characteristics

Performance and Efficiency of Turbine and Axial-Flow Compressor Stages

By J. R. WESKE,[1] CLEVELAND, OHIO

The investigation reported in this paper deals with the flow through blade rows of turbine or axial-flow compressor stages on the basis of airfoil theory. Correlation was established by means of this theory between the aerodynamic characteristics of blade elements and the performance characteristics of the blade row. Test results from a rotating grid of sufficiently low solidity to eliminate interaction of neighboring blades are presented. It was found that the coefficients of lift of blade sections in rotating grids, especially near the hub of the wheel, are higher than might be expected from wind-tunnel tests. Methods of experimental aerodynamic investigation were adapted to the rotating-blade grid for the determination of the profile drag, in an attempt to broaden the basis for a rational determination of the efficiency of turbine and compressor stages.

Introduction

CONSIDERABLE progress has been made in the calculation of the performance of turbine and compressor stages through the development of mathematical theory of potential flow of the working fluid through the blade row. Solutions have been obtained for the lift forces on blades in axial-flow blade rows as related to the lift force on a similar airfoil in the wind tunnel, for the performance of a radial-flow impeller with a given number of blades as related to the corresponding hypothetical impeller with infinite number of blades. Furthermore, the potential theory gives numerical data, e.g., in regard to the effect of thickness of profiles in axial-flow grids upon performance. Thus the potential theory is justifying its value in design, since the results obtained with it are in closer correlation with actual performance data than those of earlier theories.

These advances in potential-flow theory, which strictly apply to perfect fluid, accentuate the need for a logical procedure of calculation relating the known basic laws of fluid flow of the real fluid, in particular the laws of fluid resistance to the performance of the turbine or compressor stage. Of particular interest are (*a*) the effect of skin friction and turbulent-energy dissipation upon efficiency, and (*b*) the effect of fluid resistance in causing deviation from the performance characteristics postulated by potential-flow theory and in limiting the performance of stages.

Review of Earlier Work

The theoretical treatment of the problem of fluid resistance in blade rows and of its effect upon stage efficiency of turbines and compressors has received attention in publications of the following authors: Keller (1),[2] Ruden (2), Amstutz (3), Kraft and Berry (4), New (5), Dozinsky (6), and Soerensen (7).

Amstutz (3) analyzed the losses in a Kaplan turbine and was able to calculate an efficiency curve closely corresponding to the actual curve obtained by test.

Methods of direct measurement of the losses in fixed-blade rows by means of the wake-traverse method were developed by Kraft and Berry (4), by New (5), and are currently used as an effective aid in the design of turbines.

There is justification for the assumption that the three-dimensional nature of the flow through blade rows, whether of the axial- or radial-flow type, has a greater effect upon the efficiency in the stage than it has upon other factors of performance. Under certain conditions, as in the case of very short blades (aspect ratio: $\frac{\text{span}}{\text{chord}} \approx$ unity), the treatment of the flow as a two-dimensional flow may lead to results in regard to losses which are utterly at variance with actual losses. This fact was recognized by Dozinsky (6), who derives relations for the flow through blade rows from analogies drawn with the flow in curved ducts. However, it seems that as long as the nature of flow in curved ducts is not more fully understood in its details than at present, it would appear to be more expedient to analyze the problem of losses in blade rows in first approximation as a two-dimensional problem as done by Keller and Ruden. The effect resulting from the three-dimensional nature of the flow in that case would find consideration in the choice of the numerical quantities involved, or through correction factors applied to the results obtained by the two-dimensional approach. It is obvious that three-dimensional effects are more prominent in blades of small span than in blades of long span.

It is known that the results obtained from measurements of the efficiency of fixed-blade rows cannot be applied directly to moving rows even in the case where the main flow of the working fluid takes place on concentric cylinders. Attempts to measure with fixed instruments the losses in moving rows, however, encounter disproportionately greater difficulties because of the kinematic transformation required when applying the measurements obtained in the absolute system to the determination of the drag of the moving blades.

In view of this, an investigation of the blade characteristics of a rotating-blade row and of their relation to the performance characteristics of the row seemed of timely importance.

Theoretical Analysis

The performance of turbine stages can be related to the interaction of the fluid and the immersed blades, which in the last analysis is a result of the stresses in the fluid working on the surface of the blades. Distinction is made between normal stress or the pressure and the tangential stress or the shear stress because of basic differences of these two types of stresses in regard to their origin and their nature.

The distribution of pressure over the surface of the blade is largely a function of the flow in the main body of the fluid outside of the thin boundary layer in the immediate vicinity of the solid surface. This flow may be regarded as potential flow and can be calculated by mathematical methods (8, 9), developed for its determination.

The distribution of tangential or shear stresses, i.e., skin friction, on the other hand is related to fluid friction, including turbu-

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[1] Case School of Applied Science. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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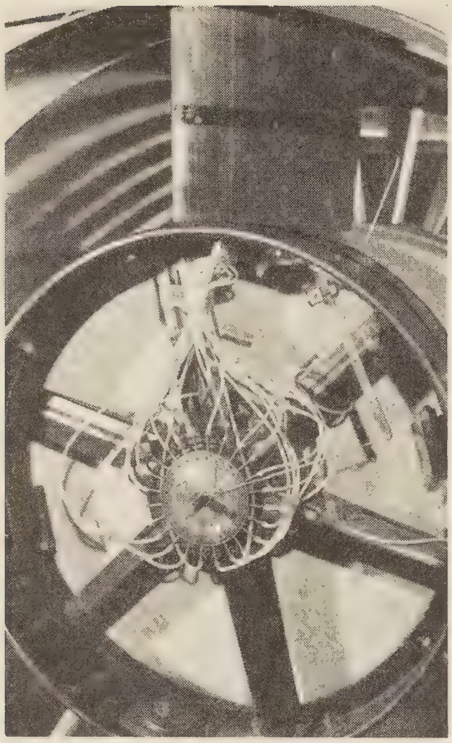


Fig. 1 Master Airfoil, Selector Device and Wake-Traverse Mechanism

lent momentum transfer. Shear stresses are a function largely of the flow in the boundary layer and depend to a lesser extent upon the pressure distribution over the surface of the blade. The tangential stresses are the source of losses in the blade row, and therefore affect the efficiency of the stage directly. It is seen that the division, originated by Prandtl, of the region of flow into a region in which the effect of viscosity can be neglected, in which the flow consequently may be regarded as potential flow, and into a boundary-layer region, is justified by the contrasting physical nature of the two flows. The present investigation is concerned primarily with the losses, and therefore with the flow, in the boundary layer.

It will be necessary to establish a relation between the factors influencing the interaction of the blades and the fluid resulting from motion of the two relative to each other and the performance data of the blade row. These relations are expressed in a number of equations which are conveniently presented in three groups, each group representing a distinctive link in the chain of the interrelation.

The three groups define the following functional relationships:

(*a*) Between the pressure and shear-stress distribution on the blade element and the resultant force and its direction.

(*b*) Between the force acting on the blade and the quantities of the velocity diagram.

(*c*) Between the quantities of the velocity diagram and the performance data.

Nomenclature

Equations will be formulated embodying the foregoing relations, in which the following nomenclature will be used:

r = radius
z = number of blades
l = chord
$t = \dfrac{2\pi r}{z}$ = pitch
$\dfrac{l}{t}$ = solidity ratio
ω = angular velocity
c = absolute velocity
w = relative velocity
β = angles in relative system
β_p = angle of profile section
$\alpha = \beta_p - \beta'$ = angle of attack
$'$ refers to mean condition
Subscript 1 refers to inlet condition
Subscript 2 refers to outlet condition
Subscript u indicates component in circumferential direction
Subscript m indicates component in axial ("meridional") direction
R = resultant force per unit radial length
N = component normal to chord of blade section
M = component parallel to chord of blade section
L = lift component
D = drag component
D_p = pressure-drag component
D_f = friction-drag component
S = component normal to grid
T = component tangential to grid
$C_L = \dfrac{L}{lq'}$ = lift coefficient
$C_D = \dfrac{D}{lq'}$ = drag coefficient
ρ = mass density
p = static pressure
η_r = efficiency of blade element
Δh = total head
$q = \dfrac{\rho w^2}{2}$ = velocity pressure

(*a*) The pressure distribution about a blade section may be obtained theoretically by potential-flow theory or experimentally, e.g., by the method described hereafter. It may be presented as a plot against distance along the chord of the blade section and against distance normal to the chord. Integration of the area under these two curves leads to the component normal to the chord N, and the chordwise component M, of the resultant-pressure force per unit length of blade along the radius, R. Let the component of the resultant force parallel to the grid be denoted by T, normal to the grid by S. Before the lift component L, defined as the component normal to the direction of the mean relative velocity w' past the blade, and the drag component D parallel to w' may be found, this velocity is to be determined from the equation

$$w' = \sqrt{\overline{C'_m}^2 + \left(r\omega - \frac{C_{u1} - C_{u2}}{2}\right)^2} \qquad [1]$$

and its direction from

$$\tan \beta' = \frac{C'_m}{r\omega - \dfrac{C_{u1} - C_{u2}}{2}} \qquad [2]$$

where

$$C'_m = \frac{C_{m1} + C_{m2}}{2}$$

These and the other quantities, defined in the nomenclature, may be identified in Fig. 3.

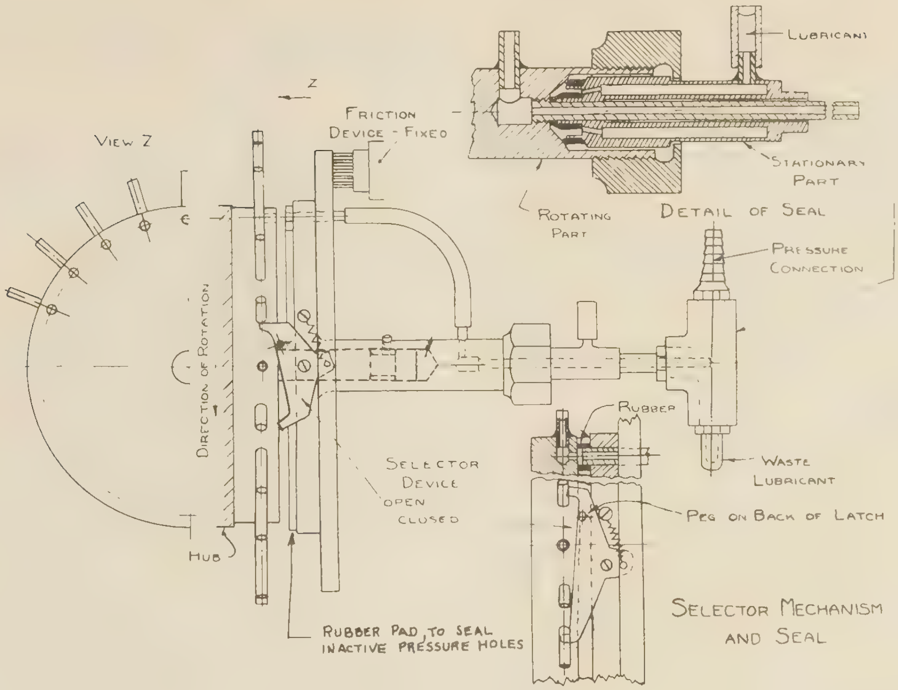


Fig. 2 Selector Valve and Seal

The drag component, obtained from pressure distribution, constitutes only a part of the drag of the blade section, namely, the pressure drag D_p. For conditions of operation of the grid sufficiently close to design conditions so that the circulation

$$\Gamma = (C_{u1} - C_{u2}) \frac{2\pi r}{z}$$

does not vary appreciably along the radius, the induced drag may be neglected, and the skin-friction drag D_f only need be considered, in addition to D_p, since in that case $D = D_p + D_f$. Methods of calculation (10) have been developed by which the skin friction may be calculated theoretically from local shear stresses for airfoils in indefinite flow. These calculations, however, are tedious and do not appear to be readily applicable to blade grids. Experimentally the most promising approach seems to be the measurement of wake traverses, discussed in a later section, which yields the profile-drag directly.

The following nondimensional forms have been used in calculations relating to blade forces

Lift coefficient $\qquad C_L = \dfrac{L}{lq'}$

Drag coefficient $\qquad C_D = \dfrac{D}{lq'}$

further

$$C_T = \frac{T}{lq'} = C_L \sin \beta' + C_D \cos \beta' \qquad [3]$$

$$C_S = \frac{S}{lq'} = C_L \cos \beta' \pm C_D \sin \beta' \qquad [4]$$

where the upper sign applies to turbine the lower to compressor operation.

(*b*) In calculating the velocity diagram of the moving row, it is assumed that the inlet conditions and the axial-velocity component at the outlet are known. The velocity diagram may then be drawn after the change of circumferential-velocity component has been calculated from the momentum equation

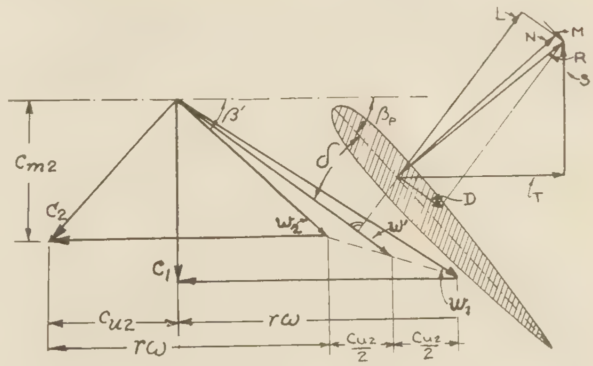


Fig. 3 Velocity and Force Diagram

$$2\pi r C'_m \rho' (C_{u1} - C_{u2}) = zT$$

which may be reduced to

$$C_{u1} - C_{u2} = \frac{l}{t} \cdot \frac{C_T}{2} \cdot \frac{C'_m}{\sin^2 \beta'} \qquad [5]$$

(*c*) The total head Δh across the blade element is given by the energy equation

$$\Delta h = \frac{zSC'_m}{2\pi r C'_m \rho' g} = \frac{C_S}{\sin^2 \beta'} \cdot \frac{l}{t} \cdot \frac{\overline{C'_m}^2}{2g} \qquad [6]$$

and the efficiency of the blade element

$$\eta_r = \frac{C_T r\omega}{C_S C'_m} \qquad [7a]$$

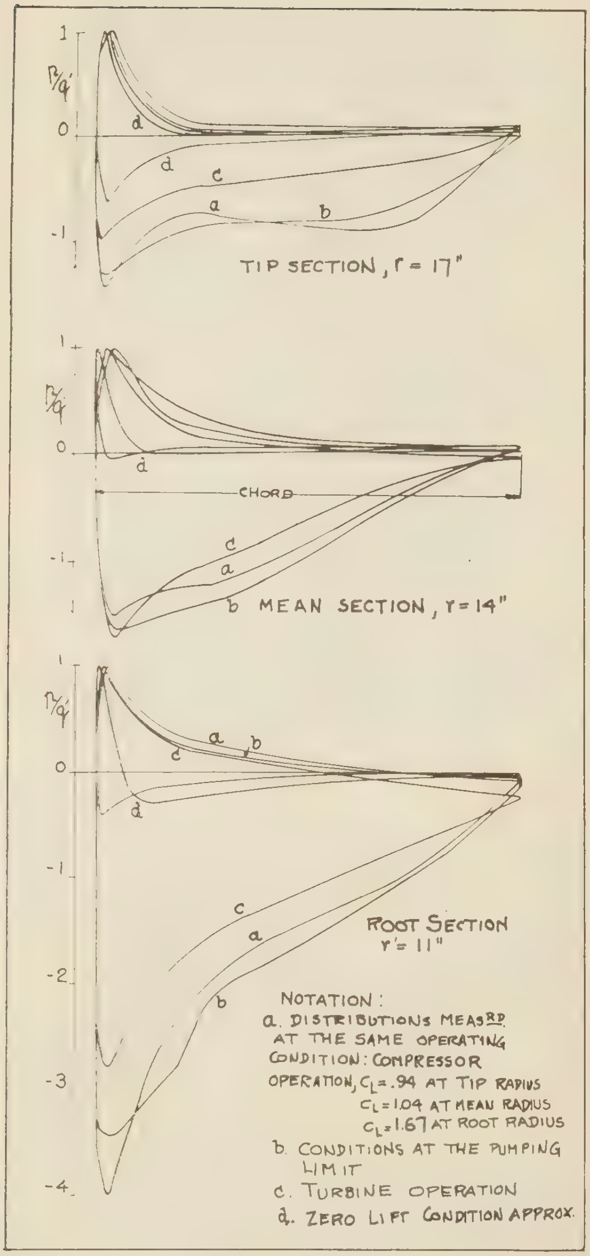


Fig. 4 (*a* to *c*) Nondimensional Pressure Distribution p/q for Tip, Mean, and Root Section of the Blade
(Grid with a single airfoil; solidity $1/t$ = 0.09.)

for turbine operation and

$$\eta_r = \frac{C_S C'_m}{C_T r\omega} \qquad [7b]$$

for compressor operation.

The efficiency, as formulated here, being based upon lift and profile drag of the blade element, does not take account of certain losses resulting from turbulent momentum transfer associated with the decay of the wake downstream of the row. These losses, however, are small compared with the profile drag in the case of two-dimensional flow for rows of conventional solidity ratios. The equations given in this section are applicable to both compressible and incompressible flow.

Experimental Work

In developing an experimental program, it seemed important to reduce the test conditions to the greatest possible simplicity. To this end choice was made of a symmetrical profile for the blade section. The direction of flow upstream was chosen to be axial for turbine and compressor operation. Test results of the single-bladed grid only have been included in the analysis, in order to exclude problems of blade-to-blade interference.

Experimental investigations were conducted on a rotating axial-flow grid of 36 in. tip diam and 20 in. root diam, consisting of blades of symmetrical airfoil section N.A.C.A. 0010, twisted for constant geometric pitch when the angle at the root section is 45 deg. The rim was rigidly connected to the blades and rotated with the wheel, thus adding to the rigidity of the latter and eliminating tip leakage. The blades have a chord of 8 in. The number of blades in the grid may be varied and grids of 1, 2, 3, 4, 6, or 12 equally spaced blades may be tested giving a range of solidity ratios $= \frac{\text{chord}}{\text{pitch}}$ on mean diameter of from 0.09 to 1.09. Spacer bolts were provided through each blade, or in place of the blades to hold rim and hub together. One of the blades referred to as the master blade, Fig. 1, was made of lucite and provided with numerous pressure holes arranged in three cylindrical sections, namely, 23 each on the mean radius, 1 in. above the root radius and 1 in. below the tip radius.

The pressure holes in each cylindrical section can be connected successively to a stationary pressure manometer by means of a selector device and seal, shown in Figs. 1 and 2, the latter figure explaining its mechanism. An earlier design of the device was described in (10).

The grid was mounted on the shaft of an electric motor and made a part of a test setup, also described in (10), which was placed into the working section of the return-type wind tunnel of the aerodynamics laboratory of Case School of Applied Science.

Provisions were made to operate the grid both as a compressor and as a turbine. The grid was rotated at speeds ranging from 500 to 1000 rpm. Conditions of operation were varied (*a*) by varying the quantity of air circulated in the wind tunnel, and (*b*) by varying the passage area of an adjustable diaphragm located at the outlet of the cylindrical section, 4 ft downstream of the grid, shown in Fig. 1. This diaphram was found necessary to obtain operation at and beyond the pumping limit, for compressor operation of the grid.

Since the blades are twisted for constant geometrical pitch at the setting just indicated the condition of constancy of circulation, hence of flow on concentric cylinders, is fulfilled only at the point of operation which divides the regions of turbine and compressor operation. Deviations from the assumed condition of flow on concentric cylinders, while inappreciable at small lift coefficients, are presumed to become more severe as the lift coefficients become large. These deviations cause a variation of axial-velocity component between inlet and outlet for which reasons each of these quantities had to be determined separately.

The following measurements were taken:

1 Total- and static-pressure measurements both upstream and downstream of the grid.

2 Measurements of static pressures at the hub and at the casing, both upstream and downstream.

3 Pressure distribution measurements.

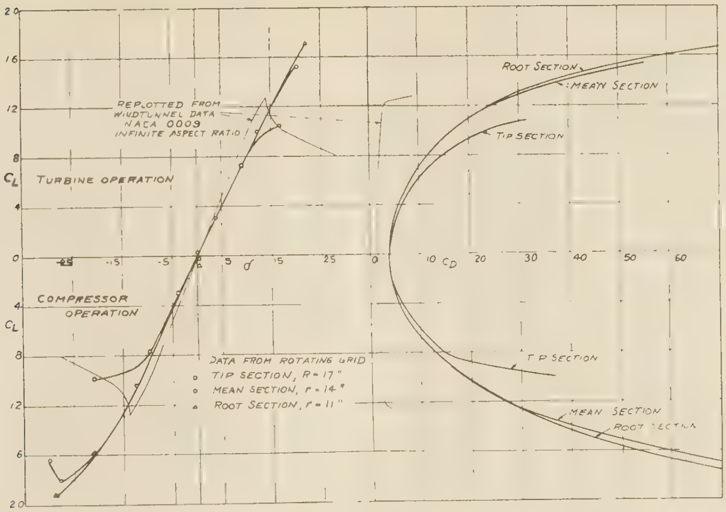
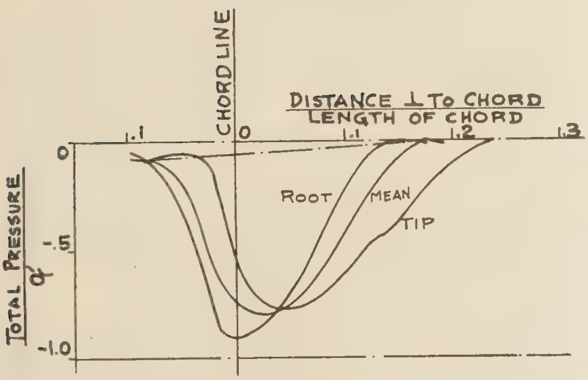


Fig. 5 Lift and Pressure-Drag Coefficients, Tip, Mean, and Root Sections
(Solidity l/t = 0.09.)



RADIUS:	TIP	MEAN	ROOT
ANGLE OF BLADE	29	37	41
LIFT COEFF.	.92	1.04	1.67
RATIO WAKE AREA / q'	0.14	0.115	0.116

Fig. 6 Wake Traverses

4 The rotative speeds were held constant at specified values by a stroboscopic device.

5 Special measurements were obtained of the profile drag of blade sections by means of wake traverses.

Test Results

Measured pressure was plotted along the chord of the blade section and normal to the cord. The force components per unit length of blade in a radial direction normal and tangential to the chord, respectively, were obtained from these pressure-distribution diagrams and the corresponding resultant force by vector addition, Fig. 3. Not included in the latter is the skin-friction drag which will be treated later.

The data of the velocity diagram, Fig. 3, were calculated directly from measured data, namely, C_{m1}, C_{m2}, and $r\omega$, or from pressure-distribution measurements, namely, C_{u2} from Equation [5], ω' and β' from Equations [1] and [2].

The lift coefficient $C_L = \dfrac{L}{lq'}$ and the pressure-drag coefficient $C_{D_p} = \dfrac{D_p}{lq'}$ were calculated from the respective components of the resultant force, Fig. 2.

The experimental data thus obtained are presented in the following figures:

Fig. 4 (*a* to *c*) shows nondimensional pressure distributions $\dfrac{p}{q'}$ for tip, mean, root radius, for one airfoil in the grid, solidity 0.09.

Fig. 5 shows lift and pressure-drag coefficients, solidity 0.09. For purpose of comparison, Fig. 5 also shows the corresponding coefficients from wind-tunnel tests, reduced to infinite aspect ratio, replotted from (12). The large increase of measured drag coefficients over corresponding wind-tunnel values may be ascribed partly to impairment of the aerodynamic properties of the airfoil surface by the pressure holes and partly to low effective aspect ratio.

It was of interest to ascertain experimentally the magnitude of the profile drag of the blade sections, only a part of which is represented by the pressure drag. To this purpose a total-pressure tube was mounted on the rotating hub placed at a distance of 1.2 in. in the chordwise direction downstream of the airfoil, as shown in Fig. 1. This tube was pointed upstream in the direction of the mean relative flow and connected through the seal to a stationary pressure gage. A simple mechanism was constructed by which the total-pressure tube could be moved between measurements a known small distance in the circumferential direction through operation of the selector device. In this manner, complete total-pressure traverses of the wake were obtained without requiring stoppage of the unit between individual measurements.

While in the present stage, the results of these tests fail to give all the information needed to determine the profile drag precisely, due to the fact that it was not possible to obtain static-pressure readings in the wake simultaneously and coincidently with the total-pressure readings, nevertheless they permit a qualitative insight.

Fig. 5 shows three typical wake traverses obtained from the grid with a single airfoil in place (solidity ratio 0.09) at the same condition of operation at three different radii, namely, at 1 in. above the hub, at the mean radius, and at 1 in. below the rim. Given are the lift coefficients obtained from pressure-distribution measurements, and the areas of the wake profiles. It is noted

that the wake area divided by the velocity pressure of mean relative velocity is smaller for the mean blade section than the corresponding quantity for the section near the tip, although the lift coefficient of the latter is smaller. If the ratio, wake area/velocity pressure of mean velocity, may be taken as proportional to the profile drag, it may be concluded that contrary to what might be expected from the two-dimensional theory, the profile drag near the tip is larger. Three possible explanations are:

1 It may be argued that, since the circulation is not constant along the radial extent of the blade, this result may have been caused by a displacement of the flow outside the wake.

2 The apparent increase of profile drag near the tip may be the result of losses from interference originating in the corners between the blade and the rim. That this latter cannot be the only cause is brought out by the fact that a correspondingly larger value of the ratio was not measured at the section near the hub.

3 The boundary layer of the airfoil and likewise the wake are displaced radially outward by centrifugal force. In order to substantiate the theory of radial displacement of the wake, unexposed Ozalid paper was glued on the surface of the blade around certain of the pressure holes. A mixture of air and ammonia vapor was then emitted by way of the pressure seal through these holes, and an indication of the direction of flow in the boundary layer was obtained by a darkened streak produced by the ammonia vapor on the Ozalid paper. Available photographs of these records are not suitable for reproduction but they indicate clearly that at all points there is an appreciable radial velocity in the boundary layer, particularly on the suction side of the airfoil.

It is concluded from the foregoing that, in determining the profile drag of a blade section of rotating axial-flow grids, account must be taken of the three-dimensional nature of the flow in the boundary layer and in the wake of the blade.

Performance Characteristics

The performance characteristics of the rotating-blade row were calculated from the results of the pressure-distribution measurements for the three blade sections investigated, namely, the total head from Equation [6], and the efficiency of the blade element η_r from Equations [7a] and [7b]. The results are plotted in Fig. 7. Also plotted on this figure for comparison are corresponding curves calculated from the wind-tunnel characteristics of the symmetrical airfoil profile N.A.C.A. 0009, for infinite aspect ratio. The experimental efficiency curves are based upon the pressure drag only, and no account has been taken of the friction drag of the blade sections.

Conclusions From Experimental Investigation

The following conclusions are derived from the results of the experimental investigation:

1 Tests of blades, placed at wide spacing in a rotating-blade row, show that the range of lift coefficients for the root and mean section exceed the range available for the fixed blade, and maximum-lift coefficients are higher than corresponding wind-tunnel data for compressor operation, as well as for turbine operation.

2 Stalling of the three blade sections investigated was found to occur at the same condition of operation. The stalling characteristics of the blade as far as could be discovered from pressure-distribution measurement, were much less critical than those of the same profile obtained in the wind tunnel.

3 Evidence is presented by the tests that the displacement of the boundary layer and of the wake of the blades radially outward by centrifugal force affects both the lift characteristics and the profile drag, in both cases to the advantage of the root sections of the blades.

4 Performance curves of blade sections, based upon experimental data calculated from pressure-distribution measurement of a rotating-blade row, show lower efficiencies and a higher pumping or stalling limit than corresponding characteristics calculated from airfoil data obtained in the wind tunnel.

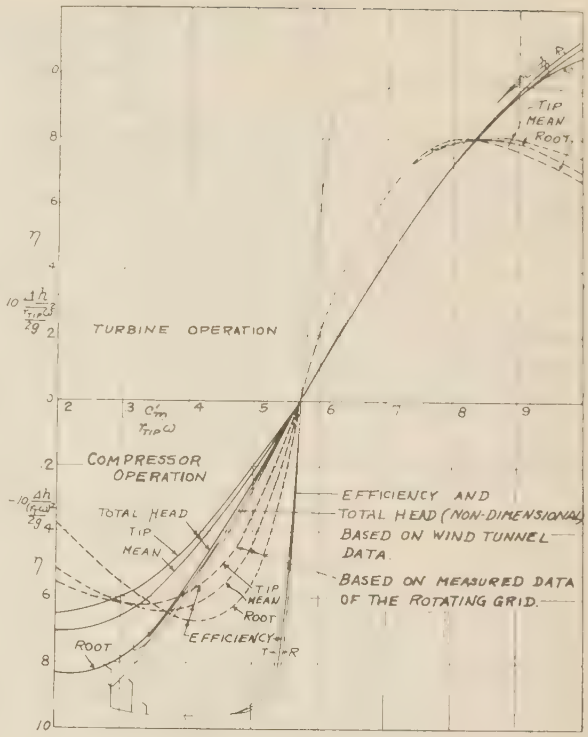


Fig. 7 Total Head and Efficiency

Acknowledgment

Acknowledgment is made to Messrs. Morris Sable, and Robert Christiansen, students of Case School of Applied Science, who constructed equipment and assisted in the experimental work.

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Discussion

G. F. Wislicenus.[3] This paper is a most important contribution and one whose significance is obvious at this time, in view of current developments in the field of compressors and turbines. While investigations of a similar nature are being carried out in the industry, to the best knowledge of this writer no one particular study duplicates the author's work. At Worthington, for example, there has been in progress for some time a series of investigations on parallel-vane systems. As a consequence of detailed work in this line, the writer is only too familiar with the inherent difficulties of the problem. The author is to be congratulated on overcoming some of the more serious technical difficulties involved, particularly in the test arrangement.

Perhaps his most valuable single finding is the experimental confirmation of the important difference between the flow in a rotating-vane system, as compared with that in a parallel stationary-vane system, that difference being the "radial" motion of the fluid particles in the boundary layer. While this writer, and doubtless other engineers, had suspected this to be a fact, the experimental proof is of great significance. The author's findings are conclusively demonstrated by his comparison between vane characteristics obtained in the wind tunnel and measurements on the inside of the rotating vane. The writer is referring to the ingenious use of ammonia-vapor traces on Ozalid paper, as well as to the critical study of the change of the maximum lift coefficient. The verification of the idea of radial motion in the boundary layer would alone justify the paper.

As for the theoretical aspects, an explanation of the difference between the through-flow components C_{m2} and C_{m1} (Fig. 3) would be helpful. From the information given in the paper, it seems evident that both the hub and outside periphery are cylindrical; in that case wherein lies the difference between C_{m2} and C_{m1}? From the velocity diagram it is seen that C'_m is assumed to be the arithmetic mean between C_{m2} and C_{m1}. This cannot be explained by the contraction due to the thickness of the vanes, since this thickness would cause C'_m to be larger than either C_{m2} or C_{m1}. Use of the arithmetic mean seems to indicate that the variations in C_m refer to the compression of the fluid. However, for the dimensions and speed of rotation given, the compression of the air must have been practically negligible, particularly with a single-vane grid. The velocity diagram is stated to have been constructed on the basis of measured data. Perhaps some amplification of this statement would clarify the points just mentioned.

The writer assumes that the "pressure drag" is that drag which is caused by a change in the static pressure distribution due to surface friction, since obviously in an ideal fluid the pressures must necessarily be distributed in such a manner that there exists no component in the direction of the flow and, consequently, no pressure drag. The differentiation between pressure drag D_p and the skin-friction drag D_f is new to this writer and an explanation would be appreciated.

In Equation [6] of the paper, Δh is stated to be the total head across the blade element. As given in this equation, it represents rather the change in the static head only, since it expresses the work due to the axial motion under the influence of the axial component S of the vane forces; it does not take into account the change in kinetic energy due to the change in rotation. As a consequence of this fact, the turbine efficiency (Equation [7a]) is actually too high and the compressor efficiency (Equation [7b]) too low. (Equation [7b] is strictly valid for a ship or airplane propeller.) With the present test arrangement, using only a single vane, this discrepancy is quite small, but would be appreciable in actual turbine or compressor wheels.

The author's conclusions are of great interest. It would be helpful if he would explain further what is meant by the statement that the stalling characteristics of the blade were "much less critical." Fig. 5 seems to indicate that, with the exception of one of the curves, the change in lift coefficient during stalling appears to be less drastic than in wind-tunnel tests. Is this the correct interpretation of item 2 of the "Conclusions?"

Does the difference (Fig. 5) between the measured drag and wind-tunnel data hold true for current designs, or has this particular setup certain features which may be responsible for this difference as well as for the low efficiency of the compressor (Fig. 7)? It is of interest to note that the drag coefficients of the turbine and the compressor bear a close resemblance (Fig. 5) but the efficiencies are quite different.

With respect to results obtained with a single vane, the average pressure change through the system would be practically negligible. This is probably responsible for the fact that the maximum-lift coefficients (Fig. 5) are nearly the same for compressor and turbine operation. If the vanes were to be more closely spaced, the pressure increase or drop through the runner would become appreciable as compared with the local pressures on the vanes. Consequently, the vane characteristics in turbine and pumping operation could be expected to differ considerably, the compressor vane acting in a field of increasing static pressure, and vice versa.

The writer is wholeheartedly in agreement with the author's policy of starting his investigations under the simplest conditions possible. It would be most valuable if these experiments could be extended to include more closely spaced vanes. Provisions have evidently been made to increase the solidity ratio to about 1.09. An even higher ratio would yield profitable results, possibly in connection with slightly thinner and suitably curved vanes. It is particularly with such higher solidity ratios that results such as those given in this paper would be of the greatest practical value for compressor and turbine development. For this reason, continuation of this work would constitute a real help to the industry.

Additional details of the technique of pressure measurements in a rotating system and the derivation of the methods of evaluation would be of great interest to many engineers who have been working with similar mechanisms. For instance, Mr. Arthur Oschwald, Jr., of the writer's company brought up the questions: How is it possible to check the measuring system for leakage during rotation; and what steps have been taken to minimize the effect of any possible leakage on the readings? It is hoped that these and other questions can be covered by subsequent publication of information on this interesting and important line of research.

Author's Closure

Dr. Wislicenus' generous comments and constructive criticism are greatly appreciated. An attempt is made to answer the questions raised in the discussion in the following paragraphs.

The velocity and force diagram, Fig. 3, of the paper, was drawn to explain the general relations involved and does not represent any of the diagrams used for the calculation of a particular operating condition. The variation of axial-velocity component was intentionally assumed to be large to show the relation of the mean relative velocity to the relative velocities at inlet and outlet. As pointed out in the discussion, variations of axial velocity are, in the absence of compressibility effects, related to variations of the axial-flow area. A virtual decrease of the axial-flow area downstream of the grid is caused by the wakes of the blades which

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is of significance, however, only in grids of high solidity. No consideration has been given in the paper to the effect of blade thickness.

The designation "pressure drag" was used with reference to the drag calculated from the measured pressure distribution about the airfoil.

The equations for the efficiency, Equations [7a] and [7b], apply specifically only to the blade grid under investigation, namely, to a single rotating grid without stationary grid upstream or downstream.

As an expression for the efficiency generally applicable to an annular element of a stage consisting of two grids, indicated by subscripts I and II, respectively, either one moving or both moving (the latter case being that of a counter-rotating stage), the following equations, corresponding to Equations]7a] and [7b] are proposed:

$$\text{turbine} \qquad \eta_r = \frac{(C_T r\omega)_{\text{I}} + (C_T r\omega)_{\text{II}}}{(C_s C'_m)_{\text{I}} + (C_s C'_m)_{\text{II}}}$$

$$\text{pump} \qquad \eta_r = \frac{(C_s C'_m)_{\text{I}} + (C_s C'_m)_{\text{II}}}{(C_T r\omega)_{\text{I}} + (C_T r\omega)_{\text{II}}}$$

It is seen that, for the case of the experimental grid, these equations reduce to the form given in the paper. The form of the equation as just given should have merit in calculations relating stage performance directly to the basic airfoil characteristics. They are believed to apply without restriction to the ducted blade grid.

Available information is insufficient to analyze fully the measured efficiencies, Fig. 7. The low values may be explained partly by the absence of the fixed grid, further by the influence of particular features of the experimental equipment, such as the increase of effective roughness of the blade by the pressure holes, and the absence of fillets at the root and the tip of the blade. Losses in grids and their causes are receiving special attention in further investigations. An extensive investigation of these and other factors has been made by A. Meldahl.[4]

In regard to the experimental demonstration of the occurrence of radial velocities in the boundary layer of the blades of an axial-flow grid, it should be stated that the author, in 1934, presented photographic records of streamlines,[5] obtained by the same method as described in the paper. The same method was applied by P. Ruden to the investigation of an axial-flow fan.[6]

A simple method was developed for checking the experimental pressure-measuring apparatus for leaks. All pressure holes on the airfoil were closed off by a strip of Scotch tape, and a run was taken at which pressure measurements were obtained successively for all pressure connections. Leakage in the rotating portion of any of the pressure leads was then indicated by a negative pressure on the manometer, good connections giving zero reading. The magnitude of the suction pressure served as a clue for the location of the leak, as it is a rough indication of the radial distance of the leak from the axis of rotation.

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Notch-Toughness Tests of Carbon-Molybdenum Pipe Material

By W. F. KINNEY,[1] I. A. ROHRIG,[2] and H. S. WALKER[3]

The development of materials for high-temperature-steam piping is of considerable importance in view of the upward trend in power-plant operating temperatures. Chemical composition, steel-melting practice, and heat-treating procedure, and their effects upon physical properties and microstructure are important factors in appraising the probable behavior of a material in service. The metallurgist and power-plant engineer are particularly interested in the influence of these factors upon notch toughness of the material. Starting with the premise that there may be some correlation between notch toughness and performance of high-temperature pipe and that significance should be attached to the interpretation of notch toughness with respect to the behavior of carbon-molybdenum pipe material in power-plant service, the authors give details of an investigation conducted in The Detroit Edison Company laboratory on notch-toughness testing. The specific problem studied was to determine the influence on the uniformity of test results and on the magnitude of average notch-toughness values, both at room temperature and at 925 F, of three variables, namely, (*a*) the type of specimen, (*b*) the heat of steel, and (*c*) the condition of the steel. In connection with this testing program, a survey of the literature was conducted and inquiries were directed to various authorities regarding their experience with this problem. The authors conclude that when the significance of notched-bar testing of carbon-molybdenum pipe material is fully understood, it may be possible to select material of the best quality for high-temperature service on the basis of notched-bar tests.

Introduction

As a result of the upward trend in steam-power-plant operating temperatures, considerable attention has been directed toward developing materials for high-temperature-steam piping. As a part of the development, much discussion has taken place regarding the application of carbon-molybdenum steel to superheated-steam piping service and the criteria by which its expected behavior in this service may be predicted. Chemical composition, steel-melting practice, and heat-treating procedure, each with its attendant effects upon physical properties and microstructure, are important factors in the appraisal of the probable behavior of a material in service. The influence of these factors upon one of the physical properties, namely, the notch toughness, is of interest to the metallurgist and the power-plant engineer, because it is believed that notch-toughness data reveal properties that cannot readily be evaluated on the basis of the usual hardness and tensile tests (1).[4]

It is recognized that the property of notch toughness is seldom used as a a measure of acceptability of carbon-molybdenum steel for high-temperature pipe, and notched-bar test failures generally have not been regarded as being analogous to pipe failures in the field. In this regard, McAdam and Clyne (2), in referring to impact tests using notched and unnotched specimens, stated: "An impact test should not be viewed as a simulation of shock in service, and should not be limited to metals that are to be subjected to impact in service." Further, Hoyt (3), in reviewing the likelihood of service failures, has stated: "We have another group of 'internal' notches which are less subject to control because they are not usually visible. Examples of these are . . . heat checks, grinding cracks, shrinkage cracks in castings and welds . . . To these may be added the fatigue crack which forms a very effective notch . . . Corrosion pits and corrosion cracks may also produce a serious notch effect."

Therefore it seems unwarranted to assume that there is no correlation between notch toughness and performance of high-temperature pipe; and a very real significance should be attached to the interpretation of notch-toughness data with respect to the behavior of carbon-molybdenum pipe material in power-plant service. However, information and test data of this nature are meager and controversial issues have been raised from time to time regarding notch toughness, particularly with reference to the apparent lack of uniformity of V-notch test results and the difference in magnitude of average V-notch values as compared with those of keyhole-notch or U-notch results.

In this regard, it might be well to consider for a moment the viewpoint expressed by Hoyt (4) in his statement: "Acceptance of the notched-bar test has not been at all unanimous. Perhaps the chief reason for this is that some experimenters report that the results are apt to be decidedly irregular among themselves and that they give one a feeling that the test lacks meaning and precision. On consecutive heats of steel the results are apt to vary considerably though they are made to the same specification and are otherwise replicas of each other. With such uncertainty and fluctuation it is difficult to correlate the results with other properties or with service records." With reference to tests upon groups of specimens from a material, Hoyt comments further (4): " . . . . . While the results are usually in satisfactory agreement one may and frequently does get quite wide variations . . . . . Does this lack of agreement reflect vagaries of the test itself or does it reveal actual variations in the material? . . . . . One frequently hears that the results are unreliable, by which is meant that, due to imperfections in the testing procedure, sometimes high results and sometimes low results will be secured from identical samples. The chance causes of the spread are assigned to the test procedure and not to the material itself." After further description of tests that were conducted to clarify this question, and in which satisfactory agreement among specimens of a homogeneous sample was obtained, Hoyt (4) concluded, "I could no longer ascribe real variations in test results to vagaries of the test and I believe the point is now seldom raised any more."

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Carbon-molybdenum pipe material is subject to these uncertainties regarding notch-toughness testing. The present investigation in The Detroit Edison Company grew out of an attempt to account for discrepancies that were noticed between notch-toughness values obtained at an outside laboratory and at the Company's research laboratory on supposedly identical Charpy V-notch specimens, when only one or two specimens were tested at a time. Subsequent tests at both locations on a large number of specimens from the same carbon-molybdenum pipe sample in the "laboratory-normalized at 1725 F" condition revealed that the average values and range of values obtained on the two machines checked surprisingly well. It was thereupon concluded that, if a sufficient number of carefully prepared Charpy V-notch specimens were tested, a reasonably close check should be obtained on the material under consideration but that a wide variation in notch-toughness values among individual specimens in any group might be expected. However, this conclusion was predicated solely upon room-temperature tests of Charpy V-notch specimens of normalized pipe, and the question was immediately raised as to whether this conclusion were also warranted for tests both at room temperature and at operating temperature, on various types of standard specimens, on various heats of carbon-molybdenum pipe material, and in various heat-treated conditions encountered in the company's high-temperature piping investigations.

The specific problem therefore was to determine the influence on the uniformity of test results and on the magnitude of average notch-toughness values both at room temperature and at 925 F, of three variables, namely, (*a*) the type of specimen, (*b*) the heat of steel, and (*c*) the condition of the steel. In connection with this testing program, a survey of the literature was conducted and inquiries were directed to various authorities regarding their experience with this problem.

Test Procedure

The tests were conducted on specimens of 10 and 12 in-nps, schedule 100 carbon-molybdenum pipe material from a number of heats which have been furnished commercially during the past 5 years and which represent the more recent installations of this steel. The material tested was made according to the silicon-killed coarse-grained melting practice. The pipe sections were received in either the "hot-rolled and drawn at 1200 F" condition or the "shop-normalized[5] at 1725 F" condition;[6] the former provided the samples, approximately 4 × 6 in. in size, which were laboratory-normalized for 1 hr at the desired temperature in an electric muffle furnace. Specimens were cut longitudinally from the mid-section of each sample, following a pattern that distributed the various types of specimen at random throughout the sample. Notches were cut in that surface of the specimen nearest the outer surface of the pipe wall. Care was taken to see that test specimens were as nearly identical as possible and truly representative of the pipe section from which they were removed.

After a survey was made of the specimen dimensions recommended by A.S.M.E. (5), A.S.T.M. (6), A.S.M. (7), and in various technical articles, the dimensions shown in Fig. 1 were selected as corresponding to approved standards in the essential details.

The scope of the tests included the following:

[5] This term, as used in this paper, describes the normalizing treatment given to pipe in the fabricating shop and is intended to distinguish that procedure from laboratory normalizing of test samples. It is advisable to preserve such a distinction in all laboratory investigations of heat-treated material.

[6] In accordance with A.S.T.M. Designation A206-41T, except that the normalizing temperature specified by the authors' company was 1725 F rather than 1700 F.

TABLE 1 NOTCH TOUGHNESS AT ROOM TEMPERATURE OF A TYPICAL HEAT OF COARSE-GRAINED CARBON-MOLYBDENUM PIPE MATERIAL USING VARIOUS TYPES OF TEST SPECIMEN[a]

Type of specimen	Notch toughness, ft-lb	
Charpy V-notch	59	
	82	Avg 77
	87	
	78 [b]	Max 87 (+13%)[c]
	72	
	80	Min 59 (—23%)[c]
	78	
	82	
	83	Avg 78
	79	
	86	Max 87 (+12%)
	87	
	55	Min 55 (—30%)
	77	
	74	Avg 80
	83	
	72	Max 86 (+ 8%)
	86	
	84	Min 72 (—10%)
Charpy keyhole-notch	46	
	47	Avg 47
	48	
	45	Max 49 (+ 4%)
	49	
	48	Min 45 (— 4%)
	47	
Charpy U-notch	46	
	47	Avg 47
	46	
	47	Max 48 (+ 2%)
	46	
	47	Min 46 (— 2%)
	48	
Izod V-notch	68	
	59	Avg 65
	61	
	68	Max 70 (+ 8%)
	70	
	62	Min 58 (—11%)
	66	
Tensile impact	150	
	150	
	152	
	181	Avg 169
	179	
	182	Max 197 (+17%)
	176	
	160	Min 150 (—11%)
	151	
	181	
	174	
	197	

[a] Heat 5085, laboratory-normalized at 1725 F for 1 hr.
[b] Tests conducted at an outside laboratory.
[c] Variation of maximum and minimum values from average value in each group.

1 Tests on 19 Charpy V-notch specimens from a typical heat in the "laboratory-normalized at 1725 F" condition, 7 of which were tested at an outside laboratory.

2 Tests of 7 each of Charpy V-notch, Charpy keyhole-notch, Charpy U-notch, Izod V-notch, and tensile impact specimens all from the same heat in the "laboratory-normalized at 1725 F" condition.

3 Tests at room temperature and at 925 F of 2 typical heats in the "hot-rolled and drawn at 1200 F" condition, using Charpy V-notch, Charpy keyhole-notch, and Charpy U-notch specimens.

4 Tests at room temperature and at 925 F of the same two typical heats in the "laboratory-normalized at 1650 F" condition.

5 Tests at room temperature and at 925 F of the same two heats and two additional typical heats, all in the "laboratory-normalized at 1725 F" condition.

These data were augmented by data on file from inspection tests on several heats that had been shop-normalized at 1725 F.

The V-notch and U-notch specimens were machined using milling cutters whose profiles had been checked and were known to be within the specified tolerances. The keyhole notch was prepared in the standard manner using a fixture to align the specimens and guide the drill. The dimensions of all specimens were checked by means of a sharp-pointed needle on the shaft of a dial gage which was rigidly mounted above a ground face

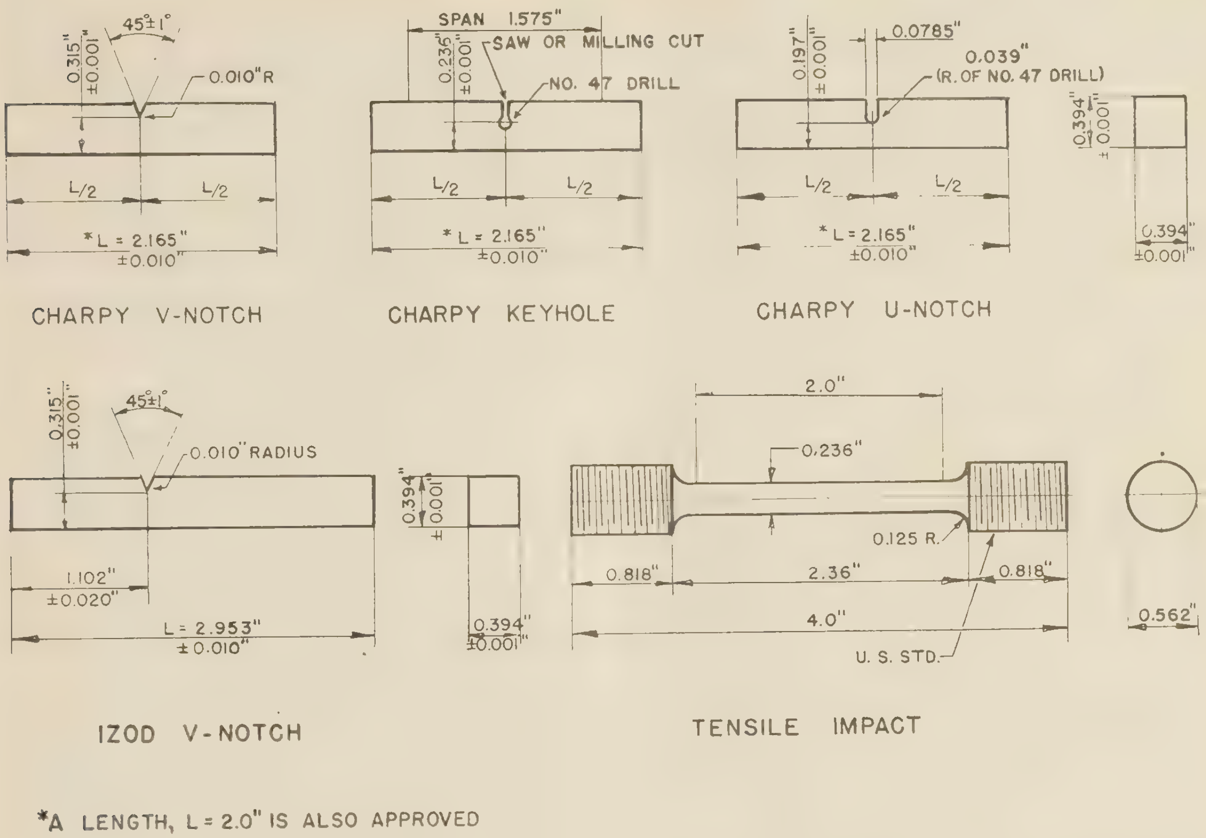


Fig. 1 Dimensions and Prescribed Tolerances[7] of Test Specimens

plate. The radius of each notch was checked by enlarging its profile 50 magnifications on a metallurgical microscope and comparing the curvature with the specified radius on the ground-glass viewing plate.

After machining, the specimens were heated at 160 F for about 16 hr to remove machining stresses.

All tests, except those that were made at the outside laboratory, were conducted on a Riehle universal impact-testing machine of 220 ft-lb capacity and of which the velocity of the striking head was about 18 fps. Inasmuch as the five different types of specimen under consideration were all tested on the same machine, this procedure obviously eliminated any variations that might be attributed to (*a*) differences among testing machines, i.e., whether of the Charpy or Izod type, or of different manufacture; (*b*) length of span; (*c*) velocity of the striking head; (*d*) shape and condition of the anvil, vise, and hammer.

The room-temperature tests were conducted at 70 to 75 F in all cases. The testing technique that was employed for the Charpy tests at 925 F was based upon similar tests that had been conducted a few years previously in which it was shown that, after soaking at 950 F for at least 1 hr, a specimen could be rapidly transferred with insulated tongs from the thermostatically controlled electric muffle furnace to the near-by testing machine in about 5 sec, in which time the temperature of the specimen would drop to about 925 F. Timing of the operation during the present tests indicated that there was only a very slight variation in the elapsed time, so the specimens were all at virtually the same temperature at the moment of impact, which is the significant factor in the technique.

The photomicrographs that were made for metallographic examination of each heat and condition of pipe material tested were taken of the V-notch specimen that broke at the value nearest the average for each respective group.

Results of Tests

The results of the notch-toughness tests conducted at room temperature on a typical heat of carbon-molybdenum pipe material in the "laboratory-normalized at 1725 F" condition using various types of test specimens are shown in Table 1.[8]

The first two groups of data afford an interesting comparison among 14 supposedly identical Charpy V-notch specimens tested at two different laboratories. The agreement between the two groups is significant both with respect to the average values obtained and to the degree of variation among individual specimens. The average of the 5 extra Charpy V-notch specimens which were tested also agreed with the averages of these two groups. However, because of the wide variation observed, it was thenceforth considered necessary to test at least 7 Charpy V-notch specimens in any group, provided sufficient material were available, in order that the test results might be representative of the sample being studied.

Selection of Types of Specimen for Further Study. As shown in Table 1, the Charpy V-notch tests of this typical heat showed relatively high notch-toughness values, and the results covered a wide range among individual specimens, from 55 to 87 ft-lb in one case. Having established the reliability of the data obtained

[7] "The accuracy that is usually maintained is 0.001 in. although if it will lower the cost of preparing test bars this can be increased to 0.003 in. on harder steels and to 0.005 in. on softer steels without endangering the reliability of the results" (7).

[8] See Fig. 2(*f*) for photomicrograph of this normalized heat.

using Charpy V-notch specimens of this heat, on the basis of tests at two different laboratories, it was felt that the magnitude and variation of these results were typical of the particular heat of steel being tested.

A comparison of the test results on Charpy keyhole specimens with those on Charpy U-notch specimens reveals not only that the average notch toughness of 7 specimens of each type was the same but that the degree of uniformity among individual specimens was practically the same. Both sets of results were considered to be very uniform. In view of this conclusion, it was decided that tests of as few as 3 specimens of either type would suffice in most cases to represent the sample under consideration. For both types of specimen, the average value was considerably lower than that obtained with the Charpy V-notch specimen.

The Izod V-notch test results were somewhat more consistent than the Charpy V-notch results in that there was a smaller spread in the values for individual specimens, and the average value was lower. On the other hand, the Izod V-notch results were not as uniform as the Charpy keyhole and U-notch results, and the average value was much higher.

In the case of the tensile-impact tests, which were introduced to indicate the behavior of the material in the absence of a notch, the unexpected variation that was encountered in the results as the tests proceeded prompted the testing of 12 rather than 7 specimens. At least a part of the observed inconsistency in results has been attributed to improper alignment of the specimen owing to the lack of spherical seats[9] in the design of the testing machine. This prevented the application of the load at the correct angle and resulted in some instances in the bending of the reduced section of the specimen at the moment of impact. This difficulty with the tensile-impact specimens tended to mask differences which might have been present in the material and, for this reason, this type of specimen was dropped from further consideration.

Therefore in keeping with the general practice of most pipe manufacturers and fabricators, who prefer the Charpy keyhole type of specimen and because of the growing popularity of the Charpy U-notch specimen as evidenced by inclusion of it in the A.S.M.E. Boiler Construction Code (5), it was decided to subject these two types of specimen to further investigation along with the Charpy V-notch type of specimen, which had been found to give relatively high average notch-toughness values, and thus provide a significant comparison with the lower values obtained with the other two Charpy specimens.

The results of the subsequent notched-bar tests, conducted at room temperature and at 925 F on various heats of steel in different conditions, are given in Table 2. The photomicrographs in Fig. 2 show the representative microstructure of each group of specimens. An analysis of these data on Charpy V-notch, keyhole-notch, and U-notch specimens discloses the following:

Variations With Type of Specimen. In most cases the average V-notch results were higher than both the keyhole and U-notch values, but in a few cases they were lower. Therefore it cannot be taken for granted that the results obtained on V-notch specimens of carbon-molybdenum pipe material will be consistently higher or consistently lower than keyhole- and U-notch results.

In general, the keyhole- and U-notch results were more uniform among individual specimens than the V-notch results obtained at any test temperature on any heat in any condition in which specimens were tested. However, attention is called to the fact that, contrary to expectations, considerable variation, roughly ±20 per cent, was encountered in one group of keyhole-type specimens and in one group of U-notch type specimens, both from heat 6347 in the "hot-rolled and drawn at 1200 F" condition when tested at room temperature, as shown in (*a*), Table 2. A comparison of the test results on Charpy keyhole specimens with those on U-notch specimens reveals that there was very little difference between the two types as regards the variation among individual specimens or the magnitude of the average results obtained at any test temperature on any heat in any condition.

Variations With Heat of Steel. In the tests at room temperature, wide differences among average notch-toughness values were obtained for V-notch specimens from various heats in a given condition, ranging in one case from an average value of 56 to a value of 105 ft-lb, as shown by a comparison of (*e*), (*f*), (*g*), and (*h*) in Table 2. However, when tested at 925 F these differences ceased to exist and complete agreement among average notch-toughness values for V-notch specimens was obtained for different heats in a given condition.

In the tests at room temperature, it was generally true that the results obtained with the keyhole and U-notch specimens were alike for various heats in a given condition. The only exception was the difference noted in the case of heats 6347 and 5085 in the "laboratory-normalized at 1650 F" condition, shown in (*c*) and (*d*), Table 2, for which no explanation is apparent. However, as in the case of the V-notch tests, complete agreement was obtained when the tests were conducted at 925 F.

Although the average notch-toughness values varied among heats, no heat consistently gave a better degree of uniformity among individual specimens than another, irrespective of the type of specimen used or the test temperature.

Variations With Condition of Steel. For all three types of Charpy specimen, the average notch-toughness values obtained at room temperature for a given heat of steel increased as a result of normalizing at 1650 or 1725 F. Tests at 925 F showed no such improvement as a result of normalizing in that the average values were about the same for a given type of specimen, for all conditions of all heats. Further, good uniformity was obtained in the tests at 925 F for all three types of Charpy specimen, for all conditions of all heats. The greatest variation in any group of specimens tested at 925 F was +9 per cent and —15 per cent for the keyhole-notch specimens of heat 5085, hot-rolled and drawn at 1200 F, as shown in (*b*), Table 2.

It should be noted that tests of material in the "shop-normalized at 1725 F" condition, using both Charpy V-notch and keyhole-notch specimens, gave lower values than those on material in the "laboratory-normalized at 1725 F" condition, as shown in (*e*), (*f*), (*i*), and (*j*), Table 2. The values for individual specimens in each group varied about as much for the shop-normalized material as they did for the laboratory-normalized material.

Discussion of Results

There seem to be almost limitless avenues of approach to the interpretation of the results of the present notched-bar tests but for the sake of conciseness, the discussion will be confined to the factors involved in the practical problems at hand, namely, "how to conduct notched-bar tests on carbon-molybdenum pipe material," and "what is the significance of the results?"

Considering for a moment the broad aspects of the determination of notch toughness of materials, there are numerous variables that affect the magnitude and the uniformity of the results. Further, the relationship between the magnitude of notch-toughness values and the service behavior of the materials always enters into the interpretation of test results and is controversial. By means of correspondence and a review of the literature on the subject, some information but few specific and pertinent data were obtained relative to carbon-molybdenum pipe material. The articles contained in the Bibliography[4] have been reviewed, among others, and found to be the most pertinent to this subject.

[9] Discussed by S. W. Lyon (8).

Information from these sources forms the basis for much of the following discussion.

Variables. The influence of many variables on notch-toughness value has been stressed in the literature. It is acknowledged that these variables, such as the striking velocity, the temperature of the test, the type and dimensions of the specimen, the heat of the steel, and the condition of the steel, all contribute to some confusion in attempting to evaluate notch-toughness results.

Data presented in the literature revealed that striking velocities were commonly within the range of 12 to 20 fps, and that caution in the interpretation of results must be exercised if the striking velocity falls outside of this range. However, as long as all tests under the present investigation were conducted on the same machine, with a striking velocity of about 18 fps, the effect of this variable on the comparability of test results does not enter into the problem.

Temperatures at which notch-toughness tests have been conducted by various investigators range from about —300 to 1000 F. Within that range, the effect of the temperature variable on both the magnitude and uniformity of notch-toughness values is of extreme importance.

Many investigators have commented to the effect that "Temperature has a prominent effect on notch toughness . . . The standard test is made at room temperature and ignores the circumstances that the same notch might produce brittle failure and a low impact value at a temperature but slightly lower" (3). Further, a recent theory regarding the importance of the test temperature on the transition point in notched-bar testing is expressed in the following remarks by Gillett (9): "The transition temperature, at which the type of fracture of a notched-bar specimen of a steel shifts from tough to brittle, is plainly a criterion through which differences not otherwise obvious might be made evident, so low-temperature notched-bar testing might be a useful tool in the constant search for the underlying causes of the 'inherent individuality' of a particular heat of steel, even though that heat might not be destined for low-temperature work." In so far as the authors have been able to ascertain, notched-bar tests of carbon-molybdenum pipe material have not been conducted at lower than room temperature; if such data are available they would be a welcome part of the discussion on this paper. Such an investigation might well be undertaken in order to further this study.

The results of the present investigation have shown that tests at 925 F fail to reveal significant differences among various heats and conditions of steel, in contrast with the marked differences that were observed when testing at room temperature, particularly with the V-notch type of specimen. It may be, as suggested by those that have conducted studies of materials at low temperatures, that even greater differentiation might be possible among heats and conditions of carbon-molybdenum pipe material by means of tests at relatively low temperatures.

Type of Specimen. In order to arrive at a reasonable answer to the question of the correct procedure that should be followed in notched-bar testing of carbon-molybdenum pipe material, frequent reference must be made to the relative merits of the sharp V notch as contrasted with the blunt keyhole or U notch. This phase of the present study found its counterpart in work of other investigators who have commented as follows:

" . . . the V-notch type of specimen is likely to be a more sensitive indicator of impact strength than the standard Charpy keyhole specimen" (10).

"The . . . (V) notch is used by some because it gives a larger numerical spread between notch tough and notch brittle materials than is obtained with the more generous radius of the Charpy (keyhole) notch" (7).

"Most commercial specifications are based on values secured from testing Charpy bars with keyhole notch . . . . . this type of notch is not so sensitive to differences in either composition or temperature as the Vee notch . . . . " (11).

The results obtained in the present investigation on carbon-molybdenum pipe material confirm these opinions of other investigators that better discrimination among various qualities of steel is afforded by the sharp V notch than by the blunt keyhole notch or U notch. It is likely that tests made using the keyhole notch or the U notch give a measure of the bend strength under conditions of sudden loading rather than of the true notch toughness of a material.

Heat of Steel. The following comments are probably the most significant that have been found pertaining to the variation that is attributable to differences among heats of steel. Armstrong and Gagnebin (11) stated, " . . . the present state of the art does not permit use of composition, grain size, and heat-treatment as criteria for resistance to impact, which indicates the advisability of subjecting each lot to test . . . "

In a discussion accompanying that paper, Rosenberg mentioned, " . . . impact tests secured on a single heat of steel are more or less peculiar to that individual heat, and will not necessarily be duplicated by another heat of the same kind of steel. Individual heats apparently have what may be termed 'inherent' resistance to impact within certain limits, and this is dependent upon factors not at present recognized . . . "

In the present investigation it is significant that although the chemical composition[10] was virtually the same for all the heats under consideration, and metallographic examination revealed only slight differences in microstructure among the heats in a given condition, the Charpy V-notch test results show marked differences among the average notch-toughness values for various heats; however, these differences are not revealed by either keyhole- or U-notch tests.

Condition of the Steel. The increase in the notch-toughness values as a result of normalizing at either 1650 or 1725 F, which was observed in the room-temperature tests using all three types of Charpy specimen, is consistent with the findings of Crocker and White (12) that normalizing of carbon-molybdenum pipe material improves the notch toughness.

The data shown in (*i*) and (*j*), Table 2, which were incorporated into this investigation from inspection test results on two heats of pipe material, shop-normalized at 1725 F, provide an interesting comparison between the practical aspects of notched-bar testing, as applied to material under consideration for installation, and the technical aspects of notched-bar testing in a laboratory investigation. It is readily apparent that, as judged by the results of Charpy V-notch and keyhole-notch tests on an adequate number of individual specimens, the laboratory-normalized material yielded higher notch-toughness values than the shop-normalized material. Examination of the microstructure reveals that these higher notch-toughness values were accompanied by relatively smaller grain size. These data appear to indicate that, on these two particular heats, the temperature might have been somewhat higher during the shop-normalizing than during the laboratory-normalizing treatment, or that the pipe had been held at normalizing temperature longer during shop-normalizing.

Summary of Discussion

The foregoing observations and discussion comprise the authors' interpretation of the test results that were obtained at

[10] Typical chemical analysis of carbon-molybdenum pipe material used in these tests: C = 0.18, Mo = 0.52, Si = 0.28, Mn = 0.45, S = 0.020, P = 0.017.

TABLE 2 CHARPY NOTCH TOUGHNESS OF TYPICAL HEATS OF COARSE-GRAINED CARBON-MOLYBDENUM PIPE MATERIAL

	Hot-Rolled and Drawn at 1200 F				Laboratory Normalized at 1650 F			
	(a) HEAT 6347 Notch toughness, ft-lb		(b) HEAT 5085 Notch toughness, ft-lb		(c) HEAT 6347 Notch toughness, ft-lb		(d) HEAT 5085 Notch toughness, ft-lb	
	Room temperature	925 F	Room temperature	925 F	Room temperature	925 F	Room temperature	925 F
V-NOTCH	20 21 Avg 23 21 26 Max 32 (+39%)* 32 22 Min 20 (-13%)* 20	56 60 Avg 55 55 54 Max 60 (+9%) 55 54 Min 54 (-2%) 54	50 39 Avg 42 63 34 Max 63 (+50%) 47 35 Min 26 (-38%) 26	47 48 Avg 50 52 49 Max 52 (+4%) 52 52 Min 47 (-6%) 51	46 33 Avg 48 28 28 Max 70 (+46%) 70 66 Min 28 (-42%) 62	57 57 Avg 58 59 58 Max 61 (+5%) 58 58 Min 57 (-2%) 61	94 81 Avg 71 77 70 Max 94 (+32%) 74 62 Min 41 (-42%) 41	55 54 Avg 57 55 58 Max 59 (+4%) 58 58 Min 54 (-5%) 59
KEYHOLE-NOTCH	26 36 Avg 36 39 33 Max 42 (+17%) 42 35 Min 26 (-28%) 38	36 33 Avg 35 35 35 Max 36 (+3%) 35 35 Min 33 (-6%) 34	39 36 Avg 35 38 33 Max 39 (+11%) 33 37 Min 32 (-9%) 32	32 36 Avg 33 35 34 Max 36 (+9%) 28 32 Min 28 (-15%) 31	46 Avg 45 41 Max 49 (+9%) 49 Min 41 (-9%)	39 Avg 39 39 Max 40 (+3%) 40 Min 39 (-0%)	37 Avg 37 39 Max 39 (+5%) 35 Min 35 (-5%)	37 Avg 37 37 Max 37 (+0%) 36 Min 36 (-3%)
U-NOTCH	30 34 Avg 37 38 41 Max 43 (+16%) 43 35 Min 30 (-19%) 40	33 34 Avg 34 34 35 Max 36 (+6%) 34 36 Min 33 (-3%) 35	29 32 Avg 32 33 32 Max 35 (+9%) 35 32 Min 29 (-9%) 32	30 32 Avg 32 31 31 Max 37 (+16%) 31 32 Min 30 (-6%) 37				

Note: * Variation of the maximum and minimum values from the average value in each group.

	Laboratory Normalized at 1725 F						Shop Normalized at 1725 F	
	(e) HEAT 6347 Notch toughness, ft-lb		(f) HEAT 5085 Notch toughness, ft-lb		(g) HEAT 3398 Notch toughness, ft-lb	(h) HEAT 24,440 Notch toughness, ft-lb	(i) HEAT 6347 Notch toughness, ft-lb	(j) HEAT 5085 Notch toughness, ft-lb
	Room temperature	925 F	Room temperature	925 F	Room temperature	Room temperature	Room temperature	Room temperature
V-NOTCH	54 Avg 63 66 Max 70 (+11%) 70 Min 54 (-14%)	58 58 Avg 58 57 59 Max 59 (+2%) 57 59 Min 57 (-2%) 58	82 83 Avg 78 79 86 Max 87 (+12%) 87 55 Min 55 (-30%) 77	57 55 Avg 55 55 56 Max 57 (+4%) 54 54 Min 54 (-2%) 55	62 51 Avg 56 48 63 Max 63 (+13%) 56 55 Min 48 (-14%) 54	102 106 Avg 105 109 108 Max 109 (+4%) 109 99 Min 99 (-6%) -	36 30 Avg 33 30 30 Max 36 (+9%) 32 33 Min 30 (-9%) 35	49 55 Avg 51 49 51 Max 58 (+14%) 58 49 Min 43 (-14%) 43
KEYHOLE-NOTCH	49 Avg 49 47 Max 51 (+4%) 51 Min 47 (-4%)	39 Avg 38 37 Max 39 (+3%) 37 Min 37 (-3%)	46 47 Avg 47 48 45 Max 49 (+4%) 49 48 Min 45 (-4%) 47	37 Avg 36 35 Max 37 (+3%) 35 Min 35 (-3%)	43 45 Avg 47 46 52 Max 52 (+11%) 48 47 Min 43 (-9%) 50	45 47 Avg 45 45 42 Max 47 (+4%) 46 46 Min 42 (-7%) 45	42 42 Avg 42 42 42 Max 42 (+0%) 42 39 Min 39 (-7%) 42	42 42 Avg 42 44 42 Max 44 (+5%) 44 40 Min 40 (-5%) 40
U-NOTCH		37 Avg 37 37 Max 37 (+0%) 36 Min 36 (-3%)	46 47 Avg 47 46 47 Max 48 (+2%) 46 47 Min 46 (-2%) 48	35 Avg 36 37 Max 37 (+3%) 35 Min 35 (-3%)	[illegible] 46 Avg 48 49 47 Max 50 (+4%) 49 46 Min 46 (-4%) 50	47 44 Avg 44 50 44 Max 50 (+14%) 44 41 Min 41 (-7%) 43		

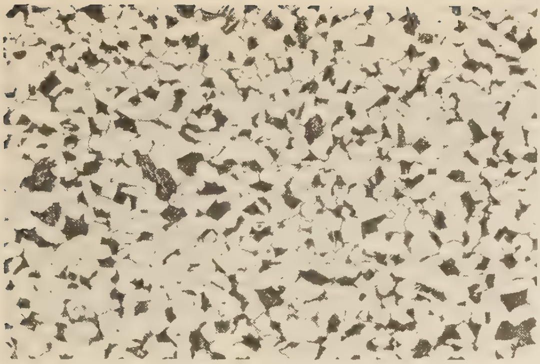
Hot-Rolled and Drawn at 1200 F

(a) Heat 6347; A.S.T.M. grain size 2–4

(b) Heat 5085; A.S.T.M. grain size 2–5

FIG. 2 TYPICAL MICROSTRUCTURE OF COARSE-GRAINED CARBON-MOLYBDENUM PIPE MATERIAL
(Etchant 3 per cent nital; ×100.)

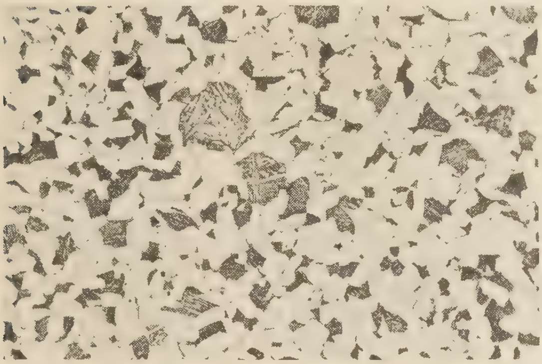
Laboratory-Normalized at 1650 F



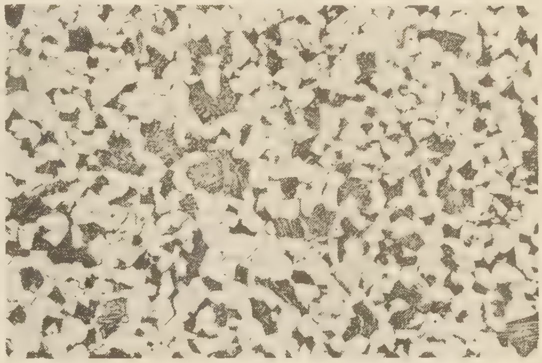
(*c*) Heat 6347; A.S.T.M. grain size 6

(*d*) Heat 5085; A.S.T.M. grain size 6

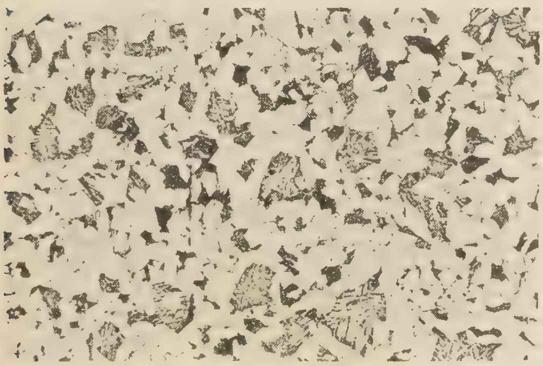
Laboratory-normalized at 1725 F



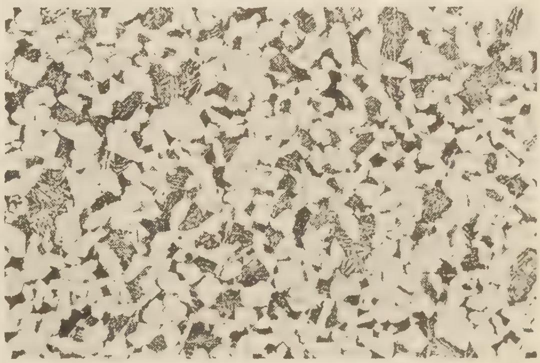
(*e*) Heat 6347; A.S.T.M. grain size 4–6



(*f*) Heat 5085; A.S.T.M. grain size 5–6



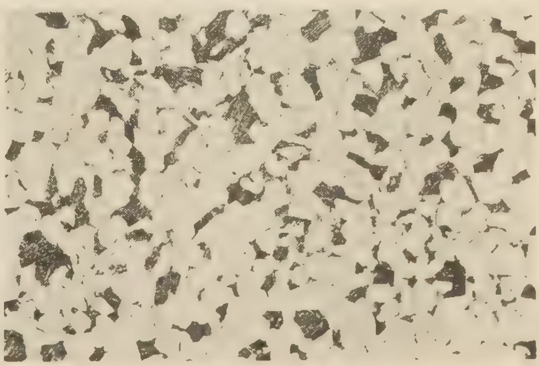
(*g*) Heat 3398; A.S.T.M. grain size 5–6



(*h*) Heat 24,440; A.S.T.M. grain size 5–6

Shop-normalized at 1725 F

(*i*) Heat 6347; A.S.T.M. grain size 2–6



(*j*) Heat 5085; A.S.T.M. grain size 5–6

FIG. 2 (*Continued*) TYPICAL MICROSTRUCTURE OF COARSE-GRAINED CARBON-MOLYBDENUM PIPE MATERIAL
(Etchant 3 per cent nital; ×100.)

room temperature and at 925 F, using various types of specimen from a number of heats of carbon-molybdenum pipe material in various conditions. This discussion has been augmented by frequent references that correlated with the findings in the present investigation.

In so far as the review of the literature is concerned, the present problem on carbon-molybdenum pipe material apparently has not been approached before from the viewpoint taken by the authors, namely, the determination of the influence on the magnitude and uniformity of results of the type of specimen, the heat of the steel, and the condition of the steel. However, this discussion would not be complete without some reference to other possible aspects of the problem that warrant further investigation. Of these, the influence of relatively low temperatures on notch-toughness values of carbon-molybdenum pipe material has already been mentioned. Also, the review of the literature reveals considerable interest in the use of double-width specimens as a source of additional information in debatable cases where tests of standard-width specimens indicate some peculiarity in the particular sample in question.

Riegel and Vaughn (1) mentioned "... when we find a heat of steel which is close to the minimum foot-pound value with our single-width specimen, we believe that a more critical examination of the steel is necessary by the use of the intensified notch effect of a specimen similar to that shown but of double the width. If, under this more severe test, the heat of steel gives double the foot-pound values of the single-width specimens, our suspicions on relative sensitivity to brittle cleavage are allayed. If the foot-pound values of the double-width specimens are less than double that of the single-width specimens, the heat is not accepted for the purpose intended."

Inasmuch as such instances occur from time to time in studies of carbon-molybdenum pipe material, it would be well to investigate the influence of double-width specimens on the behavior of various heats in order that some provision for possible tests of double-width specimens might be included in any inspection procedure involving notched-bar tests. Also, some search into the reasons for the inconsistent notch-toughness behavior of carbon-molybdenum pipe material would be in order. Data which might reveal possible relationships between notch-toughness characteristics of this material and microstructure, or steel-melting practice or heat-treating procedure would be a valuable contribution to the art.

Finally, there is the question of correlation of notch-toughness test data and expected behavior in service. While it has been possible in many instances, such as those cited by Riegel and Vaughn (1), for manufacturers to correlate notch-toughness requirements with field performance, and thereby gradually to improve their standards by better control of steel-melting and heat-treating practice, such a correlation, as it applies to carbon-molybdenum pipe material, is still vague. However, it is hoped that if, as those authors point out, wide differences in average notch-toughness values of successive mill heats have been minimized over a period of time, then a similar high standard of steel-melting and heat-treating practice for carbon-molybdenum pipe material might be maintained by continual attention to notch-toughness characteristics of the material.

Conclusions

To summarize the findings under this investigation, it can be concluded that:

1 Charpy V-notch specimens provide a more sensitive indication of differences in carbon-molybdenum pipe material than can be obtained through the use of either the Charpy keyhole- or U-notch specimens.

2 Tests made using the sharp V-notch give a more significant indication of notch toughness than either of the other two types of Charpy notch, inasmuch as the ability of a material to resist propagation of a crack under conditions of sudden loading is accurately revealed by the sharp notch.

3 In cases where the blunt type of notch is sufficiently sensitive for the intended purpose, Charpy keyhole- and U-notch specimens may be used interchangeably for testing carbon-molybdenum pipe material inasmuch as both types of specimen give substantially the same magnitude of average values for any group of specimens and about the same degree of uniformity among individual specimens in any group.

4 An adequate number of carefully selected specimens must be used to insure that the results are representative of the whole sample under consideration. This conclusion is based upon the degree of variation among individual specimens which was encountered with all three types of Charpy specimens. In general, more sharp-notched specimens should be used than blunt-notched specimens.

5 Tests at room temperature show significant differences among heats and conditions of steel, depending in degree upon the type of notch used, whereas tests at 925 F do not reveal such significant differences.

6 Results of notch-toughness tests and metallographic examination of material in the "shop-normalized at 1725 F" condition differed from those obtained on material in the "laboratory-normalized at 1725 F" condition.

Hence the ability of an investigator to determine and evaluate correctly the notch-toughness characteristics of various heats and conditions of carbon-molybdenum pipe material depends upon (*a*) selection of samples from each lot of steel in order to take into account the "inherent" resistance to notched-bar impact that is peculiar to each heat; (*b*) use of a type of notch that will reveal significant differences which may exist among heats and conditions of the steel; (*c*) careful preparation of an adequate number of test specimens to represent the whole sample under consideration; (*d*) proper selection of test conditions, and care in the conduct of the test, to include refinements in technique that will eliminate the testing variables commonly referred to in the literature.

In addition to the physical-strength tests commonly conducted on carbon-molybdenum pipe material at either room temperature or elevated temperatures, the notched-bar test should appeal to producers and consumers, not only because of the significance of the results obtained, but also from the standpoint of economy, simplicity, and expediency.

It can be ably argued by metallurgists that there are very definite influences present in every heat of steel which are directly related to the notch toughness of the material. Therefore it may not be entirely in the realm of speculation to assume that, when the significance of notch-bar testing of carbon-molybdenum pipe material is fully understood, it may be possible to select material of the best quality for high-temperature service on the basis of notched-bar tests.

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Discussion

G. F. Comstock.[11] Although we have not used the methods of testing on the type of steel to which this paper is restricted, our general experience with other kinds of steel has led us to agree with the authors in preferring the V notch to the keyhole notch for Charpy tests. Our reasons, however, are different from those of the authors, as we have found that results with the V notch seem to be more consistent so that fewer specimens are required for a reliable conclusion, and accuracy of machining is much more easily secured than with the keyhole notch.

Some strictly comparable results with the two types of notches, obtained in our laboratory from Charpy tests, at 30 deg below zero F, on standard specimens cut from practically the same locations in $^1/_2$-in. plates of commercially rolled low-alloy manganese-copper-phosphorus steel, might be cited to illustrate our experience. Two to four specimens were tested from each location, the results in foot-pounds being as reported in Table 3 of this discussion.

TABLE 3 CHARPY RESULTS ON TEST SPECIMENS

Sample designation	Location in plate	Charpy results, ft-lb at —30 F: V notch	Average	Keyhole notch	Average
4	Center	2.9 3.6	3.3	5.1 3.6 14.5	7.7
4	Near edge	7.2 7.2 5.1 8.0	6.9	26.0 30.4 26.8 28.2	27.8
5	Near edge	5.1 7.2 3.6 2.9	4.7	25.3 25.3 25.3 4.3	20.0
7	Center	2.9 3.6 2.9	3.1	3.6 2.9 3.6	3.4
7	Near edge	5.8 5.8 5.8 4.3	5.4	34.0 32.5 7.2	24.6
8	Near edge	2.9 2.2 2.9	2.7	13.7 18.8 2.2	11.6

The variation in results, leading to doubt as to the value of the averages, or as to the inconvenient necessity for considering all the individual results separately, seems much greater with the keyhole notches. Generally, except for tests at subnormal temperatures, we prefer to use the Izod method, since it is easier to secure concordant check results in that way, and more tests can be made with less machining expense from a limited amount of sample stock.

H. W. Hiemke.[12] This paper appears to be a contribution to the effort currently being made by power-plant engineers to gain a better understanding of toughness and tendencies for embrittlement of carbon-molybdenum steel at power-plant operating temperatures. From an examination of the data, it may be concluded that the materials examined by the authors demonstrated a relatively high degree of notch toughness at room temperature and at 925 F, in both the as-rolled and normalized conditions and for all conditions of test.

The authors quite properly conclude that the V-notch specimen is more sensitive than the U- or keyhole-type of notch. It is the opinion of the writer, however, that this extra sensitivity is obtained by sacrificing reproducibility of results. Also, in order to obtain reliable data with the V-notch type, it is necessary to make a larger number of tests than is the case with the keyhole- or U-type specimen. In this connection, it is desirable to quote the recommendations of the Advisory Committee of the War Metallurgy Committee regarding the advantages of using a Charpy keyhole-type specimen. The reasons for this recommendation were stated as follows:

1 Ease of reproducibility of an accurate radius of curvature of the notch.

2 Extraneous notches caused by drilling the hole are in the most desirable position with a drilled hole.

3 The area back of the notch in the standard Charpy bar is less than the simple beam Izod and a minimum amount of drag and bending occurs in tough specimens thus precluding fictitiously high values.

4 Permits correlation of the data with other results on low temperature impact as most engineering specifications require Charpy keyhole-notched specimens.

It is the opinion of the writer that the reasons advanced by the War Metallurgy Committee in favor of the keyhole-notch type far outweigh the added sensitivity found by the present authors in the use of the V-notch type.

J. L. Holmquist.[13] The authors suggest that the notched-bar impact test may be useful as a correlating device to help establish cause-and-effect relationship between obscure variables not evaluated or even detected by the common tests for chemical composition, tensile properties, and microstructure, and the performance of carbon-molybdenum pipe in high-temperature service.

It is believed to be a generally accepted fact that the notched-bar impact test is sensitive to obscure variables, which may be, for example, very small amounts of nitrogen, oxygen, or hydrogen, the presence or influence of which may not be detected by chemical analysis, tensile tests, or examination of microstructure.

It is to be noted that the notched-bar impact test does not identify such obscure variables, but merely identifies materials which may differ with respect to them although alike in all other respects. The notched-bar impact test can be used as a screening device to single out for the investigator suitable samples on which to pursue his investigations.

The "notch-sensitivity" of a metal depends upon a complex interworking of shear strength, cohesive strength, and the response of each to strain hardening. It is the writer's impression that a full understanding of these phenomena does not exist at present. But the concepts of shear and cohesive strength are more fundamental than notch sensitivity, as measured by the

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[12] Senior Materials Engineer, Bureau of Ships, Navy Department, Washington, D. C.

[13] Director of Research, Spang-Chalfant, Division of the National Supply Company, Ambridge, Pa.

notched-bar impact test. A study of the manner in which shear and cohesive strength is influenced by minor constituents or heat-treatment, employing techniques already developed, might be more illuminating than a study of their effect on the notched-bar impact test.

It does not follow that the notched-bar impact tests or shear-cohesive strength studies are the only or necessarily the best means for distinguishing between materials with respect to the presence or absence of obscure variables. Such properties as electrical resistivity, the several magnetic properties, thermoelectric emf, the thermal and stress coefficients of these properties, single electrode potentials, rate of attack by chemical reagents, etc., might provide as good or better screening devices.

It would have added considerable interest to the paper if the authors had given from their experience some examples of differences in high-temperature performance of carbon-molybdenum steels which were considered acceptable for such service by current manufacturing and testing procedures. Only one example has come to the writer's attention. In this case, graphitization occurred adjacent to a weld in a carbon-molybdenum tube. On the other side of the weld, a carbon-molybdenum steel forging was not graphitized, although exposed to the same service conditions for the same length of time.

T. McLean Jasper.[14] The writer wishes to suggest that consideration of the broad subject of testing and correlation with service must be made in connection with notch-impact testing. This has been attempted several times without any marked success. A symposium on "Notched-Bar Impact Test" was presented at the Zurich meeting of the International Association for the Testing of Materials in 1931. The report on such tests covering about 80 pages of carefully considered results on various types of test specimens and on a considerable range of steels together with discussion and summaries should not be overlooked. For the record it is desired to quote from the summary report of Dr. W. Rossenhain, president of Group A, which is attached to the symposium.

"From the reports submitted and the discussion which took place, certain definite conclusions could be gathered:

"1 With one or two dissentients, it was the opinion of those present that the notched-bar impact test is of great practical value as a reception test provided that it is carried out on a test piece of satisfactory type and that the results are interpreted within the limitations which the nature of the test imposes.

"2 It was generally agreed that the results of notched-bar impact tests cannot be used for the quantitative comparison of different types of materials but that they can be used as a test of the quality (satisfactory heat-treatment) of different batches or specimens of similar types of steel. In this sense, the test is to be regarded as little more than qualitative so that only the larger differences between results are important."

While the writer has copied only two of the seven paragraphs in the summary, it is believed that they are sufficient to inject an atmosphere of caution in the use of notched-bar impact values. The experience of the writer in endeavoring to correlate notched-bar impact results with service has been baffling. The test values presented in this paper have not remedied this defect. It is believed that the notched-bar test has been in existence for over 40 years. We are at this time without any "put-your-finger-on-the-spot" correlative values that we may apply in service with a satisfied feeling. It might seem that it would help machinery designers to avoid steels that were most sensitive to sharp corners in designs. Curiously enough notched fatigue specimens do not correlate with the single-blow notched-bar impact.

[14] Engineer, A. O. Smith Corporation, Milwaukee, Wis. Mem. A.S.M.E.

In considering carbon-molybdenum steel and the values presented, the writer would state that the test values are all good. The V notch has a higher average but a greater spread in test-value percentage. This may well be expected since the section area under the V is considerably larger than that under the U or keyhole and also because the V is much harder to machine with the same degree or percentage of precision than the other two.

In saying that the values are all good, it is meant that the values are all above 20 ft-lb and, since no strength and ductility values are available, there is no other comparison to draw. Some of our very expensive high-strength alloy steels cannot meet the values shown nor even a minimum of 20 ft-lb with the Charpy keyhole notch, yet they are successfully used in the services where shock is very severe and where high service stresses are essential for the successful operation of the part.

G. C. Riegel.[15] The authors have taken a rather bold premise concerning the correlation of notch sensitivity of steel and its performance in high-temperature pipe service. After having diligently read the paper, the writer was disappointed in not finding any service data to correlate with the metallurgical investigations of the material. It is presumed that these data will have to await some months or years of application in service.

There remains no question that the more severe the state of stress set up by a notch, the more readily steel can be made to exhibit a brittle behavior. In other words, either the severity of the notch or some other factor such as a change in the geometry of the specimen which increases the state of stress, the influence of temperature, especially a reduction from normal, will bring about a transition of the steel from a tough fracture to a brittle fracture, or failure.

It remains for the technician and the engineer to determine how far from some norm a material may depart, either in its characteristic inheritance, its pretreatment, or application in service (design considered) before one may say this heat of steel is satisfactory or that heat of steel is unsatisfactory.

The pursuit of this plan as proposed by the authors will do much to increase our knowledge and judgment in the manufacture and use of steel.

Authors' Closure

The authors appreciate Mr. Comstock's comments and the supplementary data on a different type of steel, which tend to emphasize the distinction the authors have drawn for C-Mo pipe material between the V notch, as contrasted with the the keyhole or U notch.

Mr. Hiemke has contributed to a broader evaluation of the present problem by bringing to the reader's attention the recommendations of the Advisory Committee of the War Metallurgy Committee. The four points raised are widely recognized and are usually followed in connection with notched-bar testing of most types of steel. With regard to the question of reproducibility of test results, the authors feel that, if a material is reasonably homogeneous or uniform with respect to the factors which influence notch toughness, there is ample evidence that the keyhole and U notch are likely to be more satisfactory than the V notch from the standpoint of reproducibility of test results. However, it seems that the less sensitive blunt notches might be likely to impose a certain "reproducibility of results" on materials which might not be inherently uniform or homogeneous. The findings in the present investigation tend to show this likelihood, inasmuch as it is inconceivable that the great differences in notch-toughness values with the V notches are fictitious or nonexistent if judged by the uniformity of the keyhole-notch results.

[15] Chief Metallurgist, Caterpillar Tractor Company, Peoria, Ill.

When dealing with a coarse-grained C-Mo steel such as is used for high-temperature pipe, and a wide divergence of values for individual specimens in a given group is obtained, investigators cannot afford to overlook the possibility that such variations might be attributable to the absence of uniformity of the factors which influence notch toughness rather than always to deduce that such differences can be accounted for only by "inaccuracies" in the V notch and "lack of reproducibility of results" when using the V notch. Further, the opinion is widely held that scratches or other stress raisers in notches are more likely to affect the magnitude of results when the values are low rather than high; in the present tests, relatively high values were obtained in all cases, so the effect of possible inaccuracies or stress raisers in the V notches should not reasonably be considered as accounting for the variations obtained among individual V-notch specimens.

It should not be inferred that the authors are attempting to depreciate the merits of the keyhole notch for most types of steel, but when it is recalled that the more recent heats of C-Mo pipe material are among the very few coarse-grained types of steel in service, a different aspect of notched-bar testing is introduced that seems to provide an exception to the usual concepts of this test. In this case, test results show that blunt notches do not accurately reveal true relationships among heats and conditions of this steel. The authors have therefore attempted to show by means of these data that the V notch should be used for this particular material if correct evaluation of various heats and conditions of this steel is desired.

This viewpoint is confirmed to some extent by Mr. Holmquist's remarks regarding the ability of notched-bar tests as a "screening device" to identify materials which differ from one another with respect to obscure variables, even though it is acknowledged that the test does not identify those obscure variables. Mr. Holmquist suggests that a study of cohesive strength and shear strength may reveal more explicit information regarding the effect of minor variables in composition or heat-treatment. But since these properties influence notch sensitivity, the notched-bar test might be regarded as a simpler, more practical way of revealing a composite picture of certain differences among batches of material.

Although not usually specified, the notched-bar test has been used by some consumers of C-Mo pipe material as a source of added information and, when low notch-toughness values have been encountered in any particular piece of pipe more comprehensive investigations have been initiated to insure that the pipe will be acceptable. In this way, notched-bar tests have served as a so-called "red flag" in installations of C-Mo pipe, and values for minimum notch toughness have been set, mentally at least, relatively high (about 15 or 20 ft-lb), as compared to those encountered in many low-temperature tests (about 3 to 6 ft-lb).

In commenting on Mr. Jasper's discussion, the test findings are in keeping with point 2 of the Zurich Meeting of the International Association for Testing Materials, on the subject of notched-bar tests, namely, "that they can be used as a test of the quality (satisfactory heat-treatment) of different batches or specimens of similar types of steel. In this sense, the test is to be regarded as little more than qualitative so that only the larger differences between results are important."

The present tests, using the V notch, have revealed large differences among heats and different heat-treated conditions of a single type of steel. Mr. Jasper's word of caution is particularly applicable when making comparisons among various types of steel. Inasmuch as the present tests were limited to a single type of steel and only the test temperature, the heat number, and the condition of this steel were varied, it seems that direct comparison of notched-bar results should be admissible in this case.

The authors wish to reiterate that no attempt was made in this study to correlate, at this time, test results and performance in service. Such a correlation might be possible only after many more years of service data become available. This also applies to Mr. Riegel's comment, for it should be remembered that coarse-grained carbon-molybdenum pipe is a relatively new material, compared with many other types of steel for which test results and service data are abundant.

Firing High-Pressure Furnaces

By E. G. PETERSON,[1] NEW YORK, N. Y.

Developments in the oil-refining industry, particularly in the catalytic-cracking process, have created a need for furnaces capable of carrying high pressures and sustaining high loadings. Direct-fired air heaters are in service which operate at pressures of 45 to 55 psi or higher, with furnace loadings of 500,000 to 2,500,000 Btu per hr per cu ft of volume. The design requirements of such heaters, their details of construction, and operating characteristics are discussed in this paper. The subject is of great importance since the possibility of wide application for this combustion system already exists. One of the most important fields under consideration is that of ship propulsion with power generated by gas turbines and direct-fired air heaters. "Package power" units made up of similar equipment may be used for power generation in small plants.

It has been fairly common practice for some years to fire boiler or process furnaces under moderate pressures, as, for instance, naval boilers, in which furnace pressures approximate 0.5 psig at high steaming rates. In this same application, heat release in the furnace will reach 250,000 to 300,000 Btu per hr per cu ft of volume. Limited space dictates these operating conditions. In certain industrial processes, furnace pressures have reached 5 psig but, as a rule, the firing is intermittent, occupying a relatively short period in the cycle.

Furnace pressures of 15 to 35 psig and heat liberation in the order of 900,000 Btu per hr per cu ft of furnace volume have been employed in connection with the Velox boiler, as well as in European installations of direct-fired air heaters and gas turbines for power generation. Such installations, however, have been very limited in their number and scope of application.

More recently, in this country, high furnace pressures and furnace loadings have become important in connection with catalytic refining processes in the oil industry. Many direct-fired air heaters are now in service with pressures of 45 to 55 psig or higher, and furnace loadings of 500,000 to 2,500,000 Btu per hr per cu ft of volume. Process requirement and operation of heat-recovery units (gas turbines) fix the system pressure, while economy in pressure-vessel design, rather than space limitation, is responsible for small furnace volume and consequent high rate of heat release per unit volume. In some cases the introduction of high-temperature combustion air (900 to 1000 F) during certain phases of the operation presents a problem in materials and operability. A large volume of relatively cool air, compactness of units consistent with reliability of service at high rates of combustion, and accessibility of parts requiring cleaning or repair demand the careful consideration of the designer of the burner and heater unit.

In the combustion system, Fig. 1, a large amount of compressed air is furnished by a high-efficiency axial compressor. During the process of catalytic cracking, heat is given off and the heated air returned to the inlet of the gas turbine in sufficient quantity to produce all the power required to drive the compressor. On starting up, however, none of this heat is available. Furthermore, the catalyst chambers, ducts, etc., must be brought to operating temperature. Two direct-fired air heaters are provided, one to supply heat to the process chambers, and the other to make up the balance of the energy to be transformed into mechanical power in the gas turbine for operating the compressor.

On starting up, the compressor is run at approximately one-quarter speed by means of a small motor or steam turbine. The air flows to heater No. 1, where its temperature is raised to 900 to 1000 F; the mixture of hot air and combustion gases passes through the catalytic chambers and is reheated in heater No. 2 before entering the gas turbine. The firing rate in heater No. 2 on starting is approximately 50 per cent greater than in No. 1 but, when equilibrium is reached, this burner may be shut off entirely. In some installations, however, heater No. 2 is kept in service continuously to provide additional power output from the gas turbine.

Direct-Fired Air-Heater Designs

The direct-fired air-heater unit, shown in Fig. 1, is one of the earlier designs having a capacity to heat 23,000 cfm of air from approximately 350 F to 900 F. The gas turbine is rated at 3650 hp, 2950 hp being used to compress the air; the additional 700-hp capacity is surplus power which may be used for the operation of the generator.

More recent installations utilize compressors of 40,000 cfm and 60,000 cfm capacity and 55 psi discharge pressure, with gas turbines of 7100 and 9500 hp, respectively, having correspondingly larger air heaters and burners.

Refractory-lined furnaces are largely used. All-metal furnaces have been employed with quite satisfactory results although original cost and maintenance are somewhat higher than the refractory type.

A direct-fired, air-heater burner-and-furnace unit of more recent design, operating at pressures of 3 to 25 psig and firing rates of 10,000,000 to 100,000,000 Btu per hr, is illustrated in Fig. 2. This unit was designed to heat 64,000 cfm of air from 50 to 1250 F outlet temperature, corresponding to a firing rate of 96,000,000 Btu per hr.

While there have been many improvements in the mechanical features of the burner and the heater since the first unit was designed, it is to be noted that the principles worked out for the original unit are employed in the most recent installations. The air stream is divided, and dampers are arranged to control the amount of combustion air to the burner, in order to maintain furnace temperatures sufficiently high to insure complete combustion of the fuel. The additional air to be heated, or "quench" air, passes through the annular space between the furnace and the outer shell and mixes with the hot gases after combustion of the fuel is completed. The burner is exactly the same in so far as mixture of fuels and air is concerned, but it is greatly improved mechanically over the original arrangement.

It is true that experimental units have been set up to operate at furnace loadings up to 10,000,000 Btu per cu ft per hr under conditions of furnace pressures and exit temperatures in excess of those encountered in the commercial applications mentioned, but the tendency in connection with these commercial units has been toward conservative loadings of 250,000 to 750,000 Btu per cu ft per hr, depending upon conditions of operation.

[1] Manager, Air Heater Department, Peabody Engineering Corporation.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

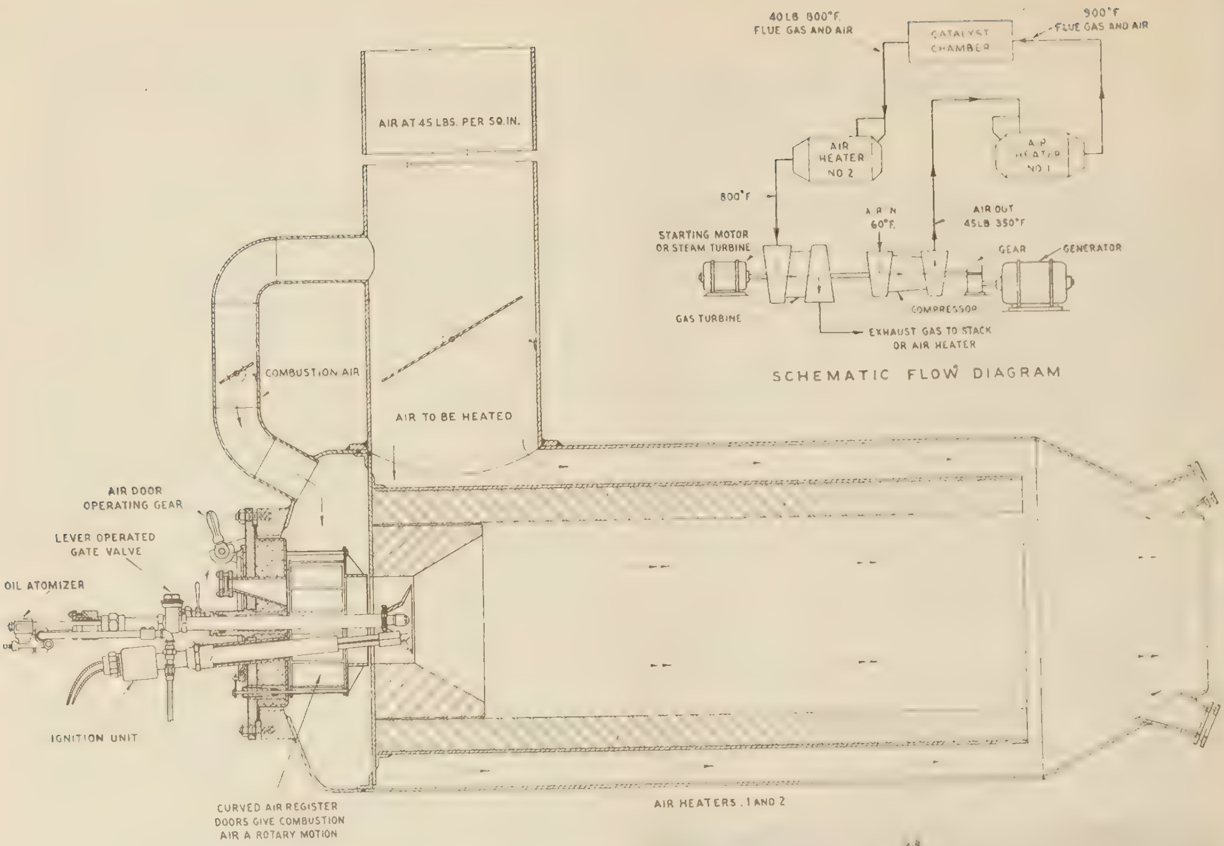

Fig. 1 Burner Arrangement

However, in one case, space limitations required a furnace loading of 2,500,000 Btu per cu ft per hr, with 45 psi furnace pressure and 950 F exit temperature. This unit has functioned satisfactorily for several years.

The original burner used in this service is shown in Fig. 3. Only the atomizer was removable with the unit under pressure. The electric igniter was fixed in position, with the result that it was necessary to remove the entire burner in order to replace a cracked porcelain or burned electrode tip. The peephole was rectangular in shape, flared to give a wide vision; however, in bolting the glass between the two rectangular plates, severe mechanical strain could be set up causing rapid failure of the glass when the additional temperature strains were imposed during operation of the unit. The mechanism for the operation of the register louvers consisted of a rotatable ring, mounted on the outside of the register front, carrying pins which engaged in the slots of the cams attached to the driving rods of the louver doors. This necessitated extending individual driving rods through the burner-front plate and fitting them with stuffing boxes to prevent leakage of air from the plenum chamber. Aside from details such as those just enumerated, the general construction features and operating principles of this burner have been retained in later designs.

Fig. 4 illustrates the design of burner now used, subject to modification to fit operating conditions of individual installations. The air doors or louvers, which give the combustion air a rotary motion, are linked together and are operated simultaneously by the master air door whose hinged rod extends through a stuffing box in the register-front plate and is fitted with operating handle and quadrant. The air-register-front plate is designed to meet pressure conditions in accordance with

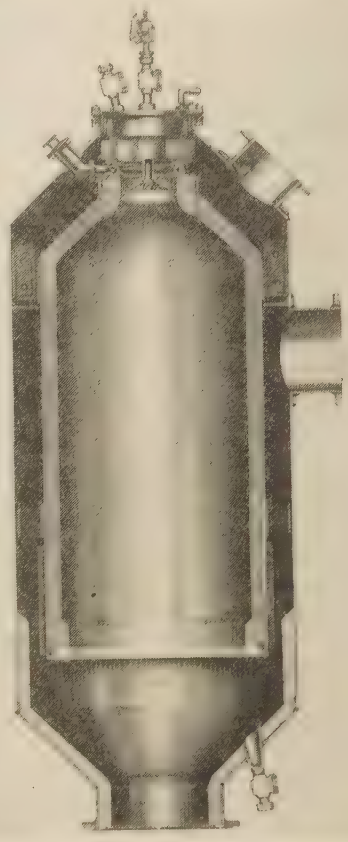

Fig. 2 Combined Gas-and-Oil-Fired Air-Heater Unit

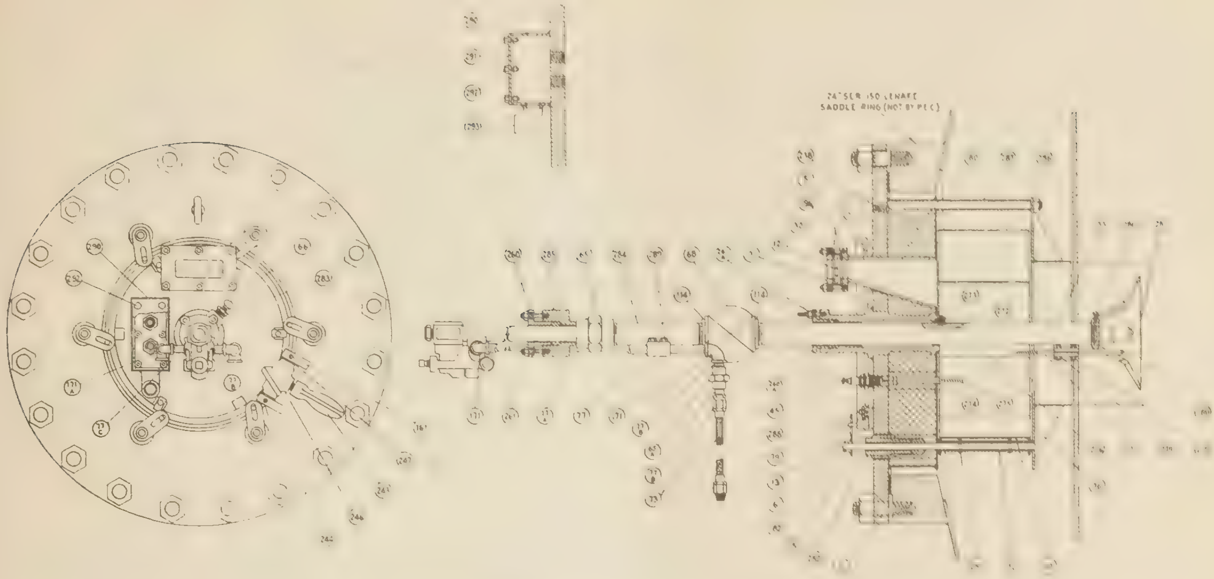

FIG. 3 ORIGINAL DESIGN OIL BURNER FOR HIGH-PRESSURE FIRING

the A.P.I.-A.S.M.E. Code for Unfired Pressure Vessels and is heavily insulated for protection against radiant heat or high-temperature combustion air. The louver doors are of heavy, flanged construction, designed to retain the proper shape under most severe air-temperature conditions. All parts of the burner in contact with highly preheated air are of stainless-steel material, and expansion provision is based upon the type of material and temperature of the combustion air.

The opening through the center of the front plate carries the oil-burner assembly consisting of the diffuser, diffuser pipe, quick-closing valve, union, atomizer stuffing box, atomizer yoke coupling with fittings, and the oil atomizer itself. The position of this assembly with reference to the burner throat may be adjusted with the unit under full pressure. Furthermore, the atomizer may be withdrawn from the assembly under the same condition. The type of atomizer may be either steam-atomizing, as shown, or wide-range mechanical-atomizing; both forms are operating successfully in this service.

Additional openings through the register front plate are provided for the peephole, electric or gas-electric igniter, flame-protection electrode, or other sensitive element, used in connection with the flame-failure shutoff controls. All of these openings are valved so they may be closed against furnace pressure; and, where the adjustment of position or withdrawal of the element is a requirement, stuffing boxes are fitted to prevent leakage of hot gases and possible injury to the operator. The valve in the peephole sleeve protects the glass from radiant heat; the glass is also covered by a wire gauze which assists in distributing heat strain and affords protection against flying glass if breakage from any cause should occur. In some installations, two igniters and flame protectors are used and openings in the register front plate must be provided for the additional unit.

In view of the fact the quick-closing valves are necessarily of steel, in order to conform to general refinery practice, the location of as many as six $2^1/_2$-in. valves on the burner front presents rather a problem when dealing with available commercial valves. In fact, there would not be room for one half that number, so it was found necessary for the company to design and manufacture valves for this special application. The same situation applied to the igniter, and to the small winch which is used for hauling it into position against furnace pressure.

DEVELOPMENT OF SUITABLE BURNERS

The gas burner consists of a tubular ring, correctly positioned to inject the gas fuel into the combustion-air stream, and protected from radiation. The port area is designed to meet operating conditions of capacity, fuel pressure, furnace pressure, fuel-heating value and density, etc. This type of gas burner is stable in operation over a wide range of capacities, pressures, and fuel characteristics.

The development of the burner for this type of service has followed a well-defined pattern. Certain essential standards were set up: (1) The burner must be efficient and capable of producing a flame to fit limited furnace dimensions; (2) it must be operable under all conditions of service; (3) it must provide for the safety of the operator and require a minimum of his time and attention.

High-combustion efficiency is a requirement which has to do not so much with economy in fuel consumption as with the ability to effect complete combustion of the fuel to avoid fouling of turbine blades or passing undesirable combustibles to the process chambers, and to confine the combustion to a relatively small volume without detriment to the furnace lining. This high efficiency is obtained through controlled turbulence resulting from correct shape and disposition of the fuel-distributing elements and baffle surfaces, and the ability to make adjustments from outside the burner with the furnace under pressure.

Under operability, we consider not only mechanical features as selection of materials of construction, allowance for expansion, one-hand control of louvers, etc., but also the fact that the burner must light off with the furnace under pressure and maintain ignition under varied conditions of load. The igniter can be adjusted for correct positioning with respect to the fuel element; adjustable baffle surfaces protect the flame from direct impingement of the air stream until combustion is well established.

Safety and ease in operation have been given careful consideration. The heavy steel register-front plate is adequately insulated against high-temperature combustion air and radiant

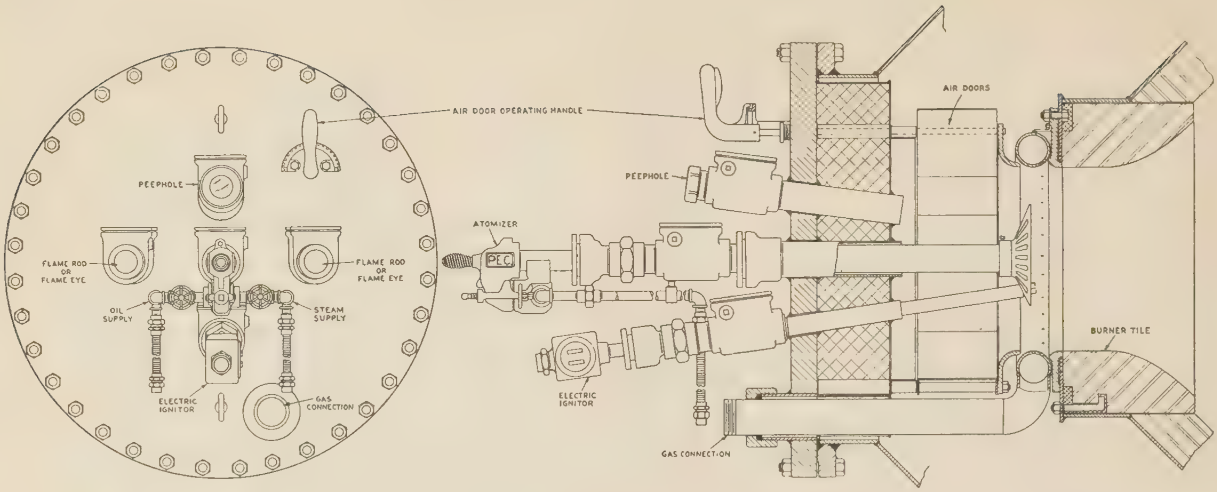

Fig. 4 Combined Gas and Oil Burner for High-Pressure Furnace Firing

heat; the heat-resisting peephole glass is covered with wire gauze and protected against radiant heat by a quick-closing valve; adjustable or removable elements are positively locked; and flame-protection devices are built into the burner to insure closing of fuel valves in case of flame failure from any cause and to prevent reignition, except as controlled by the operator.

Control and Protective Devices

The flame-protective system, Fig. 5, consists essentially of a relay which receives its impulse from a flame-contact rod or electrode when burning gas, or a photoelectric cell when burning oil, and in turn transmits this impulse to solenoid valves directly in the fuel lines or mounted on air-controlled, diaphragm-operated valves in the fuel line. The layout shown covers a dual system, consisting of two relays, two flame rods, two transformers, etc., which operate in parallel. The object of the dual system is to minimize the possibility of delays in starting or shutdowns due to short-circuit or ground in the various elements making up the control system.

In operation on oil fuel, the flame eye will be inserted in one of the flame-rod openings which are so arranged that this element can "see" the oil flame. The oil burner can be operated and adjusted by hand on starting up, by closing the key switch which by-passes the relays and opens the solenoid valves in the oil- and steam-supply lines. During the time the key switch is closed, an alarm sounds to warn the operator that the safety-control device is not in operation. Ignition of the oil flame is obtained on closing the push-button switch and, after the flame is adjusted for stable operation, the key is withdrawn from the key switch and the control takes over. During the time the burner is on oil, the gas-flame electrodes are withdrawn, at least out of the range of the oil flame, for their protection.

When starting up on gas fuel, the two flame electrodes are placed in position and the gas-supply valve in the main line is opened. Closing the push-button switch completes an electrical circuit through the transformers and opens a solenoid valve in the by-pass line to the gas-pilot burner. Ignition of the pilot burner takes place at a low rate which is increased when the pilot flame contacts the electrode. At the same time, the pilot flame is increased in size through the opening of the second valve in the pilot line, and the impulse from the relays is transmitted to the three-way magnetic valves controlling the diaphragm air supply to the main gas valve which opens and permits flow of gas to the main gas ring. When ignition is completed and the push button is released, the by-pass solenoid valve shuts, and both the center pilot burner and the main burner are under control of the relays.

Generally, the control systems are considerably simpler than that shown as they are designed for operation on only one fuel, either gas or oil. On certain gas-fired installations, where oil may be used simply as an emergency fuel, it is not considered essential to have the safety shutoff controls on both fuels. The complication in the system arises from the fact that no one as yet has devised a photoelectric device which will operate on gas fire without a dangerous time lag. It is therefore necessary to use the conductivity of the gas flame rather than radiation to a photoelectric cell which, if it were feasible, would necessitate only a change in filters to operate on either oil or gas flames.

Modifications to this control system are made to suit operating conditions, burner capacities, etc. However, the general features or principles apply to all installations. In any case, the most important consideration is the proper location of the elements which transmit the flame impulse to the relays.

Details of Modern Heater

From the beginning of our connection with this type of work, the essential form and dimensions of the direct-fired air heater have been considered from the viewpoint of the burner designer. To the best of our knowledge, the majority of, if not all, air heaters of this type, whether of our own or other manufacture, have followed the same basic principles developed for the first installation. On first approach to this problem, it was recognized from previous experience with furnace loadings up to 1,000,000 Btu per hr per cu ft of volume, that the high combustion rates involved required a high-efficiency turbulent-type burner, and that such a burner would give its best performance with a limited quantity of excess air through the burner proper. The first step, then, was to divide the air stream so that the quantity of combustion air could be properly controlled and the burner function within its efficient range. The remaining air was diverted to the annular space between the furnace and the outer shell, or allowed to enter the furnace for cooling purposes, outside of the combustion zone. Structurally, the heater shell is designed in accordance with the A.P.I.-A.S.M.E. Code for Unfired Pressure Vessels of materials and workmanship as specified by the Code, with the proper disposition of refractories and insulation to prevent metal parts attaining temperatures beyond those for which they are designed. Air passages must be so

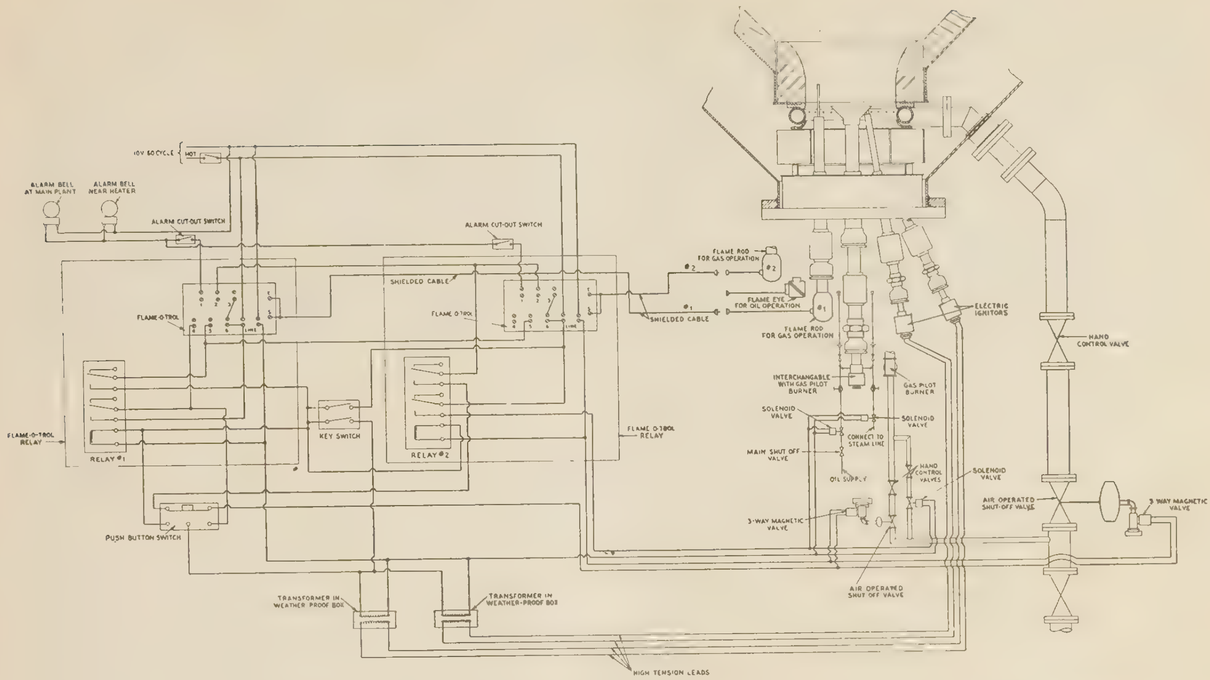

Fig. 5 Dual-Flame Protective System

designed as to keep down pressure loss, and dampers must be easily operated and their controls accessible to the operator, in order to insure that adjustments will be made when required for satisfactory operation.

The flow of heat through the furnace lining and through the outer-shell insulation must be considered in designing the air passages, selecting materials, and applying stiffening to the various members. When units are insulated externally, the thickness of such insulation must be proportioned to the type and thickness of the furnace lining, also to the velocity and quantity of air passing through the annular space between the furnace and outer shell, in order that metal temperatures in the furnace casing and shell are kept within reasonable limits. Convection transfer rates to the cooling air in the annular passage are relatively low, since velocities ordinarily do not exceed 20 fps. However, the area in contact with the air is quite large, as radiation of heat across the passage from the furnace casing to the shell makes the inner surface of the shell effective in the transfer of heat to the cooling air.

Toward the end of the annular passage some insulation should be provided on the inside surface of the shell as the cooling effect of the "quench" air diminishes. The air passage is thus restricted, velocity increased, and a somewhat higher heat-transfer rate obtained for a short distance.

In some cases, insulating firebrick, coated with a special cement to resist sloughing effect of high-velocity air, is used for protection of the shell; it is relatively inexpensive and easy to install. On other units, high-efficiency block-type or castable insulation is employed, requiring a cover of alloy steel for protection against erosion by the air and furnace gases. Where alloy steel is used in this manner, allowance must be made for its higher coefficient of expansion, also for its high temperature compared with the protected shell. Unless provision for this difference in expansion between shell and insulation cover plate is made, buckling of the light cover plate will take place.

When alloy steels are used in the construction of pressure parts of the vessel and are joined to carbon steel, the difference in expansion rate will set up severe strain unless the joint is properly located.

Many Applications Possible

While we have quite properly confined our discussion to applications in the catalytic refining field, in which we have had experience with upward of 100 installations, there are other applications outside of this field which are gaining in importance and which will demand more thought and consideration on the part of the engineering profession when the present demand on our energies for production of indispensable necessities, such as 100-octane gasoline, begins to abate somewhat. Of these other applications that of ship propulsion by power generated by gas turbines and direct-fired air heaters probably holds the greatest appeal, although we should not lose sight of the fact that there are many problems in other lines which are opening up to consideration. The possibility of applications of direct-fired air heaters to driers, incinerators, etc., and of the development of "package units" for power generation in small plants, requiring a minimum of technical skill for installation, operation, and maintenance, appears to be extensive.

The author realizes that only the high spots of this comparatively new subject have been covered and that he has dealt more in generalities than in specific data. However, the object of this paper has been to acquaint the prospective user of this type of equipment with what he should expect from the designer of the unit rather than to give instruction in the art; also to promote discussion on present applications, whether commercial or experimental, and to induce interest in the future broadening of the uses to which the equipment may be put.

Discussion

A. M. G. Moody.[2] It would be interesting if the author

[2] Chief Research Engineer, DeLaval Steam Turbine Co., Trenton, N. J. Mem. A.S.M.E.

would add to his already valuable paper some information on the degree to which he has been able to get uniform flow across the exit from the combustion chamber. If the products of combustion are to be used in a turbine, it is imperative that the flow be at a fairly uniform temperature, having no local hot or cold spots. If this condition does not prevail, a smoothing-out section would have to be added, the weight and volume of which would be directly chargeable to the combustion chamber.

Author's Closure

It is recognized that in the refinery installations the volume of duct work connecting the air heater to the vessel in which the heated air is to be used may be many times the volume of the heater itself. Therefore temperature distribution at the heater outlet is not considered of primary importance and no provisions have been made in these installations for obtaining data on this condition.

Where the air heater discharges directly into the inlet of a combustion-gas turbine the distribution of temperature at the heater outlet must be such that local departures from the mean do not exceed 25 F, plus or minus. This has been accomplished satisfactorily by application of positioning control to the individual louvers of the burner air register. Control of the flow of combustion air can be obtained within the relatively small space of the burner air register where the velocity is at a maximum. This construction eliminates the necessity of providing adjustable dampers, vanes, etc., in the larger and less accessible quench air passages toward the hot end of the chamber.

"Temp-turb" Temperature-Control System

By J. R. CAMPBELL,[1] ONTARIO, CALIF.

The "Temp-turb" system of temperature control, described in this paper, consists basically of a means for utilizing the kinetic energy in a flowing fluid in a manner to control the temperature of the fluid. A turbine wheel, the blades of which change their position with temperature changes, has all, or a sample of the flowing fluid, passed through it. Direction of rotation, rate of rotation, and torque output are a function of temperature when fluid-flow rate is constant. Power available from the turbine wheel actuates the valves which determine the fluid temperature.

The amount of work available from any basic temperature-sensitive element is relatively small and must be substantially amplified by some form of external energy before the response to temperature change can be made to register effectively on the control means. In aircraft, the control means might be an exit flap, such as on an intercooler; a butterfly valve, such as in a heating and ventilating duct; a mixing valve, such as in a heating and ventilating system; shutters or flaps for oil coolers; dump valve for thermal deicing system, etc.

The "Temp-turb" system of temperature control uses the kinetic energy of the flowing fluid being controlled to amplify the work done by the basic temperature-sensitive element. The kinetic energy available in fluid-flow systems in aircraft is generally substantial, and enough energy can be harnessed to operate control means without introducing a harmful pressure drop.

Fig. 1 shows a Temp-turb control unit, as used in the Douglas A-20 for controlling the temperature of air leaving a combustion heater. This air is for cabin heating, windshield defrosting, and turret heating. The Temp-turbs are set to maintain a control point of 200 F ± 15 deg and actually control to within ± 5 deg. As shown in Fig. 2, the control unit is furnished as an integral part of the modulating valve. Operation of the assembly is to allow more air to flow over the heater as temperature tends to increase, and to reduce the air flow as temperature tends to decrease. Heaters in this system are ram-operated.

Operation of Control Unit

Fig. 3 indicates the operation of the rotor or turbine wheel for the type of Temp-turb used on the Douglas A-20. Turbine blades are made of high-activity bimetal 0.005 in. thick and are calibrated to be straight at the control point. Any deviation from the control-point temperature results in a curvature of the turbine blade (the direction of curvature depending upon the direction of the temperature deviation). As the device is a radial-flow turbine, the curvature of the blade imparts a torque to the rotor which is transmitted through a gear reduction to the modulating valve.

After the turbine blades are formed and stabilized (to maintain calibration under all conditions to which control might be subjected), it is not feasible to vary the control point of the unit by varying the form of the blades. Therefore, the angle at which the fluid enters the rotor is varied. Referring to Fig. 1, the inlet guide vanes are mounted as cantilevers at the edge of the inlet nozzle, and their free ends engage a shift ring by means of small pins. Moving the shift ring in either direction from neutral varies the direction of fluid flow over the rotor to either side of pure radial flow. The amount of deviation from pure radial flow is determined by the amount the shift ring is moved. When the flow through the rotor is deviated from radial, the rotor assumes a new temperature-control point. This adjustment allows the control point to be varied about ± 40 F from the normal.

The assembly, shown in Fig. 2, operates on a sample of air which varies from about 5 to 10 per cent of rated flow for a full-open valve. Fig. 4 is a partial cutaway view of the assembly in Fig. 2 and shows the inlet scoop and exhaust outlet from the turbine back to the duct.

Numerous other applications of the Temp-turb principle of temperature control are in the developmental stage; however, the two examples mentioned should provide an adequate picture for an understanding of the system in general.

The work available in any Temp-turb unit is a function of the following:

1 Deviation from the control point.

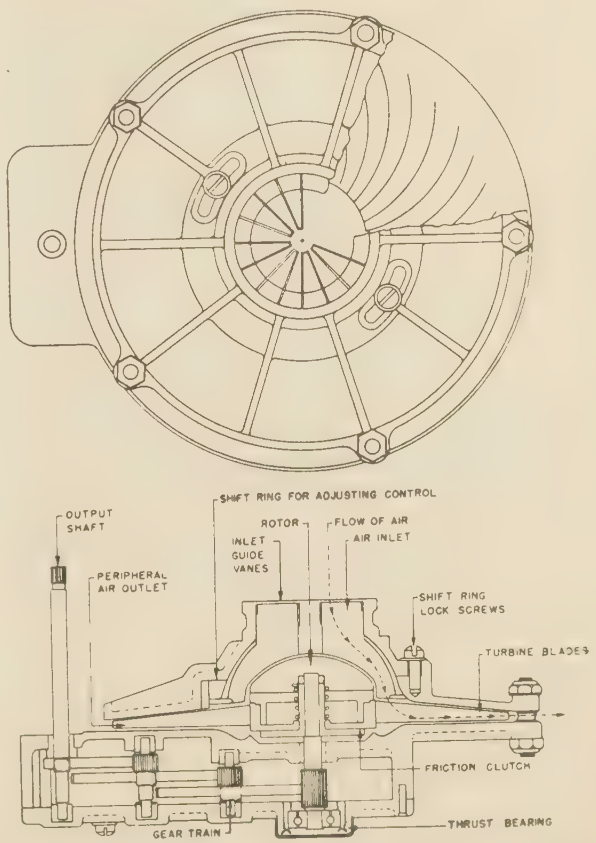

Fig. 1 Type of "Temp-turb" Unit Used on a Douglas A-20 for Controlling Temperature of Air Leaving a Combustion Heater

[1] General Electric Company. Mem. A.S.M.E.

Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

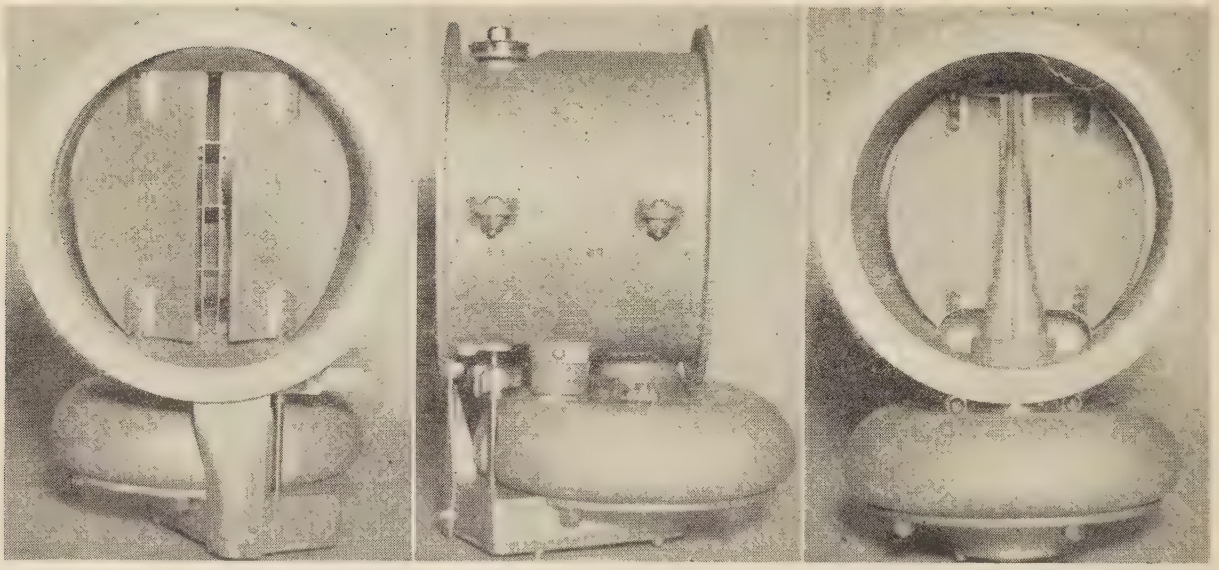

Fig. 2 Control Unit Is an Integral Part of the Modulating Valve

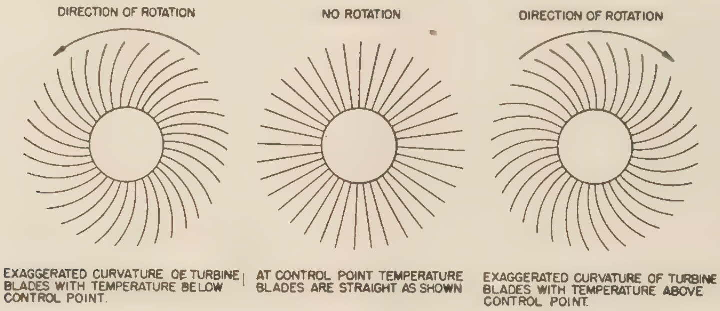

Fig. 3 Operation of Turbine Wheel on Typical "Temp-turb" Control Unit

2 Mass flow of fluid, which for air has to be corrected for

(a) Temperature } Density
(b) Altitude }

The temperature sensitivity is a function of the following:

1 Massiveness of the basic temperature-sensitive elements.
2 Velocity of the fluid being controlled.
3 Gear ratio between rotor and control valving.
4 Thermal lag of the system.

When designing a Temp-turb unit for a specific application, it is first necessary to determine the following items:

1 Control point.

2 Degree of accuracy to which temperature must be controlled.

3 Maximum work required to operate valving or control means. This may be broken down into the following: (a) Maximum torque required. (b) Rate at which control means must be moved when operating under maximum-torque conditions.

4 Mass-flow rate of the fluid in system.

5 Allowable pressure drop (drop due to Temp-turb).

6 If system involves flow modulation, what is the minimum flow allowable. In this system, it is always necessary to have a sample of the fluid flowing over the Temp-turb rotor at all times. It is not practical to design for a minimum flow of less than the amount needed for a sample.

7 Allowable space.

8 Allowable weight.

9 Cross-sectional area of ducting or piping in the zone where Temp-turb is to be installed.

10 Whether or not the control point has to be adjustable: (a) As a service operation; (b) as an "in-use" operation.

Principal Characteristics of System

General characteristics of the Temp-turb system of temperature control are as follows:

1 Inherently nonhunting: The deflection of the blades, whether made of temperature-sensitive material or actuated by a temperature-sensitive element, is proportional to temperature changes; therefore, the output of the rotor is proportional to temperature changes. A large temperature deviation produces a large corrective force and a large rate of correction. A small deviation produces a small corrective force and a small rate of correction. As temperature error is being corrected, the correcting force and speed (produced by rotor) are reduced in proportion to the error until the control point is reached, at which time corrective force and speed of correction become zero.

Numerous tests under transient conditions have proved that

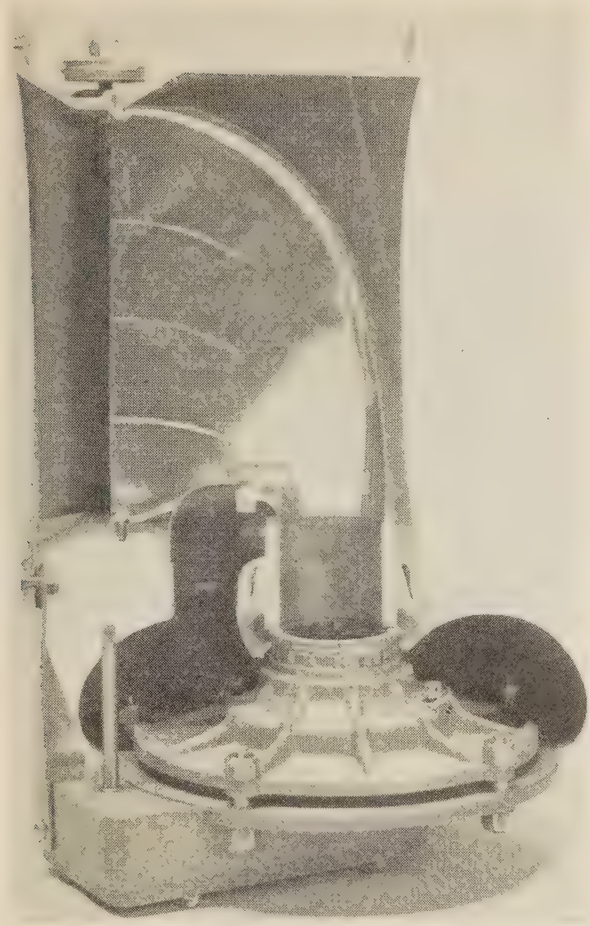

Fig. 4 Partial Cutaway View of Temp-turb Assembly of Fig. 2

the system is inherently stable. Of course, the thermal balance of the system must be within reasonable limits, and the gear ratio between rotor and control means also must be within reasonable limits.

2 Falling torque characteristics with velocity pressure increases above a certain value: The turbine blades, being operated by resilient thermoelements, are spring-loaded into proper position. This feature allows the blades to be blown into a neutral position under conditions where excessive velocity pressure might result in damaging loads or wear on the mechanism. This factor is of value on high-speed aircraft where ram pressures at times become quite high.

3 Adequate sampling of fluid being controlled: Temperature-sensitive elements have large area exposed to flowing fluid, which is particularly helpful under conditions wherein there is "temperature layering" in the duct. Also, the system is adaptable to special types of sampling scoops.

4 Low thermal lag: Bimetal temperature-sensitive elements are thin and subjected to relatively high-velocity fluid flow, resulting in almost instantaneous response to temperature changes.

5 Not affected by vibration: The torque produced by a vibrating turbine blade is the same as for a nonvibrating blade.

6 Output related to demand on unit: As flow rates increase, causing greater loads on control means, the rotor output increases accordingly.

7 The type of Temp-turb assembly as shown in Fig. 2 has the characteristic of not being affected by radiant heat from a glowing combustion chamber on a combustion heater. The sampling scoop and the location of the Temp-turb control unit make the device respond to air temperature only. This condition is of importance when the control is mounted immediately downstream from the combustion heater.

Operating Experience

Experience to date indicates no unusual difficulties from a manufacturing standpoint. Any remarks on cost must be predicated on the somewhat limited manufacturing and commercial experience encountered to date. Based upon this experience, it may be said that Temp-turb type controls cost no more than competitive devices.

Concerning service life, Temp-turb controls on test have demonstrated that they have a trouble-free life of greater duration than other forms of controls used for the same operations. It is likely that this is due largely to freedom from complexity. Two units on life test actually worked better after 3000 hr than at the start of the test, due to "wearing-in" of bearings.

The temperature range, over which the Temp-turb principle could be applied, is approximately from —60 F to +600 F. This is on the basis of response and durability of the bimetal temperature-sensitive elements. Greater problems are offered in bearing design at extreme temperatures, and at this time the bearings and their lubrication actually determine the applicable range. On this latter basis, the range is about —60 F to + 350 F.

Fatigue Studies on Urea Assembly Adhesives

By A. G. H. DIETZ[1] and HENRY GRINSFELDER[2]

Assembly adhesives made of "cold-setting," urea-formaldehyde, synthetic resins, are employed in the fabrication of wooden aircraft such as gliders and motor-driven airplanes. Adhesives so used are frequently subjected to vibrating loads which induce alternating stresses in the glue; of these shear, in the plane of the glue, and direct tension, perpendicular to the glue line, are the two which are most likely to bring about failure. It is well known that most materials fail under alternating stresses smaller in magnitude than the static stresses required to cause failure, and synthetic-resin assembly adhesives are no exception. Little has been known quantitatively, however, respecting the behavior of such adhesives under alternating stresses.

In this paper are presented the results of researches into the failures to be expected in glue joints of this kind, particularly the nature of the failure, cycles of stress to failure at various amplitudes and stresses, and the probable "fatigue limit" of such an assembly. Tests were carried out at room temperatures on an assembly adhesive representative of those employed in aircraft fabrication.

Materials and Test Specimens

The research on "cold-setting," urea-formaldehyde, synthetic resins, as representative of assembly adhesives used in wood aircraft fabrication, was carried out as follows:

Panels. Panels of overlapping $^1/_8$-in. rotary-cut maple veneers were made up, as shown in Figs. 1 and 2. The two outer members *a* overlapped and were glued to the two inner members *b* over a distance approximately $1^1/_2$ in. The line *b-b* was glued throughout its length, so that the center member behaved as a single piece $^1/_4$ in. thick. Maple was chosen because it is standard shear-block material, is relatively difficult to bond, and is moderately strong.

The panels were prepared according to the formulation: Urea-resin powder, 70 parts; water, 30 parts.

Spreading was by hand brushing and was on both surfaces being bonded. The assembly time ranged from 5 to 25 min, and in all cases the assemblies were closed immediately after spreading and kept closed until the pressure was applied. Bonding was for 16 hr at a temperature of 75 F, and the bonding pressure was 200 psi applied by means of a Black "compressometer."

Two sets of panels were made because difficulties encountered in obtaining satisfactory results with the first set necessitated the fabrication of more material. Panels were numbered as follows:

First	Second
A2	AA1
A3	AA2
A4	AA3
A5	AA4
A6	AA5
A7	AA6

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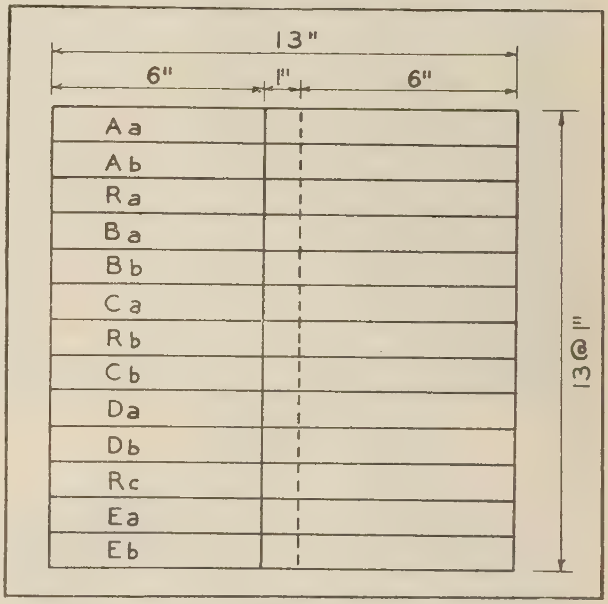

Fig. 1

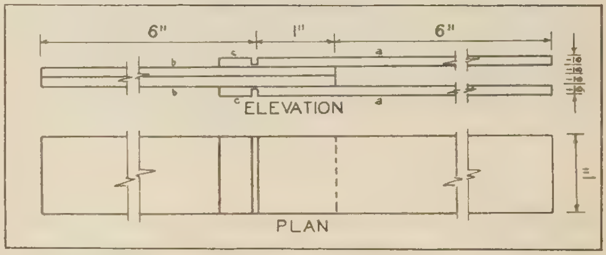

Fig. 2

Specimens. Specimens 1 in. wide were cut from the panels, as shown in Figs. 1 and 2. The overlap was cut to 1 in., as shown in Fig. 2, to provide 2 sq in. of glue line, although a few specimens were cut to $^3/_4$ in. for experimental purposes. In some instances the remaining piece *c* was removed, but in the majority of the tests it was found better to leave it in place.

At the ends of glue lines *a-b*, Fig. 2, beads of glue had squeezed out into the internal corners. In some instances the bead was removed and in some it was notched with a saw, but it was found by test that the glue actually cracked after a few alternations of stress, so that is was unnecessary to pay any particular attention to the fillet caused by glue bead.

Specimens were marked as shown in Fig. 1. Those marked Ra, Rb, and Rc were intended for "modulus-of-rupture" tests, that is, to determine the breaking strength of the assemblages. Specimens Aa, Ba, . . . were intended for vibration, specimens Ab, Bb, . . . for static shear. It was not possible to adhere to this schedule rigidly for the first set of panels because of experimental difficulties to be described, but the specimens were in general so employed in the second set of panels. Moisture content of the material was approximately 9 per cent during the tests.

TEST METHODS

Vibration

Machine. Because of the number of specimens involved and the length of time required for some of the tests, it was necessary to employ a machine which could handle several specimens at one time. The machine is shown in Fig. 3. In this apparatus the

Fig. 3 Vibration Machine With Two Alternating Stress Specimens in Place; End Supports Are Semiclamped

specimens, in groups of two, are held between upright pairs of cold-rolled steel plates which in turn are bolted to a heavy horizontal steel bar fastened to the steel base plate supporting the entire apparatus. The centers of the specimens are forced back and forth by a pusher attached to a driving rod pinned to a connecting rod in turn mounted on an eccentric which forms part of a heavy steel driving wheel. The eccentric can be adjusted to give a throw ranging from zero to $1^1/_2$ in. The driving wheel is mounted in a heavy bronze bearing and is belt-driven by a 1-hp motor. The machine can be operated at various speeds, but in these tests was limited to 1100 to 1300 rpm.

End Supports. Three different methods of supporting the ends of specimens were tried. The first, rigid clamping, as shown in Fig. 4, resulted in fairly frequent fatigue failures in the wood at

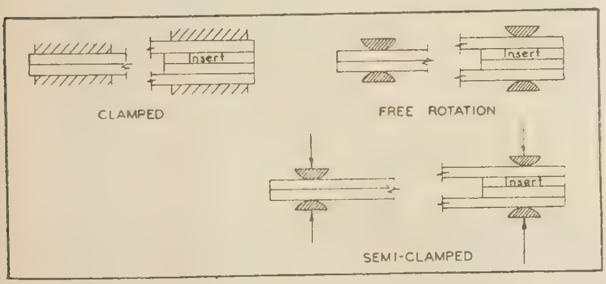

Fig. 4

the supports. Free rotation of the ends, the second method of support, required large deflections at the centers to yield appreciable stresses and caused numerous wood fatigue failures at the center. The end support finally adopted, and the one which yielded generally satisfactory results, consisted of rounded supports, between which the specimens were firmly held by bolts exerting pressure as shown. This allowed a little rotation at the ends, resulting in a semiclamped condition.

Pusher. Four different positions of the pusher were tried, as shown in Fig. 5. Position 2 was first employed, but resulted in fatigue failures in the wood rather than glue-line failures because even the slight compression applied by the pusher to the glue line prevented tensile stresses from building up to the point where

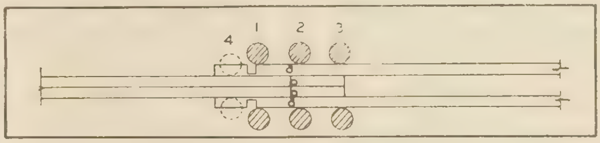

Fig. 5

failure could occur. Such a condition would not generally be found in actual practice. Position 3 was somewhat better, but position 4 resulted in rather quick wood failures. Position 1, finally adopted for the pusher, proved best from all standpoints. It allowed tensile stresses to develop freely. Fairly high percentages of glue-line failures resulted when the combination of pusher in this position was employed, and when saw cuts were held approximately $^1/_{32}$ in. short of glue line *a-b* to avoid local concentrations of stresses which would cause the wood to fail in fatigue. The pusher was not pressed firmly against the wood but was allowed to clear by a few thousandths of an inch or, at most, to come into light contact.

Tests were carried to failure or to 5,000,000 cycles. Two, four, or six specimens were tested at one time, depending upon the amplitude.

Static Shear

Static-shear control specimens were placed in the tension grips of a 10,000-lb Riehle beam-type universal testing machine and pulled apart, as shown diagrammatically in Fig. 6. Under

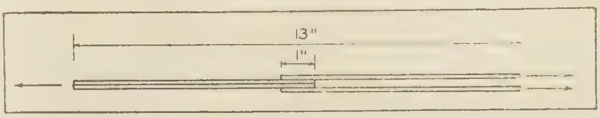

Fig. 6

this load, shear stresses are set up in the glue lines *a-b*, accompanied by some tension perpendicular to the glue line occasioned by the eccentricities in the joints.

Static Bending

Modulus of Rupture. Selected specimens, generally those marked Ra, Rb, Rc, were loaded statically to rupture to determine the breaking strengths and deflections so that the amplitudes to employ in the vibration tests could be decided. As shown diagrammatically in Fig. 7, specimens were supported at the ends

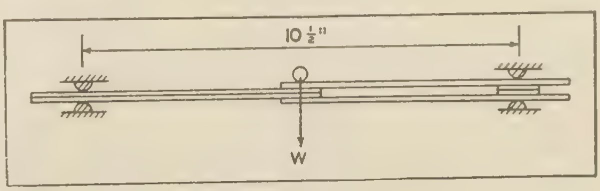

Fig. 7

in the same manner as the vibrated specimens and were loaded with dead weights at the center in the same manner as the vibrated specimens, except that loads were carried to failure. Loads and deflections to failure were both recorded.

Calibration. Each vibration specimen was calibrated by load-

ing in the same manner as the modulus-of-rupture specimens, except that loads were carried only to deflections corresponding to the amplitudes through which the specimens were to be vibrated.

In order to determine the end restraining moments at the supports induced by clamping and semiclamping, a number of specimens were tested twice, once as freely supported simple beams with no end restraint, and again as clamped or semiclamped specimens.

If d_f = deflection at center of a freely supported beam loaded at the center

d_c = deflection at center of a beam completely restrained at its ends and loaded at the center

W = concentrated load at center

L = length of beam

E = modulus of elasticity

I = moment of inertia

M_f = bending moment at center of a freely supported beam loaded at the center

M_c = bending moment at center of a beam completely restrained at its ends and loaded at the center

M_r = restraining moment at ends of a completely restrained beam

$$d_f = \frac{1}{48}\cdot\frac{WL^3}{EI} \qquad [1]$$

$$d_c = \frac{1}{192}\cdot\frac{WL}{EI} = \frac{1}{4}d_f \qquad [2]$$

$$M_f = \frac{WL}{4} \qquad [3]$$

$$M_c = \frac{WL}{8} \qquad [4]$$

$$M_r = -\frac{WL}{8} \qquad [5]$$

For semiclamped specimens, the deflections are intermediate between Equations [1] and [2], and the corresponding center moments are intermediate between Equations [3] and [4]. By determining the deflections for Equation [1] and for the semiclamped condition, the center moments for the semiclamped condition can also be determined, and from these the maximum tensile stresses perpendicular to the glue line at the overlap. Shear stresses can be determined from the load, whose magnitude is known as a function of the amplitude (or deflection).

Fig. 8 shows the approximate relationship between the external

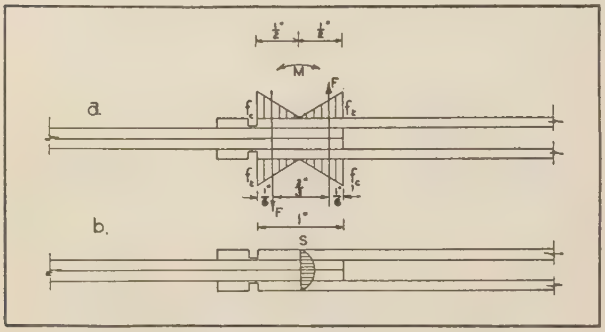

Fig. 8

center bending moment M and the internal resisting moment. With the pusher in position 1 (Fig. 5), tensile and compressive stresses are set up across the glue lines and are approximately triangular in distribution. The varying stresses f have resultants F, acting at the centers of gravity of the triangular stress volumes. The equal and opposite resultants F form a couple $F \times \frac{2}{3}^{\text{in.}}$ which resists the external moment M. The resultant F is equal to

$$F = \frac{2f}{2} \times \frac{1}{2}^{\text{in.}} \times 1^{\text{in.}}, \text{ and } F \times \frac{2}{3}^{\text{in.}} = \frac{f}{3} \qquad [6]$$

or

$$M = \frac{f}{3} \text{ and } f = 3M \qquad [7]$$

The actual stress distribution is somewhat different from the simplified diagram, here showm, inasmuch as M varies slightly from one end of the overlap to the other because of shear. This tends to increase f in some portions of the overlap and to decrease it in others. For practical purposes, the stress distribution is closely enough that shown.

Average shear stress in the plane of the glue lines is given by the familiar expression

$$s = \frac{VQ}{bI} \qquad [8]$$

and is as shown as in Fig. 8(b). This is the average condition. It is probable that the true shear stress deviates from this at the ends of the overlap.

TEST RESULTS

General

Test results are summarized in Tables 1 to 3.

Considerable scatter in the results is found, although trends are clearly visible. Marked difficulty was encountered in obtaining satisfactory failures at the glue lines; failure was as apt to occur in the wood as in the glue line. This was particularly true of the vibration specimens.

Static Tests

Static Shear. Table 1 gives the average results of static shear

TABLE 1 STATIC SHEAR

Panel	Average load, lb	Average shear stress, psi	Average per cent wood failure
A2	2857	1428	75
A3	2626	1313	83
A4	2842	1422	70
A5	2664	1332	99
A6	2788	1394	90
A7	2960	1458	95
Averages[a]	2916	1458	80
AA1	2864	1432	90
AA2	2851	1425	95
AA3	2929	1465	85
AA4	2700	1350	70
AA5	2706	1353	90
AA6	2792	1396	100
Averages[a]	2807	1403	90

[a] Averages of all individual tests, panel averages weighted in accordance with number of tests from each panel.

tests for the various panels. In the first set, average strength is 1458 psi, accompanied by 80 per cent average wood failure. In the second set the corresponding figures are 1403 and 90, respectively.

The shear-strength averages are less than are found in the standard shear block but are considerably higher than in the standard plywood shear specimens. The shape of the specimen and its general proportions, inducing variable amounts of tension because of eccentric loading, as well as nonuniform shear distribution in the joint, must account for the different values attained.

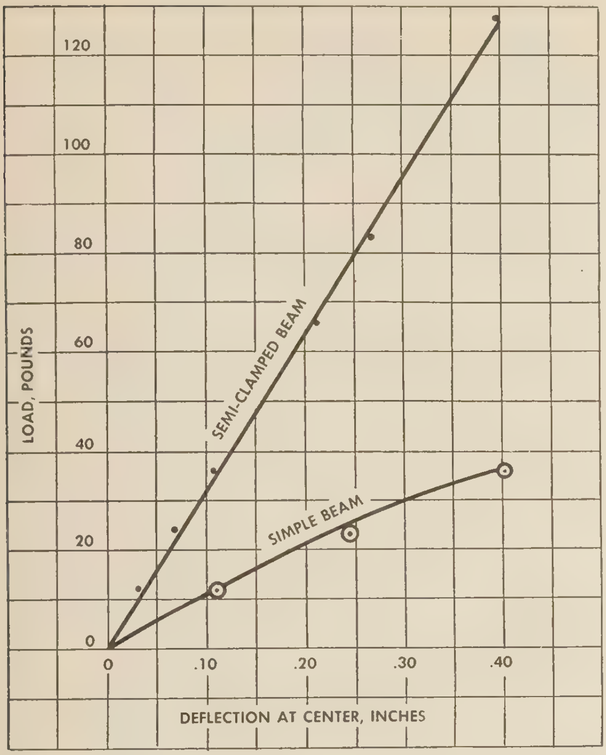

FIG. 9 TYPICAL LOAD-DEFLECTION DIAGRAMS; SIMPLE AND SEMI-CLAMPED BEAMS

Modulus of Rupture. Fig. 9 is a typical set of load-deflection curves for a modulus-of-rupture specimen. The lower curve is for the specimen simply supported, the upper for the semiclamped end. In general, deflections for the semiclamped condition were approximately one third those for the simple beam, indicating a center moment of approximately $\frac{WL}{7}$. Center moments for the clamped and semiclamped tests ranged from slightly better than $\frac{WL}{6}$ to $\frac{WL}{8}$.

Average values of modulus of rupture for the various panels tested, averages for the two sets of panels, and over-all averages are given in Table 2. Shear stresses were calculated by Equation [8] and maximum tension across the glue line from center moments determined from Equations [6] and [7].

The first set of panels yielded somewhat higher results than the second, but over-all averages and limits were as follows:

Shear: average 190, low 120, high 230 psi
Tension: average 815, low 530, high 1015 psi
Wood failure: average 65, low 45, high 100 per cent

Alternating Stresses. Table 3 summarizes the tests in which glue-line failures occurred under alternating stresses at amplitudes of $^5/_{16}$, $^1/_4$, $^3/_{16}$, $^1/_8$, and $^1/_{16}$ in. Tests in which wood failed in fatigue are not included. Furthermore, the results are for clamped or semiclamped specimens, and for specimens in which failure was complete, except for those which ran to 5,000,000 cycles without failure. The same results are presented graphically on logarithmic paper in Fig. 10. The following appear to be the chief points of interest:

1 Maximum amplitude, $^5/_{16}$ in. induced maximum tensile stresses perpendicular to the glue line approximately two thirds of those found in the modulus-of-rupture tests. Minimum ampli-

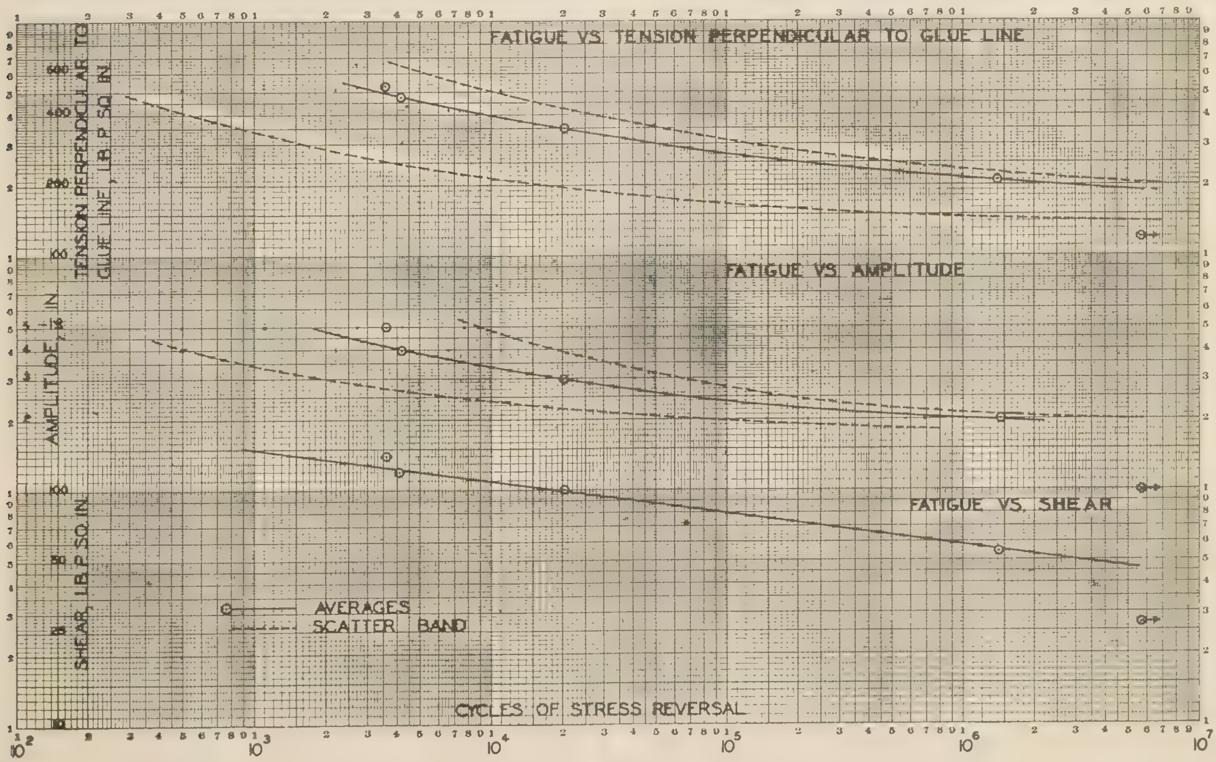

FIG. 10

TABLE 2 MODULUS OF RUPTURE

Panel	Load, lb	Shear stress, psi	Tension across glue line, psi	Per cent wood failure
A2	203	230	865	50
A3	132	150	695	75
A5	152	170	800	100
A6	148	165	775	90
Averages[a]	174	195	815	70
AA1	203	230	1015	50
AA2	187	210	935	90
AA3	170	190	850	95
AA4	167	190	835	45
AA5	172	195	860	70
AA6	106	120	530	45
Averages[a]	165	185	815	65
Total average	169	190	815	65

[a] Average of all specimens, not of averages.

TABLE 3 DEFLECTIONS, STRESSES, CYCLES

Specimen	Amplitude, in.	Shear stress, psi	Tensile stress, psi	Number of cycles	Per cent wood failure	Static shear Stress, psi	Static shear Wood failure
A3-Ea	$^5/_{16}$	135	510	8800	5		
A4-Bb		135	515	500	50		
A4-Ea		135	520	1100	60		
A5-Ba		135	520	7700	85		
A5-Ea		115	440	4400	40		
A6-Ea		140	540	1100	60		
AA1-Aa		175	660	2700	70		
Averages		140	530	3600	55		
A3-Rc	$^1/_4$	115	440	500	40		
A4-Rc		125	470	500	60		
A6-Rc		115	440	6600	60		
AA1-Ba		135	510	2200	15		
AA3-Ba		115	510	11000	20		
Averages		120	475	4200	40		
A3-Db	$^3/_{16}$	80	310	4400	10		
A4-Db		95	360	49500	40		
A5-Db		85	330	40700	50		
AA1-Ca		90	350	600	0		
AA3-Ca		145	420	7400	5		
Averages		100	350	20500	30		
A3-Cb	$^1/_8$	40	130	570000	80		
A4-Cb		50	190	220000	15		
A6-Cb		50	190	5780000	85		
AA2-Da		65	240	688000	30		
AA3-Da		65	280	1040000	5		
AA4-Da		50	230	149000	5		
Averages		55	210	1408000	40		
AA4-Ea	$^1/_{16}$	27	120	5770000	None	1733	95
AA5-Da		27	120	5770000	None	1462	85
AA5-Ea		27	120	5770000	None	1862	90
AA6-Da		26	115	5770000	None	1347	45
AA6-Ea		26	115	5770000	None	1712	65
Averages		27	120	5770000	None	1603	75

tude, $^1/_{16}$ in. induced tensile stresses approximately one seventh of those found in the modulus-of-rupture tests.

2 Shear stresses were quite low, compared with the static shear tests; the maximum at $^5/_{16}$ in. amplitude was 5 per cent of the static shears.

3 Failures were primarily tension perpendicular to the glue line and were the kind that would be found in most actual structures.

4 At $^1/_8$-in. amplitude, one specimen ran to better than 5,000,000 cycles. The others failed at approximately 150,000 to 1,000,000 cycles.

5 At $^1/_{16}$-in. amplitude, all specimens ran to better than 5,000,000 cycles without any signs of failure.

6 When tested in static shear, the unbroken vibrated specimens yielded results which actually averaged a little higher than the static controls. This indicated that at least there was no loss in strength

7 In those vibration specimens which failed during test, lower percentages of wood failure were found than in the modulus-of-rupture controls. No definite trend with respect to numbers of cycles and stress could be discerned.

8 In Fig. 10, the numbers of cycles of stress reversal are plotted against amplitude, shear stress, and maximum tension perpendicular to the glue line. Curves are drawn for the arithmetical averages and for the approximate limits of the scatter bands in the graphs of cycles versus amplitude and cycles versus tension perpendicular to the glue line. Scatter for the shear results is not plotted, inasmuch as shear was a minor factor in causing failure. The curves of amplitude and maximum tension versus cycles to failure flatten to practically zero slope well above the 5,770,000-cycle point.

Conclusions

Conclusions must be drawn with some reservations from these tests, inasmuch as the scatter in the results renders generalizations somewhat difficult. It appears, however, that the following may be said with some assurance with respect to stresses at room temperatures:

1 A joint having the general proportions of those tested, and subjected to vibrations, is as apt to fail in fatigue in the wood as in rupture in the glue line.

2 Inasmuch as the sections tested were quite heavy and spans were short, so that higher shears were induced in the glue lines than would customarily be found, it seems safe to say that failures in glue lines of this type will occur primarily because of tension perpendicular to the glue line (and the grain of the wood) rather than by shear in the glue line.

3 If maximum tensile stresses perpendicular to the glue lines of these adhesives are kept below 120 psi, there should be no failures. As a matter of fact, extrapolation of the observed curves indicates that it is highly probable that stresses can be 150 to 160 psi, that is, 20 per cent of the "modulus of rupture," or static ultimate stresses. If stresses reach 200 psi, failure may be expected to occur at 5,000,000 cycles or less, and as the stresses go still higher, the number of cycles to failure becomes rapidly smaller, with highly variable results. The "fatigue limit," if that term may be employed here, appears to be 20 per cent of the ultimate.

The foregoing conclusions are based upon tests made at room temperature upon specimens at approximately 9 per cent moisture content.

Application of these results depends upon the ability to determine stresses in vibrating structures, by analysis or test. Wherever members cross each other, bending moments are apt to set up tensile stresses perpendicular to the glue line and the grain of the wood. Such stresses may be uniformly distributed or may vary from maximum to minimum approximately as in the tests here described. In such instances, the dimensions of the overlapping parts should be large enough to keep the tensile stresses from exceeding 20 per cent of the tensile strength of the wood perpendicular to the grain. For species such as maple and birch, alternating tensile stresses of 125 to 150 psi may be employed, for other species such as spruce or mahogany, the stresses will be less.

Acknowledgments

Acknowledgments are due particularly to John Barry, Research Assistant, who carried out the experimental work, and to Prof. W. M. Murray, of the Department of Mechanical Engineering at the Massachusetts Institute of Technology, who was instrumental in the design of the testing apparatus. The laboratory staff of the Resinous Products and Chemical Company made the test material.

Heat and Vapor Transfer in the Dehydration of Prunes

By R. L. Perry,[1] Davis, Calif.

Quality of prunes is known to be influenced by the temperature and relative humidity of the dehydrator air, and by the time required for drying, long exposures to high temperatures at high humidities resulting in excessive caramelization and cooked taste. To determine the actual temperatures of prunes during dehydration, readings were made by thermocouples inserted at different points in prunes for three test runs at about 166 F, and relative humidities of 15, 27, and 43 per cent. To permit adapting the data obtained to commercial dehydrators, surface thermal conductances were computed. The vapor conductance at the surface was estimated from the thermal conductance, by employing the heat-mass transfer analogy. From the vapor conductance and the evaporation rate, the difference in vapor concentration required to move vapor from the prune surface to the air stream was calculated. Moisture movement within the prune was investigated by cutting the flesh of samples, removed after 3, 6, 10, and 16 hr drying, into two parts, that near the surface, and that near the pit.

Nomenclature

The following nomenclature is used in this paper:

A = area of prune surface, sq ft
A_d = area of prune surface, per pound of dry matter, sq ft per lb
B = drying coefficient, slope of free-moisture line on semilogarithmic paper, 1/hr
c = unit heat capacity, Btu/lb F
c_d = heat capacity of prune, per pound of dry matter, Btu/lb F
C = moisture concentration, lb per cu ft
D = diameter, ft
e = base of Naperian logarithms, 2.7183
E = equilibrium-moisture ratio, pounds of moisture per pound of dry matter
f = unit surface thermal conductance, Btu/hr ft^2 F
f' = unit surface vapor conductance, lb/hr ft^2 (lb/ft^3) or ft/hr
F = free-moisture ratio, pounds of free moisture per pound of dry matter. ($F = M -$ E)
H = relative humidity, per cent
k = thermal conductivity, Btu/hr ft^2 (F/ft)
k' = moisture conductivity within prune, lb/hr ft^2 (lb/ft^4) or ft^2/hr
M = total-moisture ratio, pounds of moisture per pound of dry matter
n = exponent of Prandtl modulus in forced-convection equations
N = prune count, number of dried prunes, of 0.2 moisture ratio, per lb
q_c = convective- and radiant-heat-exchange rate, between prune and its surroundings, Btu per hr
q_s = sensible-heat-gain rate of prune, Btu per hr
q_v = heat rate in evaporation, Btu per hr
r = latent heat of evaporation of water, Btu per lb
R = radius, ft
t = temperature, F
V = air velocity in free cross section of tunnel, fpm
w = evaporation rate, lb per hr
W = weight of prunes
α = diffusivity of water vapor through air, ft^2/hr
γ = weight density, lb per cu ft
θ = time, hr
μ = weight viscosity, lb/hr ft

Subscripts:

i, for prune-air interface
0, for outside
∞, for remote air stream
1, initial

Dimensionless moduli:

fD/k = Nusselt modulus for convective heat transfer at a fluid boundary
$f'D/\alpha$ = Nusselt modulus for convective mass transfer at a fluid boundary
$c\mu/k$ = Prandtl modulus of fluid properties for convective heat transfer
$\mu/\alpha\gamma$ = Prandtl modulus of fluid properties for convective mass transfer
$k'\theta/R^2$ = Fourier modulus for mass conduction in solids
$f'R/k'$ = Biot modulus for mass transfer at a solid-fluid interface

Introduction

In this paper, data on the temperature and moisture distribution in prunes during dehydration are presented. Surface conductances for heat and vapor are computed from the data, and the vapor concentration and relative humidity at the prune surface are found. The deviation of moisture distribution within the prune from that which would occur with constant moisture diffusivity is disclosed.

Studies by Moses, Cruess, et al (1)[2] fixed the over-all limiting dehydrator conditions of temperature and humidity for production of prunes of high quality. Although it was known that prolonged high air temperature and high humidity gave a dark, caramelized, low-grade prune, the actual prune temperatures during drying were not observed. In order to extend the results of their dehydration tests made at constant air temperature and humidity to commercial practice, where counterflow, parallel-flow, center-exhaust, end-exhaust, and cross-flow dehydrators are used, the prune temperature itself was needed.

The empirical equations developed by Guillou (2) led to the first marked advance in prune-dehydrator design technique since

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Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to Bibliography at the end of the paper.

the work of Christie and Ridley (3), but they left several questions unanswered. No explanation was available for the observation that variations in relative humidity did not influence drying rate unless the relative humidity exceeded 35 per cent. This was contrary to the observation that in some other materials the drying rate is proportional to wet-bulb depression. Another unanswered question was why the equilibrium-moisture content, determined by plotting drying rate against moisture ratio, differed from that observed by Guillou (4), Table 2, on prunes held in atmospheres of different humidities until constantweights were reached.

The investigations reported herein were planned to throw more light on the answers to these questions. No previous attempts to measure either temperature or moisture in fruit-drying dehydration had been noted in the literature.

The Dehydration Process

When a fresh prune is placed in a high-temperature air stream of a moisture concentration lower than that which represents equilibrium with the prune flesh at the air temperature, heat is transferred by convection and radiation to the prune surface; thus

$$q_c = fA(t_\infty - t_i) \qquad [1]$$

At first this heat results principally in a rise in temperature

$$q_s = cW\frac{\partial t}{\partial \theta} \qquad [2]$$

in which t represents the average temperature of the prune, the interior being warmed by conduction from the surface. However, as the surface layers rise in temperature, the vapor concentration at the surface rises. When it exceeds that in the adjacent air stream, vapor escapes through the resistance offered by the adjacent air-vapor mixture

$$w = f'A(C_i - C_\infty) \qquad [3]$$

The vaporization absorbs heat

$$q_v = wr \qquad [4]$$

or, combining Equations [3] and [4]

$$q_v = rf'A(C_i - C_\infty) \qquad [5]$$

(An additional quantity of heat is necessary to warm the vapor from the temperature at which it escapes from the prune to that which it attains in the air stream. This is a minor item which can be neglected, especially since only part of it has to pass through the resistance of the adjacent air-vapor mixture.) The heat available from convection and radiation is utilized in temperature rise and evaporation

$$q_c = q_s + q_v \qquad [6]$$

As the temperature rises, the heat available from convection and radiation decreases, but the evaporation rate increases, so that there is soon little heat available for further temperature rise. The prune temperature is then, combining Equations [1] and [5] and noting that q_s is negligible in Equation [6]

$$t_i = t_\infty - \frac{f'r}{f}(C_i - C_\infty) \qquad [7]$$

Equation [7] is of limited utility because of inability to establish the value of the surface-moisture concentration C_i, except indirectly from test data. The surface concentration is related to the moisture ratio of the prune flesh, and to the temperature, but varies with the drying history of the prune. It is evident, of course, that as vapor escapes from the surface, a moisture-concentration gradient is established within the prune, so that moisture moves to the surface, but not rapidly enough to maintain the initial concentration. The evaporation rate thus decreases gradually, and the temperature approaches more closely that of the air. Even with the help of the data at hand, however, the formulation of an ideal system, which would describe adequately the dehydration of a prune, has not been realized. The continually changing balance between vapor and heat transfer which determines the drying rate and the prune temperature is not yet mathematically predictable, and reference must still be made to empirical relations such as are presented herewith.

Test Apparatus and Procedure

Two groups of tests were conducted. In the first group, simultaneous observations of weight of sample and of temperature at prune surface and prune pit were made during constant-condition dehydration runs. The initial and final dimensions, volume, and density were noted, and the final moisture content measured. In the second group, several prunes were removed at successive drying times, and the flesh was dissected into two sections, that near the skin and that near the pit, for moisture-content determination. Subsequently, the dried skin was planimetered to find its area, in order to learn the area and thickness of the segments which had been removed.

Temperature-Distribution Tests. In each run of the first group,

Fig. 1 Test Tray of Prunes, at End of Run

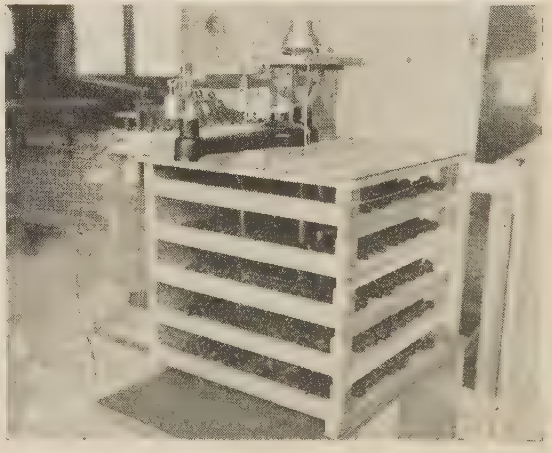

Fig. 2 Test Tray, Surrounded by Guard Trays

temperatures at three points in each of five prunes were taken by means of No. 36 gage copper-constantan thermocouples, read with a Leeds & Northrup No. 8662 portable semiprecision potentiometer. One thermocouple was inserted just under the skin at the top of each prune, the junction being so close to the surface that it could be seen through the translucent skin, and the leads continuing just under the skin for about $^5/_8$ in., to minimize conduction errors. One thermocouple was placed next to the pit. In three of the five prunes, the third point was just beneath the skin on the side which was to lie on the tray, while in the other two, the third point was just under the skin at the side.

These five test prunes were placed among twenty others of similar size and maturity on a tray of $^1/_4$-in. plywood, which could be weighed as drying proceeded, the thermocouple extension leads serving as suspensions from the scales. This tray, Fig. 1, was hung in the center of a larger guard tray which was included in the center of a stack of five, Fig. 2, placed in the dehydrator tunnel, 17 in. square in cross section. Only the lower tray in Fig. 1 was weighed, the two above it being access hatches which fitted into the two upper guard trays. Although the suspending leads were sheltered from the air stream by the tubular shields noticeable in Fig. 1, and the tray was restrained laterally by bridles, it was found necessary to close the air-control damper while weighing was being done.

Four runs were made, each at about 166 F air temperature, and at an air velocity of 600 fpm, in the free cross section of the tunnel. It was found during the first run that the thermostat was in need of repair, having a range of 7 F, so that, although the periodic heat exchange with the prune was of some interest, the data were of little direct value. Subsequent runs were made with manual control, an auxiliary heater being adjusted with a variac. General data for these runs are given in Table 1.

TABLE 1 GENERAL DATA ON PRUNE TEMPERATURE TESTS

Run number	2	3	4
Average air temperature, deg F	169	166.5	166
Average wet-bulb temperature, deg F	122	134	105
Average relative humidity, per cent	27	43	15
Tray loading, lb per sq ft	3.98	3.90	3.94
Duration of run, hr	16.1	18	17.2[a]
Final-moisture content,[b] 20 prunes, per cent	20.3	21.0	19.3
Average weight, lb:			
Initial, 5 temp test prunes	0.0500	0.0511	0.0519
Initial, 20 additional prunes	0.0502	0.0485	0.0488
Final, 5 temp test prunes	0.0163	0.0168	0.0186
Final, 20 additional prunes	0.0169	0.0170	0.0168
Moisture ratio, pounds of moisture per pound of dry matter:			
Initial, average of 25 prunes	2.76	2.67	2.58
Final, average of 25 prunes	0.26	0.27	0.24
Average volume, cu ft:			
Initial, 5 temp test prunes	0.000713	0.000728	0.000757
Final, 5 temp test prunes	0.000205	0.000212	0.000233
Average density, lb per cu ft:			
Initial, 5 temp test prunes	70.2	70.2	68.7
Final, 5 temp test prunes	79.6	79.3	69.9
Axial dimensions, 5 temp test prunes, ft:			
Initial major axis	0.134	0.135	0.133
Initial horizontal minor axis	0.102	0.102	0.102
Initial vertical minor axis	0.100	0.101	0.101

[a] Tunnel air damper inadvertently shut at 1 hr, opened at 1.58 hr.
[b] Wet basis.

After completing each run, the final moisture content of the twenty companion prunes was determined with the electric-resistance moisture tester of the Dried Fruit Association. The thermocouples were examined to discover if they had inadvertently been misplaced in completing the lead connections or in installing trays in the tunnel. In general the top and side thermocouples influenced the wrinkling in such a way that the junction was on a ridge, as shown in prune B, Fig. 3, with the following exceptions:

Run 2, prune 3: Top thermocouple in a ridge, but not at apex, being about $^1/_{16}$ in. or $^1/_3$ of ridge height below apex, see A, Fig. 4.
Run 3, prune 4: Top thermocouple in valley, as B, Fig. 4.
Run 4, prune 2: Top thermocouple in valley, as B, Fig. 4.
Run 4, prune 2: Side thermocouple near ridge at edge of bottom, as C, Fig. 4.
Run 4, prune 5: Side thermocouple in double ridge, as D, Fig. 4.

As the prunes softened with rise in temperature and loss of moisture, the bottom began to flatten, so that at about 2 hr of drying, the flat spot was about $^1/_4$ as long and as wide as the prune. Wrinkling was just discernible at this time. After 6 to 8 hr, the flat spot was about $^2/_3$ as long and as wide as the prune.

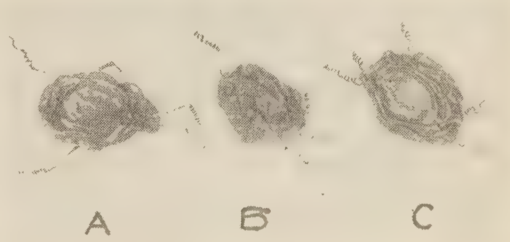

FIG. 3 DRIED PRUNES CONTAINING THERMOCOUPLES

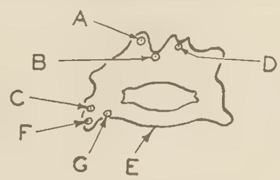

FIG. 4 CROSS SECTION OF DRIED PRUNES SHOWING THERMOCOUPLE LOCATIONS

As the end of drying was approached, the bottom shrank up toward the pit, Fig. 3, prunes A and C, so that it was cupped with a slight convexity at the middle, Fig. 4, E. The bottom junction was usually found just under the pit, in the bulge, with the following exceptions:

Run 3, prune 1: Bottom thermocouple at ridge at edge of bottom, as F, Fig. 4.
Run 3, prune 5: Bottom thermocouple near ridge at edge of bottom, as G, Fig. 4.

Moisture-Distribution Tests. Two runs were made for measuring moisture distribution, each at 166 F and 105 F wet-bulb temperature. In these runs, four lots of similar prunes were weighed and placed on the trays for drying. After 3 hours, samples from the first lot were removed, weighed, and the skin, with an adhering adjacent layer of flesh, was sliced off for a moisture determination. The flesh adhering to the pit was also removed, and moisture determinations were made on it and on the pit. A similar procedure was followed at 6, 10, and 16 hr. In the second run, the slices from the top, sides, and bottom of the outer layer were kept separate. Some loss, from 6 per cent in the 3-hr sample, to 2 per cent in the 16-hr sample, occurred in the dissecting, and this was allotted to the inner and outer layers on a proportional-weight basis.

The moisture determinations were made by drying to constant weight in an atmospheric oven at 65 C (149 F), for want of a vacuum oven. After the moisture was found, the skin sections of the second run were soaked, the flesh removed, and the skin was dried in small pieces for the purpose of measuring its area with a planimeter. This procedure permitted estimating the average thickness of the top, bottom, and side slices of the second run.

Test Results

Temperature Distribution. The temperatures of the prunes rose rapidly in the first half hour, until within about 20 F of the air temperature, as shown in Fig. 5. They then rose more slowly, until at the end of drying, the surface temperatures were only 2 to 3 deg below air temperature. Because of trouble in

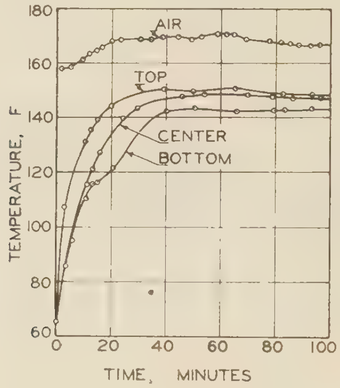

Fig. 5 Temperatures of Prune During Initial Warming; Run 2, Prune 5

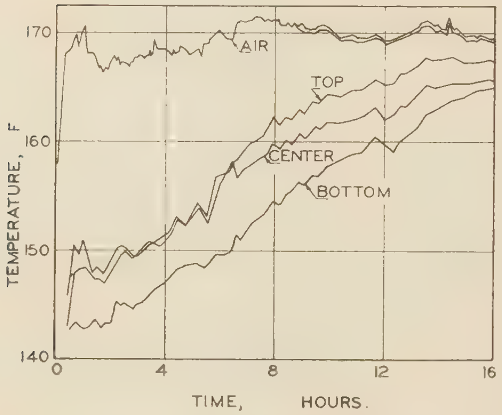

Fig. 6 Graph of Original Temperature Data; Run 2

controlling the air temperature precisely (there was a periodic fluctuation of 1 to 2 deg in air temperature even with a steady heat input, noticeable in the air-temperature curve in Fig. 6) and of the rapidity with which the prunes tended to respond to a change in air temperature, the data were first plotted as broken lines from point to point, Fig. 6. From such preliminary graphs, smoothed curves were sketched by eye, with consideration of any air-temperature changes which were occurring simultaneously. The air temperature, the mean of the prune surface, and of the prune-center temperatures, and the weight of the 25 prunes are shown for Runs 2, 3, and 4, in Figs. 7, 8, and 9, respectively. The maximum and minimum prune temperatures are also shown in these figures, the curves overlapping at some points as one prune rose above or fell below the others.

The considerable variations in temperatures of prunes in the same runs are illustrated in Figs. 10, 11, and 12. Thus in Run 2, Fig. 10, center temperatures of two prunes differed as much as 13 F at one time, nearly as much as the difference of the mean of the five prunes from the air temperature. In Run 3, Fig. 11, however, there was much less variation. That the peculiar dip in the curve for the coldest prune at $12^1/_2$ hr in Run 4, Fig. 9, is not an error is substantiated by the additional curves for this run, Fig. 12, where it can be seen that the temperature of several, but not all, of the points dropped temporarily at this time. The drop at $13^1/_2$ hr was obviously due to a drop in air temperature. No explanation is available for this peculiar behavior, except possibly sticking and subsequent loosening from

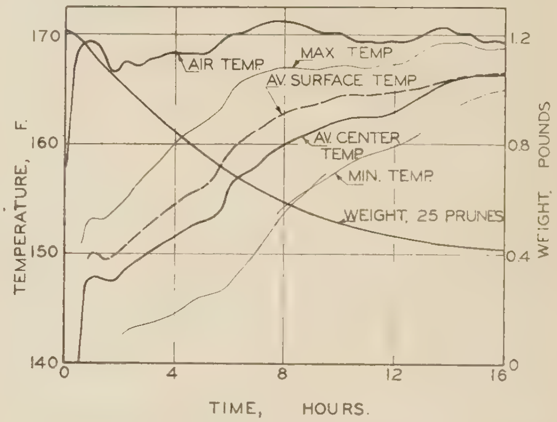

Fig. 7 Weight of 25 Prunes, Air Temperature, and Average Temperature of Prune Surface and Prune Center; Run 2

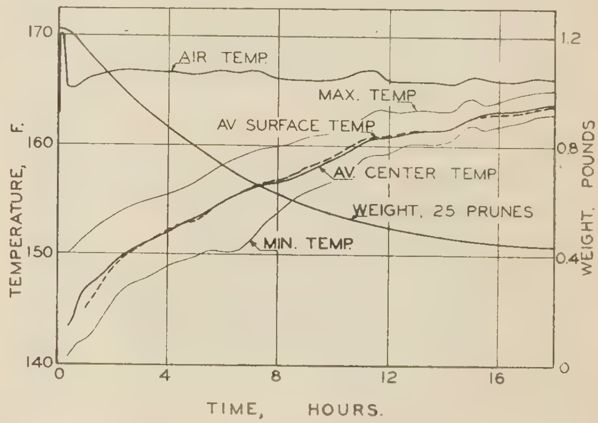

Fig. 8 Weight of 25 Prunes, Air Temperature, and Average Temperature of Prune Surface and Prune Center; Run 3

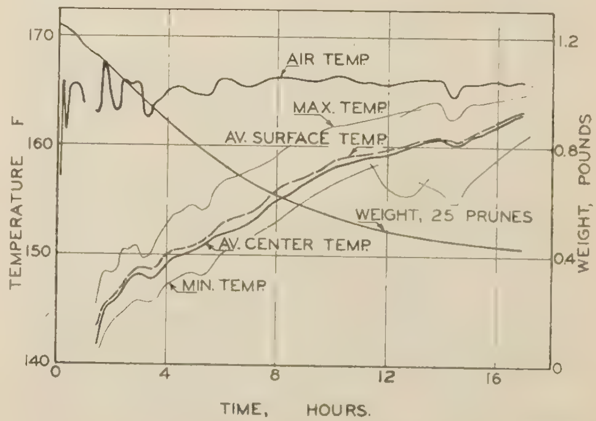

Fig. 9 Weight of 25 Prunes, Air Temperature, and Average Temperature of Prune Surface and Prune Center; Run 4

the tray caused by shrinking. This would permit part of the prune to dry slowly for a time, and then evaporate rapidly when it loosened, giving a temporary depression for a limited area.

The mean difference between the top-, side-, or bottom-surface temperature and the center temperature, was computed by giving the side point 4 times the weight accorded the top and bottom points, because the arrangement of the prunes on the tray suggests that about these proportions of its surface have similar exposures. Unfortunately, this was not realized when the tests were planned, so that the points were not placed in these proportions. From the mean center temperature of the five prunes, and the mean surface-to-center difference, the mean surface temperature was found. From the surface temperature and the air temperature, the differences between the air and the prune-surface temperatures were obtained, as shown in Fig. 13.

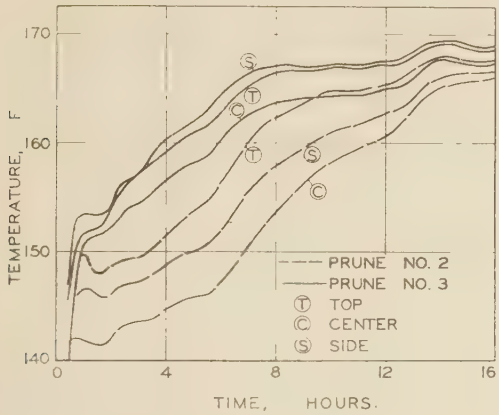

Fig. 10 Temperatures of Top, Side, and Center; Prunes Nos. 2 and 3, Run 2

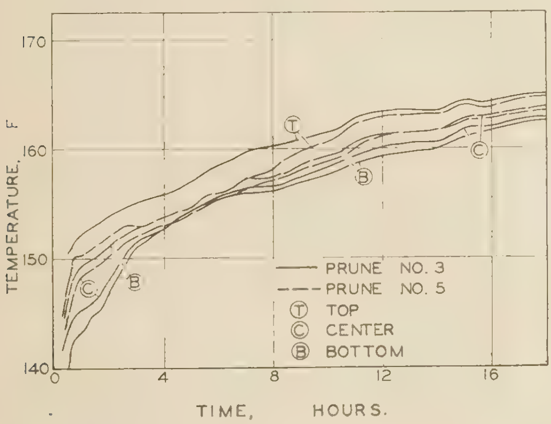

Fig. 11 Temperatures of Top, Center, and Bottom; Prunes Nos. 3 and 5, Run 3

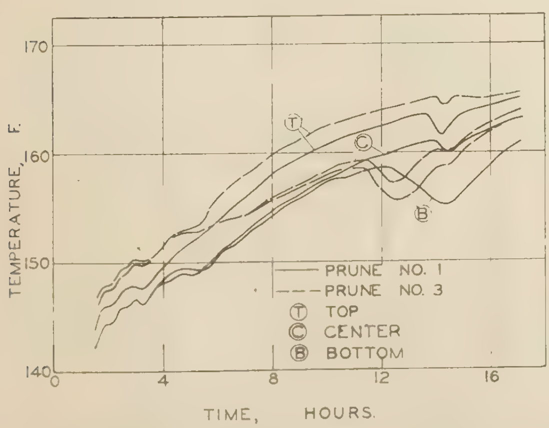

Fig. 12 Temperatures of Top, Center, and Bottom; Prunes Nos. 1 and 3, Run 4

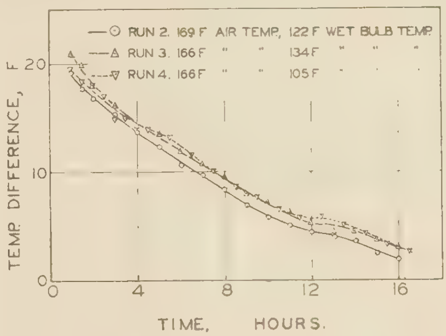

Fig. 13 Difference Between Air and Prune Average Surface Temperature

Evaporation Rate. The evaporation rate is indicated by graphs of free-moisture ratio (pounds of free moisture per pound of dry matter), plotted against time on semilogarithmic coordinates, Figs. 14, 15, and 16. The free-moisture ratios were calculated from the prune-weight data, the final-moisture content, and the equilibrium-moisture data, Table 2. The points on these curves did not fall on a single straight line, the slope becoming greater in the middle of the drying period and falling off at the end. For each straight section, the moisture ratio is

$$F = F_1 e^{-B\theta} \qquad [8]$$

and the evaporation rate

$$\frac{dF}{d\theta} = -BF_1 e^{-B\theta} = -BF \qquad [9]$$

TABLE 2 EQUILIBRIUM-MOISTURE RATIOS FOR FRENCH PRUNES AT 150 F (4)

Relative humidity, per cent	Moisture ratio: Pounds moisture / Pounds dry matter
18	0.044
28	0.052
62	0.34
74	0.63
81	1.05
95	1.75

Heat Transfer. The rate of heat transfer to the prune was taken to be the sum of the sensible- and latent-heat rates. The sensible-heat rate per pound of dry matter is the product of the rate of temperature rise and the heat capacity per pound of dry matter. The unit heat capacity of cellulose and levulose being about 0.3 Btu/lb F, the heat capacity per pound of dry matter is

$$c_d = 0.3 + 1.0\,M \qquad [10]$$

The latent-heat rate per pound of dry matter was taken to be the latent heat of water at the temperature of the surface, multiplied by the evaporation rate per pound of dry matter. This calculation neglects the heats of solution of dissolved components, which are not known, but which are probably small. Except during the first half hour, when the rate of temperature rise is high, and the rate of evaporation low or possibly negative, the sensible-heat rate is less than 1 per cent of the latent-heat rate.

In order to compute the unit surface conductances, it was first

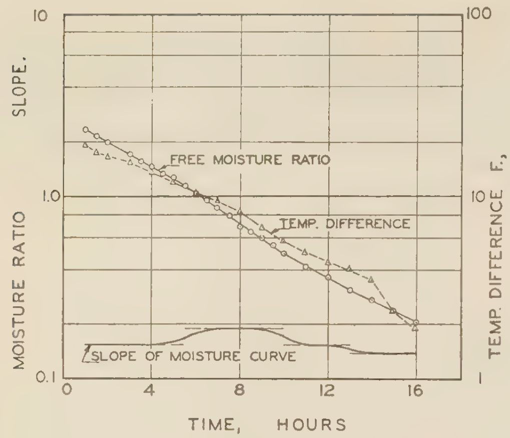

FIG. 14 FREE-MOISTURE RATIO AND TEMPERATURE DIFFERENCE; RUN 2

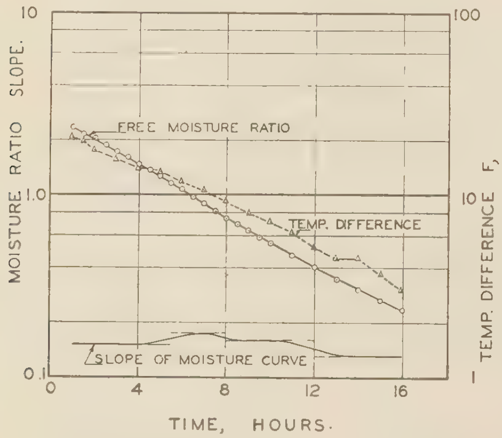

FIG. 15 FREE-MOISTURE RATIO AND TEMPERATURE DIFFERENCE; RUN 3

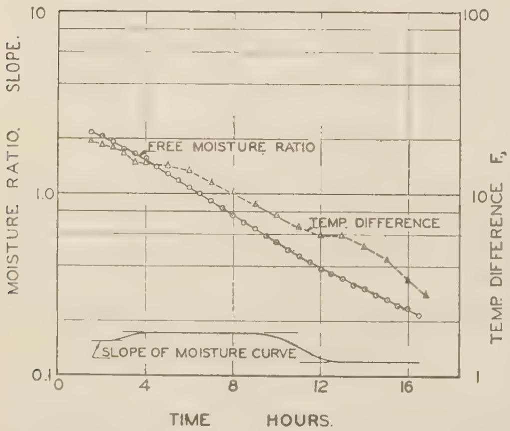

FIG. 16 FREE-MOISTURE RATIO AND TEMPERATURE DIFFERENCE; RUN 4

necessary to find the surface area of the prunes. The initial surface area was taken to be that of a spheroid with a major axis that of the prune, and a minor axis the mean of the two minor axes, which were nearly equal. The final surface area was found from the area data of the dried samples in the second group of tests. The 3, 6, 10 and 16-hr samples were found to have a dried-skin area of 70, 67, 74, and 62 per cent, respectively, of the initial spheroid surface, averaging 68 per cent. Lineal shrinkages of strips cut in two directions around a prune were 78 and 82 per cent, representing an area shrinkage of 64 per cent, but the figure of 68 per cent from the second group of tests was thought to be more reliable. In order to establish the surface area as a function of the moisture ratio, an additional point was obtained from the observation that the prunes were beginning to wrinkle at about 2-hr drying time, when the moisture ratio was about 2.05. At this time, the moisture loss has been primarily from the outer layers, and the volume shrinkage has corresponded to the skin-area shrinkage so that wrinkling has not occurred. After this, the skin does not dry or shrink as rapidly as the prune does, so that wrinkling results.

The relation between prune volume and moisture ratio was found by plotting average density, as found from the initial and final volume and weight data, against moisture content on a wet basis, giving nearly a straight line, Fig. 17. To facilitate calculations, this was converted to density as a function of moisture ratio, and is given with prune-volume and surface-area data in Fig. 18. The volume at a moisture ratio of 2.05 was then used to fix the surface area of the spheroid at this moisture ratio, wrinkling just having started as previously noted. The surface area-moisture ratio curve in Fig. 18 is strictly valid only for the dehydrator conditions of the tests. In counterflow, and other dehydrator arrangements, where the temperature and humidity change from point to point, the relation between the moisture

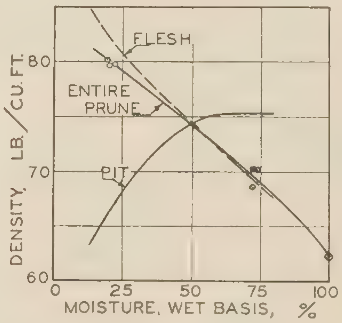

FIG. 17 PRUNE-MOISTURE CONTENT AND DENSITY

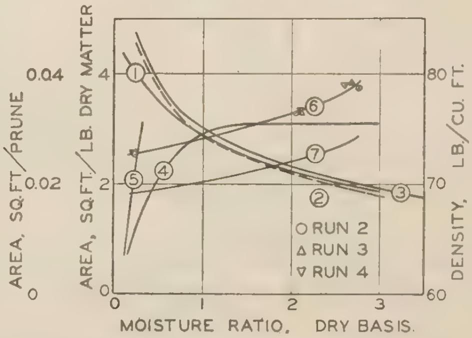

FIG. 18 PRUNE-MOISTURE RATIO, SURFACE AREA, AND DENSITY

Curve 1 Density of entire prune vs. moisture ratio of prune
Curve 2 Density of prune flesh vs. moisture ratio of prune
Curve 3 Density of prune flesh vs. moisture ratio of flesh
Curve 4 Density of pit vs. moisture ratio of prune
Curve 5 Density of pit vs. moisture ratio of pit
Curve 6 Surface area, sq ft per prune vs. moisture ratio of prune
Curve 7 Surface area, sq ft per lb of dry matter vs. moisture of prune

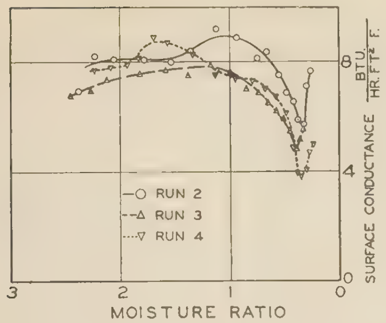

FIG. 19 SURFACE THERMAL CONDUCTANCE PER SQUARE FOOT

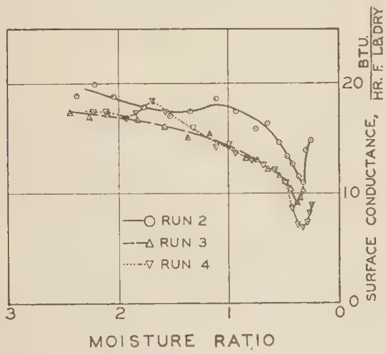

FIG. 20 SURFACE THERMAL CONDUCTANCE PER POUND DRY MATTER

content of the surface and interior will vary, and the surface area-moisture ratio curves will differ somewhat.

With the surface-area, temperature-difference, and heat-rate data in hand, the unit surface thermal conductances were calculated, both on a prune-surface area and a dry-weight basis, and are given in Figs. 19 and 20. The conductance on a surface-area basis is the conventional conductance, but that on a dry-weight basis is more readily usable and avoids the uncertainty of the actual prune-surface area. The conductance at first increases as drying proceeds, remains nearly constant for a time, drops appreciably near the end of drying, and then seems to rise again. The final drop and rise are puzzling, but were observed in all three runs.

Vapor Conductance and Concentration. In order to find the difference in vapor concentration at the prune surface and in the air stream, the unit surface vapor conductance is required. This is obtained from the observed thermal conductance, using the heat-mass-transfer analogy. According to Boelter, Cherry, and Johnson (5), the Nusselt modulus for mass transfer ($f'D/\alpha$) can be predicted from the Nusselt modulus for heat transfer (fD/k), and the Prandtl moduli for mass and heat transfer, ($\mu/\alpha\gamma$ and $c\mu/k$), as follows

$$\left(\frac{f'D/\alpha}{fD/k}\right) = \left(\frac{\mu/\alpha\gamma}{c\mu/k}\right)^n \qquad [11]$$

where n is the experimentally determined exponent of the Prandtl modulus for the analogous heat-transfer system. McAdams (6) recommends a value of 0.3 for n for fluids flowing normal to tubes and banks of tubes. Canceling in Equation [11], and using 0.3 for n

$$f'/\alpha = f/k(k/c\gamma\alpha)^{0.3} \qquad [12]$$

At 166 F, the value of k for dry air (International Critical Tables) is 0.0156 Btu/ft hr F, and from Hilpert's data, reported by Boelter, Cherry, and Johnson (5), α is 1.28 sq ft per hr. The density of dry air at this temperature is 0.063 lb per cu ft and the unit heat capacity is 0.24 Btu/lb F. Substitution of these values in Equation [12] gives, for the conditions of these tests

$$f' = 77f \qquad [13]$$

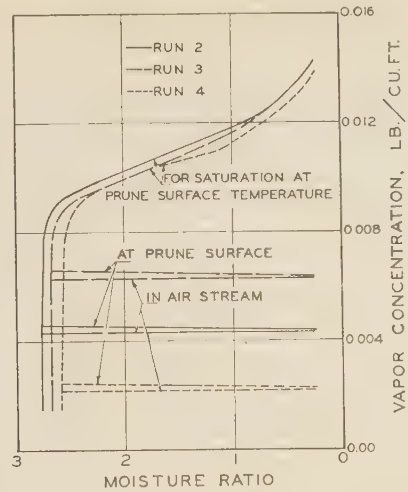

FIG. 21 VAPOR CONCENTRATION AT PRUNE SURFACE

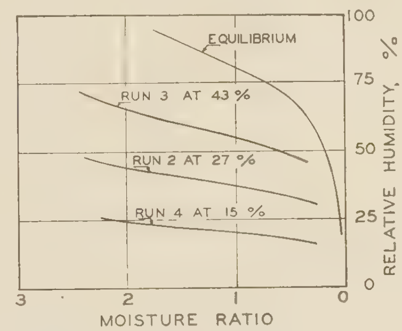

FIG. 22 RELATIVE HUMIDITY AT PRUNE SURFACE

The air-vapor mixture adjacent to the prune has an appreciable humidity and a temperature below 166 F, particularly at early stages of drying, but corrections for temperature and humidity will not change the constant of 77 by more than 1 per cent. As shown in Equation [3], the vapor-concentration difference depends upon the evaporation rate and the surface conductance. Rearranging Equation [3], and introducing the equivalent of f' from Equation [13]

$$C_i - C_\infty = 0.013\, w/fA \qquad [14]$$

It was more convenient to use, instead of the evaporation rate w, the rate per pound of dry matter from Equation [9], $-BF$, and the surface area per pound of dry matter, A_d, which gave

$$C_i - C_\infty = -0.013BF/fA_d \qquad [15]$$

Vapor-concentration differences and vapor concentrations at the prune-air interface are given in Fig. 21. The vapor concentrations for saturation at the prune-surface temperatures, needed in finding the relative humidity at the prune-air interface, are also shown. These curves are based on a uniform air temperature of 166 F, using the data in Fig. 13, to give prune-surface temperatures, and adding the vapor-concentration differences to

TABLE 3 PRUNE MOISTURE-DISTRIBUTION DATA

Time of removing sample, hr	3	6	10	16
Weight, lb:				
Initial, top flesh	0.0053	0.0085	0.0075	0.0054
Initial, side flesh	0.0191	0.0185	0.0215	0.0232
Initial, bottom flesh	0.0057	0.0062	0.0059	0.0048
Initial, inner flesh	0.0121	0.0091	0.0095	0.0114
Initial, pit	0.0022	0.0024	0.0023	0.0024
Initial, entire prune	0.0443	0.0446	0.0467	0.0473
Final, top flesh	0.0036	0.0042	0.0027	0.0017
Final, side flesh	0.0137	0.0099	0.0079	0.0075
Final, bottom flesh	0.0042	0.0038	0.0026	0.0016
Final, inner flesh	0.0107	0.0061	0.0045	0.0041
Final, pit	0.0022	0.0024	0.0022	0.0022
Final, entire prune	0.0344	0.0264	0.0199	0.0170
Moisture ratio, pounds of moisture per pound of dry matter:				
Initial, all flesh	3.40	3.34	3.04	2.82
Initial, pit	0.34	0.34	0.34	0.34
Initial, entire prune	2.90	2.83	2.63	2.46
Final, top flesh	1.91	1.11	0.43	0.20
Final, side flesh	2.15	1.28	0.46	0.22
Final, bottom flesh	2.16	1.64	0.75	0.20
Final, inner flesh	2.82	1.85	0.87	0.36
Final, pit	0.34	0.33	0.27	0.17
Final entire prune	2.03	1.26	0.55	0.24
Density, lb per cu ft:				
Initial, all flesh	69.0	69.1	69.5	69.9
Initial, pit	75.5	75.5	75.5	75.5
Initial, entire prune	69.3	69.6	69.8	70.2
Final, top flesh	71.8	74.5	80.5	84.5
Final, side flesh	71.3	73.7	80.2	84.2
Final, bottom flesh	71.3	72.5	76.9	84.7
Final, inner flesh	69.9	72.0	76.0	81.6
Final, pit	75.5	75.1	71.5	66.0
Final, entire prune	70.9	73.3	77.2	80.2
Free-moisture concentration, lb per cu ft:				
Initial, all flesh	53.1	53.0	52.1	51.3
Final, outer flesh	48.1	41.1	25.4	12.4
Final, inner flesh	51.5	46.3	34.4	20.0
Free-moisture concentration ratio, final/initial:				
Outer flesh	0.91	0.78	0.49	0.24
Inner flesh	0.97	0.88	0.66	0.39
Volume, cu ft:				
Initial, outer flesh	0.000436	0.000481	0.000503	0.000477
Initial, inner flesh	0.000174	0.000132	0.000137	0.000163
Initial, pit, and 1/2 inner flesh	0.000117	0.000097	0.000099	0.000114
Initial, pit, inner, 1/2 outer flesh	0.000422	0.000414	0.000418	0.000434
Initial, entire prune	0.000639	0.000644	0.000671	0.000673
Final, top flesh	0.000050	0.000056	0.000034	0.000020
Final, side flesh	0.000192	0.000134	0.000098	0.000089
Final, bottom flesh	0.000058	0.000052	0.000034	0.000019
Final, inner flesh	0.000154	0.000085	0.000059	0.000050
Final, pit	0.000030	0.000031	0.000031	0.000032
Final, pit, and 1/2 inner flesh	0.000107	0.000074	0.000060	0.000057
Final, pit, inner, 1/2 outer flesh	0.000334	0.000238	0.000173	0.000146
Final, entire prune	0.000485	0.000358	0.000257	0.000211
Axis lengths, ft:				
Initial, major	0.132	0.133	0.135	0.136
Initial, average minor	0.102	0.102	0.104	0.104
Final, major, of equal spheriod	0.117	0.106	0.094	0.089
Final, minor, of equal spheriod	0.089	0.081	0.072	0.068
Surface areas, sq ft:				
Initial spheroid	0.0378	0.0381	0.0390	0.0396
Final spheroid of equal volume	0.0290	0.0239	0.0189	0.0169
Dried skin, top	0.0053	0.0054	0.0056	0.0054
Dried skin, side	0.0161	0.0152	0.0172	0.0137
Dried skin, bottom	0.0051	0.0048	0.0060	0.0055
Dried skin, total	0.0265	0.0254	0.0288	0.0246
Final, top of equal spheroid	0.0058	0.0051	0.0037	0.0037
Final, side of equal spheroid	0.0176	0.0143	0.0113	0.0094
Final, bottom of equal spheroid	0.0056	0.0045	0.0039	0.0038
Depth of slice, volume/area, ft:				
Final, top	0.0086	0.0110	0.0092	0.0054
Final, side	0.0109	0.0094	0.0087	0.0095
Final, bottom	0.0105	0.0116	0.0086	0.0050
Depth ratio, depth/minor semiaxis:				
Final, top	0.19	0.27	0.25	0.16
Final, side	0.24	0.23	0.24	0.28
Final, bottom	0.24	0.29	0.24	0.15
Mean radius of sphere of equal volume, during time increment, ft:				
Inner flesh	0.029	0.028	0.026	0.024
Outer flesh	0.045	0.041	0.036	0.033
Entire prune	0.052	0.047	0.041	0.038
Mean radius ratio, fraction of entire prune mean radius:				
Inner flesh	0.56	0.60	0.62	0.64
Outer flesh	0.87	0.88	0.88	0.88

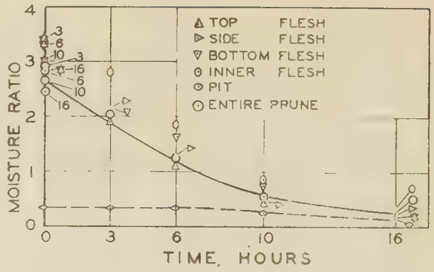

Fig. 23 Moisture Ratios of Subdivisions of Prune Flesh

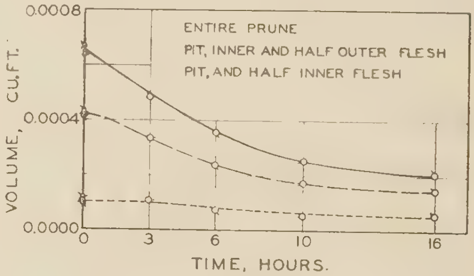

Fig. 24 Prune Volume During Drying

the average vapor concentration for each run. This was done to avoid distraction by insignificant fluctuations which would result from plotting the data directly with its short-time irregularities in air temperature and humidity. The relative humidities at the prune-air interface, calculated from vapor concentration and vapor concentration for saturation, from Fig. 21, are given in Fig. 22.

Moisture Distribution. The first moisture-distribution test was of limited value because the depth of cut was not evaluated. The second run yielded the data in Table 3, upon which Fig. 23 is based. In comparing the moisture concentrations in top, side, and bottom slices, variations in depth of cut, noted in Table 3, must be borne in mind.

To determine the depth of cut, the volume and surface area of each section was needed, the weight of dry matter and moisture content being known. The area was given directly by planimetering the dried skin, as previously described. The density of the flesh, used in finding the volume, was obtained by subtracting weight and volume values for the pit from those for the whole prune, yielding separate density data for flesh and pit, given in Figs. 17 and 18.

The sets of original data for the 3, 6, 10, and 16-hr observations were necessarily on different lots of similar prunes, which as can be seen in Table 3 varied in initial dimensions and moisture content. In order to estimate what the condition of each sample would have been at the other times, the time-volume curves in Fig. 24 were prepared.

Discussion of Results

Drying rates in these tests were a little slower than those indicated by Guillou's Equation [2] for drying in hardware cloth racks

$$B = 0.2(t/165)^4(V/600)^{0.2}\,\frac{(100 - H)}{60}\,(N/50)^{0.67} \ldots [16]$$

The mean drying coefficients for a change of moisture ratio from 2 to 0.2 lb per lb were 93, 87, and 99 per cent, respectively, of the prediction of equation for Runs 2, 3, and 4 of the first group, and 93 per cent of the predicted value for the 16-hr sample of the second group.

Temperature of Drying Prunes. Temperature differences be-

tween air and prune were nearly the same for all three runs, Fig. 13, even though the wet-bulb depressions were 32.5, 47, and 61 deg for Runs 3, 2, and 4, respectively. This result is in accordance with the fact that the drying rate is not appreciably affected by changes in relative humidity of the dehydrator air below 35 per cent. The temperature difference depends primarily upon the drying rate and the surface conductance; thus, if the drying rates and surface conductances are the same, temperature differences will also be the same.

It does not seem worth while to develop an equation giving prune temperatures directly, because, in the usual commercial dehydrator, constant drying conditions do not occur. The air-to-prune temperature difference can be estimated from the drying rate and surface conductance. Then the prune temperature can be obtained by subtracting the temperature difference from the air temperature. The surface thermal conductance per pound of dry matter can be given roughly for these runs as

$$fA_d = 10 + 4M \qquad [17]$$

At the start of drying, this can be expected to vary with the 0.8 power of the air velocity, the flow pattern being that in a channel, while near the end of drying it may vary with the 0.6 power, flow being around irregular shapes on the tray surface. It will be proportional to the prune count to the $^2/_3$ power, since it is expressed on a unit-weight basis, for which the surface is proportional to the $^2/_3$ power of the number of prunes per pound. It will be a little higher for tray loadings of less than 4 lb per sq ft, but will be materially reduced by heavier loadings which can be obtained only by piling prunes, obstructing air access to prune surface.

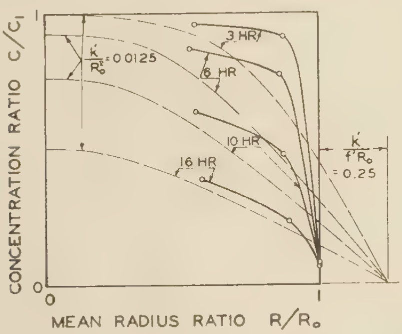

Fig. 25 Moisture Distribution During Drying

Vapor Concentration and Relative Humidity. The vapor-concentration difference between the prune surface and the air stream is small, so that it has a minor influence upon vapor concentration at the surface. The relative humidity at the prune surface is dependent primarily upon the vapor concentration in the air stream and the surface temperature of the prune. The evaporation rate is not greatly influenced by the relative humidity of the air stream until the relative humidity at the prune surface rises high enough to raise the equilibrium-moisture ratio appreciably. As shown in Table 2, the equilibrium-moisture content does not rise rapidly with relative humidity until humidities of over 50 per cent are reached. The free-moisture ratio (total- minus equilibrium-moisture ratio) was not greatly influenced by raising the air-stream relative humidity from 15 to 27 per cent, Fig. 22, but the further rise to 43 per cent reduced the free-moisture ratio materially at the start of drying.

As the end of drying is approached, the relative humidity at the prune surface drops because the prune-surface temperature approaches that of the air. A low relative humidity is important here if a low final-moisture ratio is required.

Moisture Distribution. A simple, but inadequate, ideal system with which to compare the moisture-distribution data, is that of a homogeneous sphere which dries without shrinking, with a constant moisture conductivity and a constant surface vapor conductance. The moisture distribution for this sphere can be found directly from its thermal analog. Assuming k'/R_0^2 to be constant at 0.0125 (1/hr) and $k'/f'R_0$ to be 0.25, the moisture-concentration ratios are given by the dashed curves in Fig. 25, using values for temperature distribution in a sphere from McAdams (6) or Boelter et al (5). These constants were arbitrarily selected to approximate the actual distribution at 16 hr.

From the volumetric data of Table 3, the mean radius with respect to volume was computed for the outer surface, outer layer of flesh, and inner layer of flesh, for each of the time intervals (viz., 0–3 hr, 3–6 hr, 6–10 hr, and 10–16 hr) of the moisture-distribution run. The free moisture concentration in pounds per cubic foot was then expressed as a fraction of the initial free moisture concentration, and plotted against the mean radius ratio in Fig. 25. The surface concentration was taken to be that in equilibrium with the prune-surface air-vapor mixture, at the relative-humidity values in Fig. 22. The departure of all the surface points, and the internal points at times other than 16 hr, from the ideal system is marked. The comparison is of course considerably qualified by the fact that the observed points are placed at the mean radius ratios of the sphere which was changing in volume with time. The result is similar to that noted for a slab of soap by Hougen, McCauley and Marshall (7).

The concentration gradients, represented by the observed points, indicate that the moisture conductivity within the prune decreases as the flesh becomes drier. As long as the thickness of the layer of decreased conductivity is not too great, the resistance is not greatly increased because the path is short and the cross section, nearly that of the prune surface, is large. Near the end of drying, however, the distance that moisture must travel through low-conductivity material is greater, and the path near the center is of smaller cross-sectional area, so that the drying rate tends to drop off. If prunes are to be dried only to 20 per cent moisture (16.7 per cent on a wet basis), the departure from a logarithmic curve is not pronounced, but for low terminal-moisture contents, a reduced drying rate can be expected.

Summary and Conclusions

1 The temperature difference, air stream to prune surface, dropped from 20 F at the end of the first hour, to 3 F at the end of drying, for prunes drying from a moisture content of 2.7 to 0.27 lb per lb of dry matter in 16 hr.

2 The unit surface thermal conductance rose from 7 Btu/hr ft² F at the start of drying to 8 at intermediate stages, and then dropped to 6, at an over-all air velocity of 600 fpm, measured in the free tunnel cross section.

3 Changes of air-stream relative humidity do not appreciably influence drying rates of prunes if the relative humidity is less than 35 per cent because:

(*a*) The equilibrium moisture content - relative humidity curve does not rise rapidly until relative humidities over 50 per cent are reached.

(*b*) The concentration difference between the prune surface and the air stream is small. The moisture conductivity of prune flesh is low, compared with the surface conductance, so that most of the resistance is within the prune.

(*c*) The prune-surface-temperature depression is only a fraction of the wet-bulb depression, so that the vapor concentration required for saturation at the prune-surface temperature is much higher than it would be at the wet-bulb temperature.

4 The moisture distribution within a prune during drying indicates a decrease in moisture conductivity with decrease in

moisture content, so that the differential equation for diffusion for a system with constant properties (the Poisson-Fourier equation) is not applicable.

5 Although the drying rate is nearly proportional to the free-moisture ratio in the usual drying range (down to 0.2 lb of moisture per lb of dry matter), it decreases for lower moisture contents, because of the decrease in moisture conductivity of prune flesh. Extrapolation of the evaporation rate moisture content curve beyond the usual drying range thus intersects the moisture-content axis at an abnormally high value which does not coincide with observed equilibrium-moisture-content values. This is of commercial importance only where very low final-moisture contents are required.

Acknowledgments

The author wishes to thank Coby Lorenzen, Jr., for preparing and installing the thermocouples in the prunes; B. D. Moses, Rene Guillou, and Coby Lorenzen, Jr., who helped in conducting the tests; and L. M. K. Boelter who gave stimulating suggestions on technique and on the analysis of results.

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A New Approach to the Problem of Conditioning Water for Steam Generation

By R. E. Hall,[1] Pittsburgh, Pa.

Modern boilers, because of their high operating pressures and ratings, and their high rates of heat adsorption, intensify the problems associated with the continuous operation which is of such critical importance in our vast productive program. For instance, actual consumption of the boiler tubes by the high-pressure boiler water must be continuously guarded against. Destructive boiler scale such as that composed of sodium silicate, sodium-aluminum and sodium-iron silicates, sodium sulphate and sodium phosphate, must not be permitted to form. The boilers must not fill up with sludge, and the turbines must be kept free of deposits on their intricate blading. Suspecting that sodium for a number of reasons is of destructive influence in the boiler water, the author has initiated building boiler waters on a potassium basis, with affirmative results that are set forth in this paper. Starting with substitution in the boiler water of potassium for the universal sodium equilibrium, the author has been led to give consideration to the problems of "hide-out" of boiler-water salts, corrosion by the bonded oxygen of the boiler water, silica in boiler and turbine, equilibria developed in the superheater in any carry-over of boiler water, and under certain conditions, protection from embrittlement. In systematizing the results which must ensue from the complex network of interrelations of temperature, pressure, and concentration, under these conditions, he makes use of the conceptions of the concentrating-film boiler water, generalizes the temperature-pressure-concentration relations in the form of a Δt function, and points out how the relationship of incongruent solubility between water and the silicates must be controlled.

Introduction

Some years ago, introduction of equilibrium-wise thinking into boiler-water chemistry brought into use maintenance of the phosphate equilibrium in the boiler water as the final step in its conditioning for elimination of the alkaline earth-metal scales. Inasmuch as engineers were demanding larger boiler units, higher pressures, higher ratings, and higher rates of heat absorption, but, particularly in the industrial field, felt well-nigh frustrated by the drastic restrictions (1)[2] on feedwater deemed necessary therefor, this innovation was timely in removing these restrictions, and thus contributing to the progress achieved in attainment of the objectives sought.

Scarcely were the alkaline earth-metal scales under control, however, when new difficulties arose in the form of siliceous deposits comprising silica and sodium silicate, sodium-aluminum and sodium-iron silicates. The phosphate equilibrium was not designed to, nor could it, prevent their formation. At about the same time, another phenomenon, the hiding-out of boiler-water salts in the boilers of high pressure and high rating, became well recognized. Purer water for the boiler was conceived to be an answer to this problem, but the purer boiler waters which resulted brought to light a problem of their own, namely, furrowing and channeling of tubes, with development of large amounts of magnetic iron oxide, and final failure; or, development of large quantities of magnetic iron-oxide sludge without channeling or furrowing in evidence at any special point. Since these cases of special corrosion have been attributed to dissolved oxygen entering the boiler in the feedwater, tremendous effort and expense have been brought to bear on removal of the last vestige of dissolved oxygen from the feedwater, but without remedying the condition. As a last straw, as boiler-water alkalinities have been lowered almost to the vanishing point in another move to combat such corrosion, there has arisen an epidemic of silica deposition in the lower pressure stages of turbines, not amenable to steam-washing, but requiring caustic-washing (26) or complete shutdown of the unit for cleaning. Elimination of the silica content in the boiler water would seem to be the obvious move to make; but soft boiler deposits, relatively high in magnesium content, and not removable by blowdown, then become another cause of irritation and distress.

Faced with deficiencies of circulation in hard-worked tubes that might very well function as planned if these other problems were solved, the engineers have come to the point of bluntly questioning the chemists whether further progress in conditioning boiler waters is possible.

Inauguration of Potassium Equilibrium in Boiler Water

During the summer of 1942, with the end in view of attempting to break the deadlock which existed relative to solving these different problems, we inaugurated the drastic change of maintaining the boiler water on a potassium equilibrium, in place of the sodium equilibrium which has characterized boiler waters from the beginning of steam generation. As illustrated in Table 1, boiler deposits in the evaporative sections, before the advent of phosphate equilibrium in the boiler water, consisted mainly of calcium and magnesium salts with variety provided at times by silica, sodium silicate, and magnetic oxide of iron. After the advent of phosphate equilibrium, the deposits as illustrated in Table 2, while containing minor amounts of calcium

[1] Director, Hall Laboratories, Inc. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Joint Research Committee on Boiler Feedwater Studies and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

Note: Space does not permit publication as an appendix of the tabulated data which have been used, but a complete set of these tables is on file at the Engineering Societies Library, 29 West 39th Street, New York, N. Y., for reference. The original data are available in the references of the Bibliography.

and magnesium salts, were dominated instead by sodium salts, i.e., sodium silicate, sodium-aluminum silicate, sodium-iron silicate, sodium phosphate, and sodium sulphate, along therewith, silica and magnetic oxide of iron.

It was decided to get rid of sodium in the boiler water and to substitute potassium therefor. Potassium and sodium are chemical equivalents, but are not always peers chemically, as is well illustrated by the simple examples of hard and soft soaps, and such differences as make potassium silicate a preferred binder in special flux coatings for welding rods, and a more successful vehicle than sodium silicate for pigments in some decorative paints (39). The problem to be studied was whether the substitution of potassium for sodium might demonstrate a difference at the temperatures and pressures of the boiler-water cycle which would eliminate alkali-metal deposits.

Messrs. L. E. Hankison and M. D. Baker of the West Penn Power Company expressed willingness that this experiment be put into practice at the Springdale Station of the company. It was Hankison, who at the same station had previously carried on the experimental work in practical operation for the phosphate equilibrium (2).

From the time it started, this work has continued at Springdale and, because results are remarkably definite, will continue. Meanwhile, in the early spring of 1943, with the results at Springdale as a background, we brought the system of potassium equilibrium to the attention of the engineers in a few plants which had specific problems to which the system pertained.

Fundamental Phenomena of the Boiler-Water Cycle

The troublesome problems, which have been noted, existed in mild form in the older lower-pressure boilers, but were considerably minimized by low ratings and by scale formation. The cleaner boilers that have resulted from phosphate-equilibrium treatment, along with higher pressures, temperatures, ratings, and higher rates of heat absorption, have magnified these trouble-making factors seriously.

Why?

To answer that question, we must bring clearly into focus what happens as the dilute over-all boiler water traverses those surfaces where steam is generated, where the boiler tubes are exposed on the heated side to furnace temperatures of 2000 to 2500 F, where they must rapidly absorb the furnace heats both of radiation and convection, and must transfer this heat to the water they contain at a rate such that their external surfaces will remain at 800 to 900 F, or thereabouts. In the boiler-tube raceway, unimpeded flow of boiler water and formation of myriad steam bubbles therefrom interest us most.

Whatever may be the mechanism by which the steam bubbles form, Fig. 1 will serve to illustrate, conventionally, a transient state of affairs with billions of recurrences. What happens on the tube surface A under the steam bubble? Partridge (3) a number of years ago initiated the move to answer that question when he photographed the process of calcium-sulphate scale formation, as shown in Fig. 2. In explanation, he stated: "Due to the great increase in resistance to heat flow of the gas film over the resistance of the liquid film, the metal surface tended to overheat locally, in the area within the interface." Along therewith, the film of boiler water covering the surface under the bubble concentrated or evaporated, and deposited its content of calcium sulphate in the form of scale at the periphery of the contact of the bubble with the tube surface.

May we assume therefore that wherever a steam bubble forms, or where the more severe conditions of steam-blanketing,

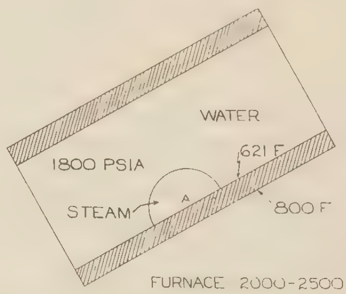

Fig. 1 Transient State of Affairs With Billions of Recurrences

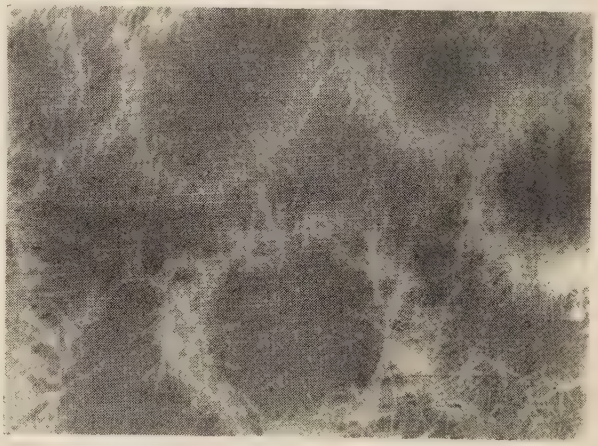

Fig. 2 Mechanism of Formation of Calcium-Sulphate Scale

TABLE 1 EXAMPLES OF BOILER (EVAPORATIVE SECTION) SCALES WITHOUT PHOSPHATE TREATMENT OF BOILER WATER

	Per cent 1	2	3	4
Calcium sulphate	74.8	3.4	1.2	33.3
Calcium carbonate	3.9	31.8	4.8	0.5
Calcium silicate	2.7	..	61.5	..
Magnesium carbonate	..	1.3	..	..
Magnesium silicate	11.5	28.7	4.1	4.6
Magnesium hydroxide	4.3	21.6	..	1.4
Magnetic iron oxide	0.4	7.5	6.7	54.8
Aluminum oxide	..	..	..	0.7
Moisture and organic	1.5	7.2	5.7	5.4
Sodium silicate	..	..	13.2	..
Silica	..	..	5.3	..

TABLE 2 EXAMPLES OF BOILER (EVAPORATIVE SECTION) SCALES WITH PHOSPHATE EQUILIBRIUM MAINTAINED IN BOILER WATER

	Per cent 1	2	3	4	5	6	7	8
Calcium phosphate	0.9	8.7	6.9	16.5	10.1	3.6	23.5	2.8
Magnesium phosphate	0.4	5.2	3.0	6.6	1.9	1.1	1.8	..
Magnesium silicate	..	..	4.6	..	2.8	2.5	0.9	..
Moisture and organic	0.8	3.0	0.7	3.3	5.7	8.6	2.4	..
Magnetic iron oxide	1.9	13.9	19.6	19.2	..	1.7	47.1	97.8
Copper	..	6.8	..	7.2	..	..	20.6	..
Sodium sulphate	..	5.0	8.5	..	..	..	0.9	..
Sodium phosphate	0.2	11.5	20.0	0.3	..	..	0.3	..
Sodium silicate	3.3	0.8	..	21.6	0.6	..	..	..
Sodium-aluminum silicate	86.7	45.0	37.6	24.2	35.0	..	..	..
Sodium-iron silicate	..	..	..	..	28.9	64.9	..	..
Silica	5.1	..	..	..	16.5	19.9	..	..

or film-boiling, as noted by Davidson and Associates (4, 5) occur, the film of boiler water on the surface will evaporate to dryness,[3] or so concentrate as to deposit its salt in solid form? To answer that question, let us consider the experiment illustrated in Fig. 3. In the flask, with its vapor-containing wall carefully insulated against loss of heat, a dilute solution of calcium chloride is boiling, at atmospheric pressure, freely discharging its steam to atmosphere through the side arm of the flask. The boiling temperature t_l, of the solution equals t_v, the temperature of the carefully heat-insulated vapor, which, however, issuing from the side arm immediately drops in temperature to 212 F. The dif-

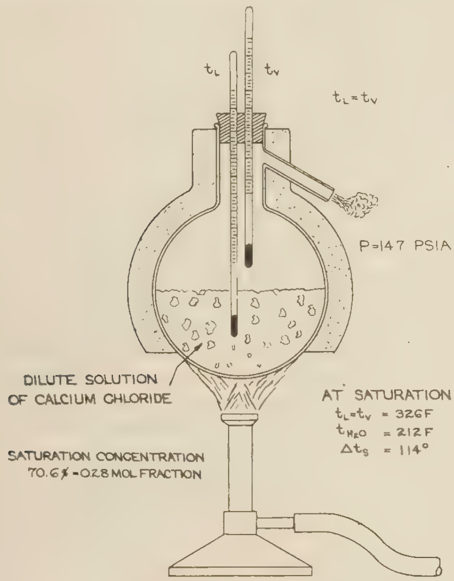

Fig. 3 Simple Illustration of Δt Function

ference between the temperature t_v of the steam in the flask and the temperature tH_2O of steam developed by pure water boiling at the same pressure is Δt. Concentration of the solution, constantly increasing because of loss of steam through the side arm, finally reaches 70.6 per cent or 0.28 mol fraction[4] which, at atmospheric pressure or 14.7 psia, is saturation concentration. The equal temperatures of solution and vapor, t_l and t_v, have likewise increased with the increasing concentration, and at saturation are designated t_s, and Δt becomes Δt_s. For the condition of saturation, t_s = 326 F, tH_2O = 212, and the steam in the flask is superheated 114 deg ($\Delta t_s = t_s - tH_2O$). The initial concentration of the calcium chloride, whether a few parts per million, or a few per cent, is a matter of indifference. If the pressure is fixed at 14.7 psia, and heat is supplied at the requisite level, the conditions for saturation as noted will eventually obtain. With continuation of boiling and discharge of steam, t_s and Δt_s will not increase, but instead, separation of solid salt will begin and continue until only solid calcium chloride remains. On the other hand, if the heat level is not sufficient to raise Δt to Δt_s, boiling will cease and equilibrium will obtain at the maximum Δt corresponding to the heat level in question, saturation never being reached unless the heat level is increased to raise Δt to Δt_s.

In the experiment, pressure is fixed at atmospheric, and if the flask contained pure water, boiling would occur at 212 F (tH_2O). With solution in the flask, however, boiling does not occur at 212 F, a fact which signifies that the vapor pressure of the solution is less than that of pure water. In general, at any temperature t, decrease in vapor pressure of any solution becomes greater as its concentration increases, and thus is a function of the concentration of the solution, or equally, of the number of solute molecules in the solution. It was Raoult who pointed out that in dilute solutions of nonelectrolytes, the function is closely that of strict proportionality (A, II).[5]

When the temperature of a solution of any given concentration is increased, its vapor pressure is thereby increased. In the experiment, with pressure fixed at atmospheric, when the temperature t_l of the solution was raised sufficiently above 212 F, the increment of temperature (Δt) over 212 F sufficed to increase the vapor pressure of the solution to atmospheric, and boiling then occurred. As evaporation proceeded, and concentration of the solution accordingly increased, the increment of temperature Δt over 212 F, required to occasion boiling, likewise increased. The relationship thus established for pressure of 14.7 psia and tH_2O = 212 F, is applicable as well to all boiler pressures and the corresponding boiling temperatures or tH_2O values. It is thus obvious that Δt is a function of the concentration of the solution or, equally, of the number of solute molecules in the solution.

In the light of these conclusions, let us examine what happens to a film of boiler water on surface A, Fig. 1, if the temperature of the surface becomes heated above boiler-water temperature, 621 F, by an amount Δt. Pressure is fixed at 1800 psia. As the film begins to boil and concentrate at this fixed pressure, because of increased temperature Δt, its vapor pressure must attempt to decrease because its number of solute molecules is increasing. This attempt, however, is offset or compensated by the tendency of the vapor pressure of the solution to increase with rise in temperature. Meanwhile concentration of the film proceeds in its effort to establish the relationship among the different forces which represents equilibrium for the particular Δt value assumed. As concentration thus proceeds, some components of the solution may reach saturation, and be deposited, while others, more soluble, carry on the work of building the concentration to the degree required for equilibrium. If the Δt assumed happens to be greater than the Δt_s of saturation concentration of all components in the solution, evaporation to complete dryness must occur. This is what happened in the development of scale illustrated in Fig. 2, since calcium sulphate at saturation has a low value of Δt_s. But if the Δt_s value of some components at saturation is greater than the Δt assumed, they will remain dissolved and merely increase in concentration by the amount necessary to establish equilibrium.

These considerations illustrate the distinction between the customary conditions obtaining in measurements of vapor-pressure lowering and boiling-point rise, and those which obtain at the steam-water interfaces in a boiler. In the first, temperature and concentration are usually the independent variables with vapor pressure the dependent variable; in the second, vapor pressure and concentration are independent variables, temperature the dependent variable. But in the boiler, pressure, fixed by the boiler, and temperature, established by the degree of overheating Δt at a surface such as A, Fig. 1, are the independent variables, and the concentration of the concentrating-film boiler water is the dependent variable. Its value is commonly expressed as a mol fraction, and frequently indicated by X_2.

We shall designate as the Δt function relations of this type, occurring at the steam-water interfaces of the boiler-water and

[3] W. Otte has assumed evaporation to dryness in all cases (49), (50).

[4] The mol fraction of any constituent in any solution is the number of mols of that constituent divided by the total number of mols in the solution.

[5] References to the Appendix are thus indicated throughout the paper. See footnote on page 457.

steam cycle. Whenever we shall refer to the Δt function, we shall thereby signify the equilibrium that will be reached or approached on the basis of the variables pressure, temperature, and concentration, operating in accordance with the rules heretofore set down. Under the dynamic conditions that exist at a surface such as A, Fig. 1, or even areas where steam-blanketing or film-boiling exists, any well-defined equilibrium for any certain set of conditions may not be reached because of insufficiency of time. The film on surface A will continually struggle to establish equilibrium, however, and in so doing will at all times follow the pattern laid down by the principles of the Δt function.

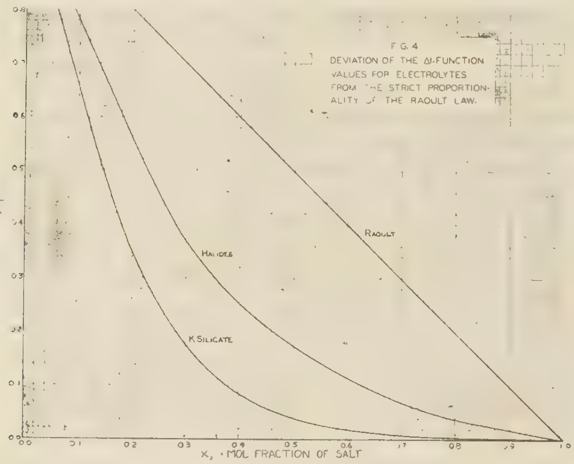

FIG. 4 DEVIATION OF THE Δt-FUNCTION VALUES FOR ELECTROLYTES FROM STRICT PROPORTIONALITY OF THE RAOULT LAW

This will be illustrated by examples. In the experiment, Fig. 3, if sodium chloride, potassium chloride, or sodium sulphate, respectively, were used in place of calcium chloride, the results would be as follows (A, III, 4, 5, 19; IV, 3, 4, 12):

Pressure = 14.7 psia	Saturation concentration X_2 = mol fraction	t_s deg F	Δt_s deg
Sodium chloride	0.109	228	16
Potassium chloride	0.123	227.5	15.5
Sodium sulphate	0.051	217.1	5.1

If the pressure were 1800 psia, as in Fig. 1, on the surface A under the steam bubble, the results would be thus:

Pressure = 1800 psia	Saturation concentration X_2 = mol fraction	t_s deg F	Δt_s deg
Sodium chloride	0.190	692	71
Potassium chloride	0.292	745	124
Sodium sulphate	0.0095	625	4

This means that if the solution of potassium chloride were evaporating on the surface A, Fig. 1, the overheating of that surface on the inner side of the tube to produce saturation in the solution would necessarily be 124 deg. For the sodium-chloride solution, overheating to the extent of 71 deg, and for that of sodium sulphate, only 4 deg, would produce saturation. If for instance the overheating at A were 35 deg, the solution of neither potassium nor sodium chloride would attain saturation, but that of sodium sulphate would evaporate to dryness.

In the generally strong solutions of electrolytes which result from the concentration of boiler water, the curves plotted on the basis of the Δt function variables provide a valuable guide in the study of these solutions. Fig. 4 illustrates the deviation of the halides (sodium and potassium chloride, bromide, and iodide) (6) and of the potassium silicates (7) in saturated solution, from the strict proportionality of Raoult's law. (P_s and P_{H_2O} are, respectively, the vapor pressures of the saturated solution and of pure water at t_s). Later sections will develop the use of these curves.

Suppose that the solution had contained two species of molecules, for instance, sodium sulphate and sodium chloride: According to the Δt function, the value of Δt is a function of the number of solute molecules present, and largely independent of their kind. Thus, in this mixture, both the sodium chloride and the sodium sulphate will each exert its influence toward increasing Δt, hence with less concentration of either one for a given Δt than if either were in solo solution. At the pressure of 1800 psia under consideration, the data of Schroeder, Gabriel, and Partridge (8), (A, VII, 2) show that with increasing concentration of sodium chloride, no marked increase occurs in the solubility of sodium sulphate. Hence if a solution containing these two components evaporated on surface A, Fig. 1, overheating of the surface of somewhat more than 4 deg, depending upon their relative concentrations, would cause saturation concentration of the sodium sulphate, and it would begin to precipitate as solid phase on the surface, while the sodium chloride would remain in solution and keep on concentrating, requiring a considerably higher temperature to effect its saturation. If in place of sodium sulphate, very soluble sodium or potassium nitrate (A, III, 10, 11; IV, 9) were the other component of the solution, the degree of overheating of the surface A which would be necessary in order to cause saturation concentration of either component would be largely increased over what it would be for either one. The significance of this is that once sufficient data have been acquired in relation to the Δt-function properties of the boiler-water salts, it will be possible to predetermine for any value of Δt, whether a boiler water upon concentration will finally become saturated with respect to one or all components, and by appropriate adjustment of their relative concentrations, to impose on the solution the desired condition of these components as Δt is reached.

Thus, in Fig. 1, we can readily picture what will happen at the surface A, if the boiler water contains constituents of limited solubility like sodium sulphate along with some caustic soda. As the temperature at A rises by some amount Δt, the film of boiler water must concentrate in accordance with its Δt-function value. Sodium sulphate will precipitate, and the caustic will concentrate. Deposition of sodium sulphate or attack of the surface by the concentrated caustic, either or both, will create resistance to the flow of boiler water. If, however, we condition the boiler water so that quite soluble salts are present to concentrate along with the caustic, then since the attainment of Δt rests mainly upon the number of molecules in solution, and not largely on their species, neither the caustic nor the soluble salts will concentrate excessively in developing the equilibrium corresponding to the rise in temperature Δt on the tube surface. By this preclusion of development in the boiler tubes of impediments to flow, maximum flow of boiler water is assured. Therewith the probability of steam-blanketing or film-boiling conditions is lessened, and the life span of the bubble, Fig. 1, and also the Δt value of the A surface are kept at a minimum.

Thus from this careful examination of events contingent upon bubble formation, or more broadly, occurring at interfaces of boiler water and steam, there evolves a pattern for molding them to the best advantage of boiler operation. Cognizance must be taken of the fact that in any boiler, there are in reality two different boiler waters to consider, namely, the dilute over-all and the concentrating-film boiler waters. In the boiler, wherever the dilute over-all boiler water governs conditions, we need consider only its isobaric-isothermal properties; but where concentration occurs, as in the concentrating-film boiler water under

the steam bubbles which are forming, or at points of steam-blanketing or film-boiling, we must give heed to its isobaric-polythermal properties as it concentrates in accordance with the Δt function. When any carry-over of the dilute boiler water enters the superheater, it is subjected to isobaric-polythermal conditions, and it will concentrate therein in accordance with the pressure and temperature relations which obtain. The concentrated magma thus formed upon passing into the turbine meets the falling pressure and temperature which characterize conditions therein, and we must consider its properties in relation to the polybaric-polythermal conditions to which it is subjected. In Table 3, these facts are briefly summarized.

TABLE 3 PRESSURE-TEMPERATURE RELATIONS AFFECTING BOILER WATER AND CARRY-OVER

	Isobaric isothermal Over-all boiler water	
Isobaric polythermal		Polybaric polythermal
A *Boiler:* Concentrating-film boiler water		*Turbine:* Gradual dilution of concentrated carry-over
B *Superheater:* Highly concentrated boiler-water carry-over		

Particularly, Table 3 emphasizes the need of a systematized procedure to take cognizance of the changes occurring in the isobaric-polythermal regions of the boiler or superheater, and the polybaric-polythermal region of the turbine, in addition to that which has been generally applied to the dilute over-all boiler water. Such systematic procedure means so balancing the proportional relationship and character of the components in the dilute over-all boiler water, which can be readily established as desired and checked by analysis, that changes of concentration occurring at interfaces of boiler water and vapor in accordance with the Δt function cannot result in development of high causticity, or in the separation of water-insoluble components, especially those siliceous in nature.

In order to arrive at the features which must be embodied in this inclusive procedure, we shall now subject the problems enumerated in the introduction, and, incidentally, factors relating to protection from caustic embrittlement also, to examination in the light of the potassium equilibrium and the application of the Δt function.

The Problems of Hide-Out of Boiler-Water Salts

The boiler-water salts which have been mainly recognized as guilty of hide-out are sodium sulphate and sodium phosphate. Some recent data, however, in which the silica concentration was followed closely, have shown that sodium silicate is equally guilty with the phosphate and sulphate.

In the range of boiler-water temperatures the solubilities of sodium phosphate and sodium sulphate (A, III, 13, 19) decrease rapidly with temperature increase. The fact that sodium silicate joins with the sodium sulphate and phosphate as a hide-out salt ties in well with the very limited solubility data on sodium silicate which are available (A, III, 15) and which show it to have a decreasing solubility curve in the range of boiler-water temperature. From the customary thinking in terms of retrograde solubility alone, however, it is difficult to see how hide-out would occur, since the sulphate and phosphate solubilities are a couple of thousand parts per million or more at operating pressures well above those in which hide-out has been characteristic. Neither is it likely that the double salts of sodium sulphate and sodium phosphate, which were recognized and discussed by Schroeder, Berk, and Gabriel (9), (A, VII, 7), are the reason for the hide-out of these two salts, as the solubilities of these double salts are appreciable.

In the illustration of overheating of surface *A*, Fig. 1, we must remember that Δt_s governs the number of molecules necessary for saturation and hence deposition, and that the apparently considerable solubility of sodium sulphate as viewed from the standpoint of parts per million shrinks to a relatively insignificant value when expressed as mol fraction. Reference to the values of Δt_s for sodium sulphate (A, IV, 12), over the range of boiler temperatures, discloses a maximum of 8 deg only and at the temperature of 600 F, of no more than 4. This means that if the water surface of a boiler tube is heated only a very few degrees above boiler-water temperature, where rapid evaporation is occurring or where there is a tendency to film-boiling or steam-blanketing, sodium sulphate will precipitate readily enough. Low Δt_s values are a characteristic of salts whose saturation concentration is given by small value of the mol fraction or, in other words, by a relatively small number of molecules in the solution. Thus sodium carbonate (A, IV, 2) is similar to sodium sulphate, its maximum Δt_s being 10 deg.

Sodium carbonate does not appear as one of the hide-out salts, presumably because of its ready decomposition into caustic soda, and because so little of it is present in the boiler waters at even medium boiler pressure. Reference to the solubility tables of sodium phosphate and sodium silicate discloses that their saturation concentrations are given by extremely small mol fractions, so that while their vapor pressures are unknown, and therefore likewise their values for Δt_s, yet doubtless these are very small and probably lower than Δt_s for sodium sulphate.

The question might be raised if the increased solubility of sodium sulphate in the presence of caustic soda at higher temperatures and pressures as found by Schroeder, Gabriel, and Partridge (8), (A, VII, 1), would not tend to minimize the hide-out tendency of sodium sulphate by considerably increasing its Δt_s value. While no values are available for the vapor pressure of such solutions, the mol fraction of sodium sulphate at saturation concentration is small even with its increased solubility in the caustic-soda solution, and the inference therefore is that deposition of sodium sulphate would occur with relatively low Δt_s. According to the work of Schroeder, Berk, and Gabriel (9), (A, VII, 7, 8), caustic soda has little influence on the solubility of sodium phosphate, and hence on the value of Δt_s, at which deposition would occur.

Plainly, the elimination of hide-out demands that the components in the boiler water shall be of much higher Δt_s than that characteristic of sodium sulphate. This viewpoint is supported by the high values of Δt_s for sodium hydroxide and sodium chloride (A, IV, 3, 5), since neither of these salts is guilty of hide-out propensities.

At one of the stations on which we initiated potassium-equilibrium conditioning during the spring of 1943, but while sodium-conditioning was still in effect, measurements on hide-out had shown the conditions to be as bad as, and probably worse than, any we had previously experienced. Both the sulphate and the phosphate skyrocketed when boiler load was reduced. With potassium-equilibrium treatment well established, successive tests have shown complete absence of sulphate hide-out, and elimination of all but the barest trace of phosphate hide-out. Why should this be the case? So little is known of the solubility of potassium phosphate (A, III, 14), that no help in answering this question is to be gotten from it. The solubility of potassium sulphate (A, III, 20) is of low mol fraction, however, about double that of sodium sulphate. Thus while the vapor pressure of saturated potassium-sulphate solutions at the temperature in question (625 F) is unknown, the Δt_s value is doubtless a little higher than that of sodium sulphate. But unless small differences in Δt_s values are of very critical influence in the occurrence or nonoccurrence of hide-out, it seems improbable that the only slightly larger Δt_s of potassium sulphate in itself could have been the major reason for its disappearance in this case. Comparison of potassium chloride and sodium chloride (A, III, 4, 5; IV,

TABLE 4 COMPARISON OF Δt_s FOR SATURATED SOLUTIONS OF K_2SiO_3 AND NA_2SO_4

(Na_2SiO_3 is similar to Na_2SO_4)

Saturation concentration Mol fraction	Per cent	Boiling temperature at P_s t_s	$t_s - t_{H_2O}$	Vapor pressure P_s at t_s, psia	Δt_s $t_s - t_{H_2O}$
		K_2SiO_3			
0.30	78.7	446	264	38	182
0.35	82.3	542	294	62.4	248
0.40	85.0	621	311	79.4	310
0.50	89.6	815	366	164	449
0.60	92.8	995	360	153	635
		Na_2SO_4			
0.058	32.7	464	456	450	8
0.046	27.3	527	520	813	7
0.012	8.7	617	614	1710	3
0.003	2.3	662	661	2380	1
>0.0	>0.0	705	705—	3206—	>0

3, 4), for example, at boiler pressure of 1400 psia (Table 5) shows that saturation concentration for the former occurs at mol fraction of 0.274, and Δt_s of 104 deg; for the latter, at mol fraction of 0.182, and Δt_s of 65 deg. These values, while favoring potassium chloride, are high enough in either case to dispel any thought of their activity in hide-out. For either potassium metasicilate or disilicate (A, III, 17, 18; IV, 10, 11), Δt_s is extremely high, and the solubilities likewise are high, in very decided contrast to sodium silicate. What we consider to be the most feasible reason for disappearance of the hide-out is our establishment in the boiler water of an environment of soluble salts, which, together with the gain from the higher Δt_s value of potassium sulphate and perhaps phosphate, effected the result.

While all the data that we would like to have for their bearing on this problem of hide-out are not of record, yet enough are available to make it feasible to conclude that elimination of hide-out requires a Δt_s value for dissolved components in the boiler water which, as concentration occurs at any steam-water interface, is considerably greater than that of sodium sulphate. With sodium equilibrium maintained in the boiler water, this means that the concentrations of salts with low Δt_s value such as sodium sulphate, phosphate, and silicate must be carefully controlled and held as low as possible. On the basis of very limited observation, it appears that the substitution in the boiler water of potassium equilibrium for the customary sodium equilibrium increases markedly the permissible tolerance of sulphates, phosphates, and silicates. Because the methods used in obtaining the solubility data for potassium sulphate are not beyond question, and because solubility data on potassium phosphate are lacking, we are engaged at the present time in the determination of their solubilities at boiler-water temperatures. As more data of the kind become available, it will be possible to bring the details of this problem into sharper focus.

The Problem of Magnesium Deposits

An excellent example of this problem is a seaboard plant in which condenser leakage of salt water supplies magnesium to the boiler without simultaneously supplying sufficient silica to combine with it as serpentine or other magnesium silicate. A soft deposit, definitely characterized by magnesium phosphate, or at times, hydroxide, develops in the boiler. Though not forming hard scale, it clings to the surfaces and resists removal by blowdown. A similar condition may arise with the use as feedwater of a natural or treated water containing excess of magnesium over silica.

Magnesium silicate, which is the most, or at least one of the

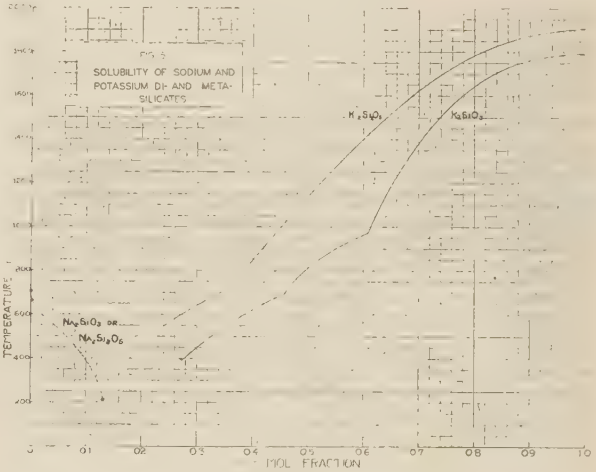

Fig. 5 Solubility of Sodium and Potassium Di- and Metasilicates

TABLE 5 CONDITIONS IN ISOBARIC-POLYTHERMAL REGION OF SUPERHEATER

Boiler pressure, psia	1400
Temperature of boiler water and saturated steam, deg F	587
Temperature at superheater outlet, deg F	1000
Degrees superheat	413

Properties of some boiler-water and other salt solutions under these conditions

Salt solution	Saturation temperature, deg F	Δt_s, deg	Saturation concentration, per cent	Condition at 1000 F
Sodium sulphate	592	5	15	Complete evaporation
Sodium phosphate[a]	Similar to sodium sulphate			Complete evaporation
Sodium carbonate	594	7	6	Complete evaporation
Sodium di- or metasilicate[a]	Similar to sodium sulphate			Complete evaporation
Sodium chloride	652	65	42	Complete evaporation
Sodium bromide	753	166	74	Complete evaporation
Potassium chloride	691	104	61	Complete evaporation
Potassium chloride plus sodium bromide	874	287	84.6 (est.)	Complete evaporation
		Δt		
Potassium chloride plus sodium bromide	984	413[c]	Unsaturated in solution	
Potassium disilicate	[b]	413[c]	Unsaturated in solution	
Potassium metasilicate	[b]	413	Unsaturated in solution	
Sodium hydroxide	[b]	413	93[d] in solution	
Potassium hydroxide	[a]	Similar to sodium hydroxide		

[a] Vapor-pressure data not available.
[b] Saturation pressure never attains boiler pressure.
[c] Saturation pressure less than boiler pressure.
[d] Concentration of the unsaturated solution.

most insoluble of the silicates, forms in the boiler water as a sludge which remains dispersed therein and is removed from the boiler by blowdown. Very likely its high insolubility plays a major role in this, resulting in its complete precipitation in the body of the dilute over-all boiler water, and thus its avoidance of taking any part in the reactions which characterize the concentrating-film boiler water.

The fact stands out that if any magnesium is to gain entry into the boiler water, provision should be made therein of sufficient silica to insure its combination with the magnesium as one of the magnesium silicates, thus avoiding magnesium phosphate or hydroxide.

Add silica for the purpose if necessary? Yes, why not? To be sure, the finger of suspicion has been leveled at silica universally, and its constant companion, sodium, has remained unsuspected and been accepted as a matter of course. The case is one of mistaken identity, and the truth is that if the influence of sodium over silica is replaced by that of potassium—if the boiler water is maintained on the potassium equilibrium—silica, in the amount required, can be accepted safely as a member of the boiler-water salts, and be of real use. The logic of this conclusion is foreshadowed by the diametrically opposite solubility properties of the sodium and potassium silicates in the range of boiler-water temperatures (A, III, 15, 16, 17, 18), as demonstrated by the curves in Fig. 5, the former being of the same retrograde type as sodium sulphate and phosphate, and the latter being of the type of sodium and potassium chloride. As regards relationships in the turbine, further facts will develop as we proceed.

The Iron-Oxide Sludge Problem: Corrosion by the Bonded Oxygen of the Boiler Water

In any plant, with the excellent deaeration characteristic of present-day operation, dissolved oxygen entering the boiler in the feedwater is of the order of a millionth of a per cent, while the oxygen it contains, bonded to hydrogen, constitutes 88.9

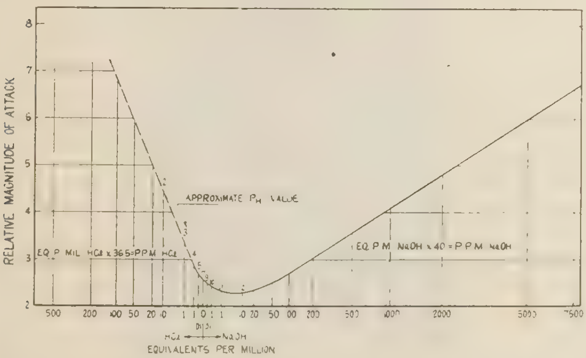

Fig. 6 Attack on Steel at 590 F by Water of Varying Degrees of Acidity and Alkalinity

per cent of its weight. If this bonded oxygen is in some way unleashed in quantity, the possibilities of damage to the boiler are better imagined than experienced. The pertinent questions are: May the bonded oxygen be readily unleashed, and if so, what measures may be taken to prevent it?

In answering the first question, we may well start with the curve in Fig. 6, developed by Partridge (10) on the basis of data obtained by Berl and van Taack (11). As demonstrated by the curve, susceptibility of boiler steel to dissolution in the contacting water increases with increase either of acidity or alkalinity. In conformity with the teaching of this curve, the over-all boiler water is in general held in the pH range of 8.3 to 12, the former being that naturally assumed by water boiled in an iron vessel (12). In this pH range the steel is only slightly susceptible to dissolution by the boiler water. Inasmuch as boiler waters are thus maintained alkaline, we shall confine our attention to the alkali side of the curve. Increasing of alkali concentration results in greater relative magnitude of attack on the steel, a result in conformity with the reaction involved

$$FeO + 2HONa = Fe(ONa)_2 + H_2O \ldots\ldots\ldots\ldots [1]$$

Thereby the steel surface is deprived of its protective film of iron oxide and renews it by reacting with water thus

$$Fe + H_2O = FeO + H_2 \ldots\ldots\ldots\ldots\ldots [2]$$

or, finally, probably, in the boiler with formation of magnetic oxide, Fe_3O_4 (10, 28). Simply, then, as recorded in Reaction [2], the bonded oxygen of the boiler water is unleashed from its hydrogen and combines with the boiler steel, thus providing the protective film so necessary therefor, and preventing or greatly retarding any further release of bonded oxygen, so long as the film remains intact. It is the entry of Reaction [1], and more specifically, the degree of alkali concentration in the reaction, which, in the end, destroys the film, and thereupon, with infinite recurrence of both Reactions [1] and [2], takes its toll of the boiler metal. It is the bonded oxygen of the boiler water which, uniting with the boiler steel, gives rise to the heavy deposits or sludges of magnetic iron oxide, sometimes with destructive furrowing and channeling, or more localized severe pitting; sometimes with merely general action on the surfaces as attested by the magnetic-oxide deposits.

Briefly, the problem of preventing this action involves the features of having alkalinity as required in the over-all boiler water, but at the same time making provision that in the concentrating-film boiler water, the concentration of alkali will be sufficiently restricted to nullify Reaction [1].

To get some idea of the alkali concentration at which Reaction [1] is sufficiently rapid to be damaging, Kaufman and Marcy in our laboratory, rotated 5 per cent solutions of caustic soda and potassium hydroxide in bombs of boiler-steel construction at approximately 470 F for a number of hours [see also (32)]. The solutions put into the bombs were clear and free of precipitate. In case the bombs were not pickled before the experiment, the solutions removed were heavily contaminated with iron-oxide sludge, presumably developed from the iron-oxide coating of the bomb walls by the mechanism of Reaction [1]. When sodium sulphite was added in sufficient quantity to account for any dissolved oxygen, similar results ensued. However, if the bombs were pickled just prior to the experiment, thus largely removing their iron-oxide coating, with or without sodium sulphite, very little iron-oxide sludge was apparent in the solution after a number of hours at the elevated temperature of the experiment. In other words, while the 5 per cent concentration of caustic was sufficient to carry through Reaction [1] with considerable speed, Reaction [2] was too sluggish to be damaging under the conditions and in the duration of the experiment. Longer duration would doubtless have occasioned formation of the iron-oxide sludge in larger amount, without segregation of the attack in any one spot. We believe these conditions parallel those in boilers which, on inspection, are found with considerable magnetic-oxide sludge distributed pretty well throughout the boiler, usually with some marked accumulation in drums or headers, but without evidence of intensified attack in any particular areas.

Are there, however, factors in the boiler which will speed up Reaction [2]? Further experiments by Kaufman and Marcy tested the effect of increasing the caustic concentration. At the same temperature previously noted, they found a 10 per cent caustic solution to attack the surface of the bomb very little in-

deed, if it was pickled prior to the experiment; when a 20 per cent solution of caustic was used, the attack, while not heavy, nevertheless was considerable, as indicated by the amount of magnetic-oxide sludge produced. In a parallel experiment with 10 per cent caustic solution, but with the surface of the bomb not pickled prior to the experiment, the attack produced a heavy sludge of magnetic oxide. As illustrated in Fig. 6, Berl and van Taack at 590 F similarly found magnitude of attack to increase with increased caustic concentration. Thus one very decided

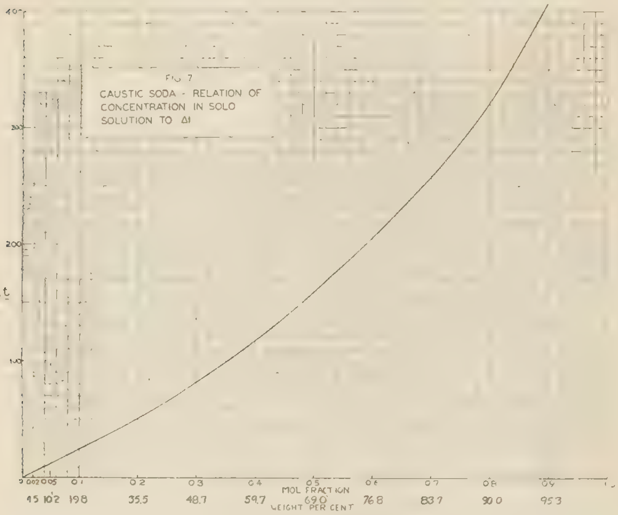

FIG. 7 CAUSTIC-SODA RELATION OF CONCENTRATION IN SOLO SOLUTION TO Δt

factor in speeding up Reaction [2] (and Reaction [1] as well) is an increase in caustic concentration. Another factor is increase of temperature, not illustrated by the foregoing experiments at fixed temperature, but undoubtedly of big influence (29).

With these factors in mind, let us consider what may happen in the concentrating-film boiler water, as formed for example on surface A, Fig. 1, under the steam bubble, where simultaneous increase both of temperature and concentration is brought to bear on a restricted area. Fig. 7 (A, V, VI) shows the result. If Δt is only 5 deg, a solo solution of caustic soda will attain concentration of approximately 4.5 per cent at equilibrium; if 50 deg, the caustic concentration will rise to 35.5 per cent; and to greater concentration if Δt is larger. At any steam-water interface, therefore, such as A, Fig. 1, and more markedly wherever steam-blanketing or film-boiling may occur, the factors of temperature and caustic concentration, increasing in unison, exercise on a restricted area their combined power of acceleration on both Reactions [1] and [2].[6]

[6] For illustrations of bonded-oxygen corrosion, see reference (10), both items. We believe Fig. 7(a) is an apt illustration of this type of corrosion, but with special circumstances leading to its development. As evident from the illustration, discrete pieces of deposit have been randomly stacked. Composed mainly of calcium phosphate and a little calcium carbonate, silica negligible, they were almost certainly formed in a part of the boiler remote from the evaporative section where they were found, and were transported in the circulating boiler water until an eddy or backwater resulted in their being gathered together as shown. Their relation to bonded-oxygen corrosion lies in their being firmly cemented together by a matrix of magnetic oxide of iron (little copper is present). The eddy or backwater indicates interruption of maximum boiler-water flow, a condition which favors steam-blanketing or film-boiling. The further impetus given to this condition by the overburden of stacked pieces of deposit quite evidently resulted in a concentrating-film boiler water of sufficient concentration to accelerate Reactions [1] and [2], and thus provide the magnetic-oxide matrix.

However vicious may be the attack on the boiler metal by the combined power of these two factors, it is made still more devastating by another factor. As shown by the vapor-pressure data of solutions of caustic soda (A, III, 6; IV, 5; V; VI), and the specific illustration of 1400 psia and 1000 F of Table 5, the concentrating-film boiler water will never go to dryness if it contains caustic soda (or, for that matter, caustic potash). On the contrary, the solution of caustic, instead of shriveling with increasing Δt on the surface, will spread instead in accord with its well-recognized characteristic of so doing (34), and thus broaden the area of attack. At the same time, the attack according to Reaction [2] by the water, held in liquid form frequently well above critical temperature of 705 F, will be maximum since the contacted surface cannot build a protective oxide covering because of the rapid dissolution thereof, according to Reaction [1] by the high caustic concentration of the contacting concentrating-film boiler water.

All factors considered, it is not surprising that attack on the metal is very rapid and devastating. Results thereof have been described by Drewry (30) and others (10, 31).

At such surfaces, concentration of caustic to the degree at which embrittlement readily occurs requires only a moderate value of Δt.

How, then, shall required alkalinity be provided in the over-all boiler water, and yet be adequately restricted in concentration in the concentrating-film boiler water?

One method consists in providing in the over-all boiler water a ratio of potassium or sodium chloride to alkalinity, such that as

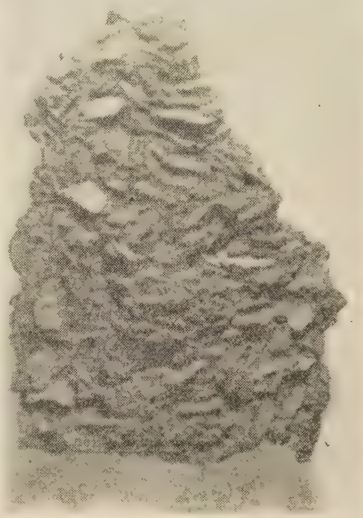

FIG. 7(a) CORROSION BY BONDED OXYGEN

concentration in accordance with the Δt function occurs as required by the Δt of any steam-blanketed surface, the alkalinity will remain of small amount and innocuous. For instance, if the boiler water contains 1 molecule of alkalinity and 4 molecules of potassium chloride, then the concentration of caustic that will result will be only one fifth of what it would be if the potassium chloride were not present. In Reaction [2], the intensity factor of liquid water in contact with the metal surface will remain, but opposing this, by strict limitation of the caustic concentration, Reaction [1] can be held in check, and the iron-oxide film on the metal, not being dissolved off so rapidly, has the chance to be protective to the metal, at least in considerable degree. It is an

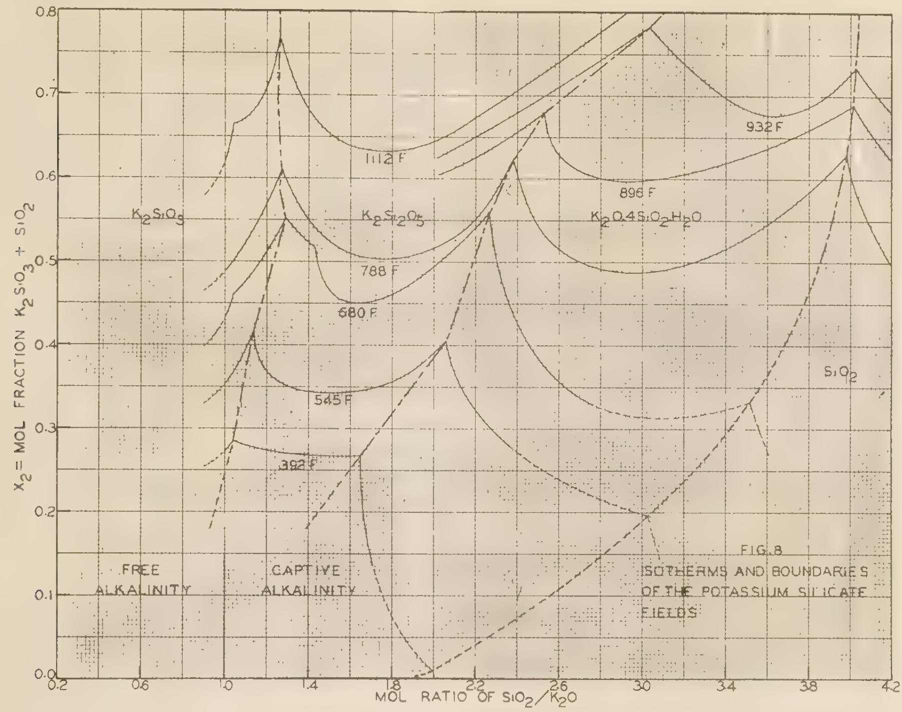

FIG. 8 ISOTHERMS AND BOUNDARIES OF POTASSIUM–SILICATE FIELDS

interesting point that as a boiler water is purer, so is it more aggressive in the development of iron-oxide sludge, unless its alkalinity is practically nil. Further, it is evident that sodium sulphate can be only slightly effective in preventing concentration of alkalinity, because sodium sulphate reaches saturation concentration at very low value of Δt, thereupon merely precipitating, and leaving it up to the caustic molecule to provide a great share of the concentration demanded by Δt, the very thing that must be avoided.

Another method of approach is to have all boiler-water alkalinity that of hydrolysis (13, 35), so that when the over-all boiler water, concentrating at a surface, becomes concentrating-film boiler water, the hydrolytic alkalinity will return to the parent molecule and not accumulate excessively. The alkalinity of hydrolysis we term "captive alkalinity," as opposed to free alkalinity, which will concentrate highly at a surface. Such captive alkalinity in the boiler water can be given by the hydrolysis of the phosphates and, of especial interest to us, by that of di- and metasilicates. The hydrolysis of the phosphates is not of so much value for this use, because they, or at least the sodium phosphate, follow sodium sulphate in their solubility relationships. On the other hand, the potassium di- and metasilicates are very soluble, keep on increasing in solubility with temperature increase and therefore are particularly valuable in control of the boiler-water alkalinity in the form of captive alkalinity. At the present time, to take care of this problem of magnetic-oxide sludge, we are using the potassium-chloride method at one plant; at another, a combination of potassium chloride and captive alkalinity, supplied by potassium silicate; and the same also at a third plant in which the potassium silicate is of value not only for captive alkalinity but to provide the silicate in the boiler water to prevent soft magnesium deposits.

In some cases, efforts to eliminate corrosion by bonded oxygen have followed the course of maintaining extremely low alkalinities in the boiler water. In this case, Reaction [1] is made negligible, and the concentrating-film boiler water may most likely evaporate to dryness, changing the concentration of H_2O in Reaction [2] from that of liquid water when caustic maintains the film liquid, to the value corresponding to the density of steam at the temperature of the surface in question. The method is effective, but not infallible, if Δt is sufficiently large, as proved by a number of instances (36, 31, 25). In any event, however, the method provides optimum conditions for silica deposition in boiler and turbine, as will become apparent in the sections which follow.

THE PROBLEM OF SILICA IN THE BOILER

There are only two possible methods of procedure relative to silica, namely, remove it, or else live with it and make it behave. In fact, silica is a component in boiler waters that must be reckoned with whether or not steps are taken for its removal, since whatever may be left after treatment for its removal accumulates in the boiler water, and in lesser or greater degree the silica problem holds sway. But, as previously noted, it has virtue in combining with magnesium, and thus preventing the soft but troublesome high-magnesium boiler deposits. As shown in Fig. 5, both sodium metasilicate and disilicate (A, III, 15, 16) in the boiler-water temperature range show retrograde solubility, that is, decrease of solubility with temperature increase, and must therefore be classed in the group of salts such as sodium sulphate, carbonate, and phosphate, and hence at surprisingly low value of Δt in the Δt-function, reach saturation and deposition on evaporative surfaces. On the other hand, both potassium metasilicate and disilicate (A, III, 17, 18) show increasing solubility with temperature increase through the whole range to their melting point. In this respect, therefore, they may be classed with the group of materials, such as sodium chloride, potassium chloride, and sodium and potassium hydroxides, which behave similarly. In contrast to the sodium silicate, their Δt_s values (A, IV, 10, 11) are extremely high, and thus wherever a concentrating-film boiler

water develops, they will aid in the prevention of deposition rather than abet it.

Table 4 comprises a comparison of Δt_s of saturated solutions of potassium metasilicate and sodium sulphate, values for the latter being used, since the solubility of sodium sulphate and sodium metasilicate are similar in type, and since no vapor-pressure values for sodium-silicate solutions are available. For example, at 621 F, potassium metasilicate has a saturation concentration of 0.40 mol fraction or 85 per cent, the vapor pressure of the solution is 79.4 psia, water at this pressure boiling at 311 F. Thus Δt_s has the surprisingly high value of 310 deg. On the other hand, sodium sulphate and undoubtedly sodium silicate practically to the same extent at 617 F, or approximately the same temperature, has a saturation concentration of 0.012 mol fraction or 8.7 per cent, the vapor pressure of the solution is 1710 psia, and water at this pressure boils at 614 F. Thus Δt_s in the latter case is only 3 deg. What this means in the concentrating-film boiler water is clearly evident from the previous discussion of the Δt function: The potassium metasilicate will not hide out, and will not be scale-forming, while the sodium silicate will hide out, and will be scale-forming. Or stated in a different way, in the 1800-psia tube, illustrated in Fig. 1, potassium silicate can never reach saturation in the concentrating-film boiler water at surface A, since the vapor pressure of its solution at saturation never attains that pressure; while with sodium sulphate and sodium silicate, Δt of not over 3 deg brings about saturation.

Fig. 8 illustrates further properties of the potassium silicate that emphasize the value of maintaining the boiler on potassium equilibrium, thus avoiding the sodium silicates. The concentration as mol fraction is plotted as ordinate, and the abscissa is the mol ratio of SiO_2/K_2O. This brings out a fact concerning the silicates that does not pertain to other boiler-water salts. If, for instance, one started with a pure solution of potassium hydroxide, the ratio SiO_2/K_2O would be zero. If silica is now added, it dissolves, and the amount of silica added may be such that the mol ratio of SiO_2/K_2O becomes 0.6, 1.6, 2.0, or more, or any intermediate figure. Let us consider what happens to a solution of silica and potassium hydroxide with varying ratio of SiO_2/K_2O, and let us choose the isotherm of 680 F, which is one of those chosen by Morey (7) in the experiments in which he obtained the data plotted in this figure. With a ratio SiO_2/K_2O of 1.0, and concentration to saturation, the solid product that precipitates is potassium metasilicate. As the ratio SiO_2/K_2O is increased, and becomes almost 1.3, the solid product that precipitates is no longer potassium metasilicate, but potassium disilicate. Continuing along the 680 F isotherm, with silica concentration increasing until the ratio SiO_2/K_2O is a little greater than 2.2, the saturated solutions precipitate potassium disilicate as solid phase. Beyond this point, the solid phase becomes tetrasilicate ($K_2O \cdot 4SiO_2 \cdot H_2O$). Finally, only when the ratio SiO_2/K_2O is somewhat more than 3.4, is the field reached in which silica precipitates. Following the 392 F in place of the 680 F isotherm, we find that at a ratio SiO_2/K_2O of 2, the dotted line separating the field of silica from the field of potassium tetrasilicate intersects the experimental isotherm of 392 F, at practically zero concentration. Therefore if the boiler water contains silica and potassium hydroxide in the mol ratio of 2, and the operating pressure is 225 psia, corresponding to 392 F, precipitation of potassium tetrasilicate and silica might readily occur in the concentrating-film boiler water. But, if the ratio SiO_2/K_2O under that condition were 1.6 or lower, then this precipitation would not occur, and at the pressures above 225 psia, and at ratios of silica to alkali in the range of 2 or less there is practically no possibility of deposition of silica or the potassium-silicate compounds in the concentrating-film boiler water for any values of the Δt function within the range of reason. The interesting point is, that the higher the operating pressure of the boiler, the more certainly this is the fact.

Now contrast with this what happens in the case of sodium silicate. The exact data are not known, as they are being worked upon by Morey (14, 15) at the present time. But one can gain an idea of what they will be by reference to Fig. 5, depicting the enormous contrast in the solubilities of the sodium and the potassium silicates. As regards stability, also, the potassium silicates have the advantage (16), since, as far as known, no tetrasilicate of sodium comparable to the tetrasilicate of potassium exists, and any decomposition of sodium disilicate results directly in deposition of silica. The relationship of incongruent solubility (27) between water and the silicates, signifying their decomposition by water, may occur in the boiler as well as in the turbine, but will be discussed as a part of the role of silica in the turbine.

Our observations under actual operating conditions of the boilers at Springdale, and of test experiments carried out in bombs, agree fully with the conclusions which are reached by study of this diagram.

But, the question may arise, this is true for the potassium silicates as compared with the sodium silicates, but what of the potassium-aluminum silicates and the potassium-iron silicates as compared with the sodium-aluminum silicates and the sodium-iron silicates? Inasmuch as only very limited (33) experimental data are available, so far as we know, dealing with solubilities of the alkali-aluminum silicates, we must revert for our conclusions to our observations in operating boilers. In the Springdale boilers, in the generating tubes, where hard, flinty deposits of sodium-aluminum silicate, mixed with silica, magnetic iron oxide, and copper, prevailed during the maintenance of sodium equilibrium in these boilers, in spite of everything that we could do, with advent of the use of potassium equilibrium and the appropriate relationship between silica and alkali in the boiler water, these deposits have completely disappeared and the tube surfaces are clean. Very likely, the vastly different solubility of the potassium silicate as contrasted with the sodium silicate is the influential factor in producing this result.

Application of the Δt Function to the Isobaric-Polythermal Conditions in the Superheater

Thus far, in our discussion of the concentrating-film boiler water, and the conclusions that follow from application thereto of the Δt function, we have directed our attention mainly to problems involving the low Δt_s values of substances like sodium sulphate, sodium phosphate, and sodium silicate. But where conditions of steam-blanketing or film-boiling are severe, Δt of the tube surface may be quite high, as indicated by finding a spheroidized condition of cementite in, or even decarburization of, the metal when metallographic examination is made. In the superheater, too, any boiler water which carries over in the steam must concentrate and come to equilibrium therewith at a value of Δt equal to the superheat of the steam. Conditions are more severe, but the principles are the same as those illustrated in the experiment, Fig. 3. It will be sufficient for our purposes to consider the temperature range from 212 to 1400 F, and, therefore, maximum Δt of 1188 deg. This involves consideration of solutions well above the critical temperature of water.

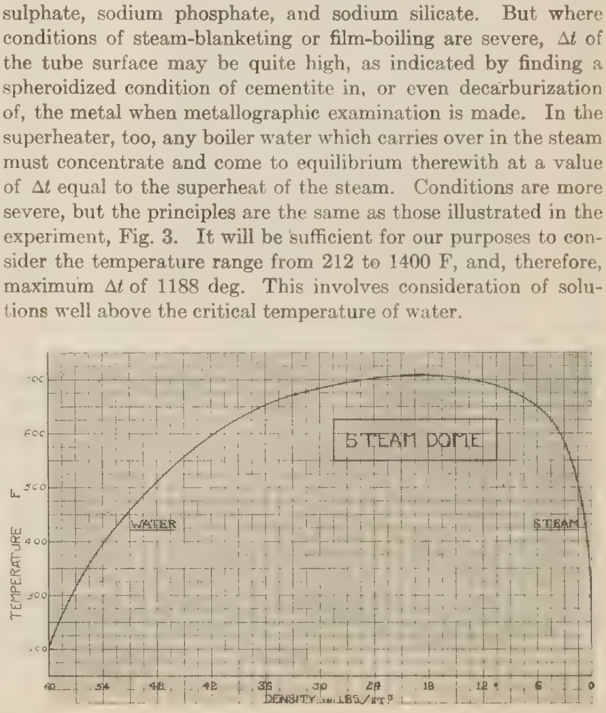

Fig. 9 The Steam Dome

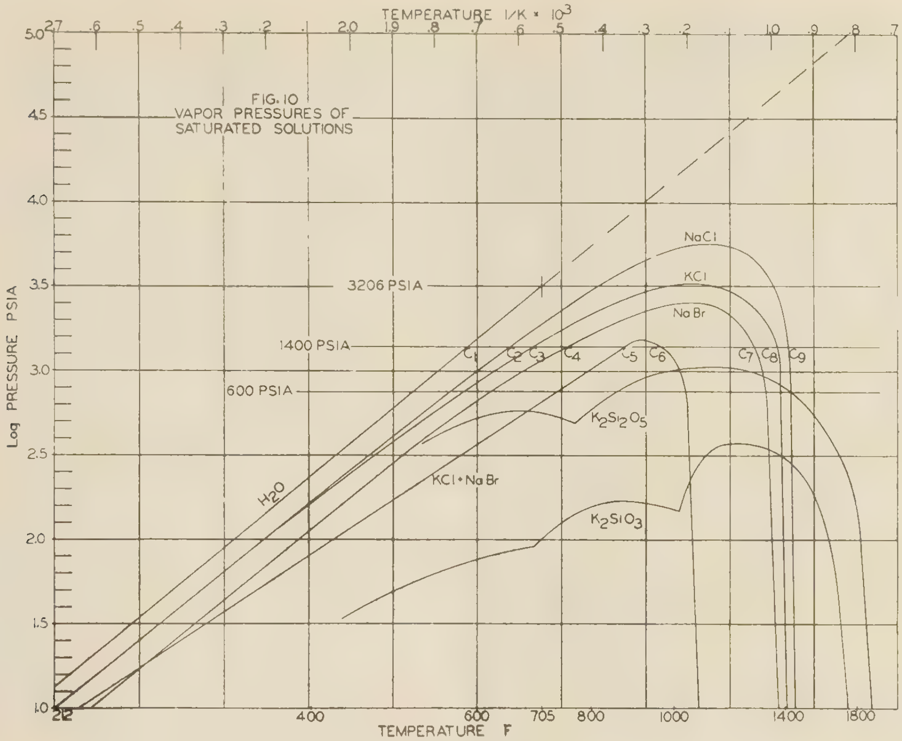

Fig. 10 Vapor Pressures of Saturated Solutions

What is the significance of critical temperature of water in relation to solutions? Morey (14), calling attention to the steam dome as illustrated in Fig. 9, points out that the densities of water and steam gradually approach each other and become equal at critical temperature without discontinuity. Increase in density of steam is very rapid as critical temperature is approached. At and just below critical temperature, the steam dome is almost flat. He states:

"There is no discontinuous change at the critical temperature of water, and there is no reason to expect a change in the solvent power of water at critical temperature when the liquid and vapor phases merge continuously into each other.

"The critical temperature of water is a characteristic of pure water only. When a binary system consisting of water and a soluble compound such as potassium disilicate is considered, the critical temperature of water is without significance. A solution of salt in water raises the boiling point, that is, decreases the vapor pressure, and if the salt is sufficiently soluble, the vapor pressure of the saturated solution at the critical temperature of water is far less than the vapor pressure of pure water."

Fig. 10 shows the relationship for a few salts (A, III, IV) of vapor pressure to temperature of saturated solutions, the data for these curves being taken from papers by Morey (7) and Keevil (6), and from the International Critical Tables (17). These curves, representing the vapor pressure of saturated solutions at ascending temperatures, evidence no break at the critical point of water and show the course that such salt solutions would take in the concentrating-film boiler water or in the superheater, in their endeavor to reach equilibrium on the basis of the Δt function at any value of Δt under consideration.

Table 5 shows what happens when such solutions of one dissolved component or in one case of mixed components (potassium chloride and sodium bromide) are carried over in the steam of a boiler operating at 1400 psia and are raised in temperature to 1000 F in passing through the superheater. The data are obtained as follows:

The isobar of 1400 psia, Fig. 10, intersects the vapor-pressure curve of water at C_1, then successively on their ascending branches, those of NaCl at C_2, KCl at C_3, etc.; and on their descending branches, NaCl at C_9, KCl at C_8, etc. The isobar does not intersect the curves of $K_2Si_2O_5$ or K_2SiO_3, or those of NaOH or KOH, which are not shown because they lie below 10 psia, the base line of the diagram. To illustrate the significance of an intersection, let us consider C_3 of KCl. The temperature corresponding to C_3 is 691 F, which is the saturation temperature for a solution of KCl with vapor pressure of 1400 psia. The saturation concentration of 61 per cent is obtained from the solubility curve of KCl (see solubility data, A, III, 5). The difference of the temperatures corresponding, respectively, to C_3 and C_1, 691 and 587 F, or 104 deg is Δt_s. Since saturation occurs at 691 F, and 1400 psia, and since the temperature in the superheater rises to 1000 F, pressure remaining unchanged, the solution must evaporate to dryness, in a manner similar to the solution of calcium chloride, illustrated in Fig. 3.

The Δt_s values corresponding to the other intersections C_2, C_4, and C_5 also lying on the ascending branches of the curves, are less than the superheat, 413 deg, hence the solutions to which they correspond must evaporate to dryness. The huge difference that may exist between solo solutions and mixtures is well illustrated by the Δt_s values of only 104 and 166 deg, respectively, for KCl and NaBr, while that of their mixture is 287 deg, or more than the sum of their separate Δt_s values. Inasmuch as the 1400-psia isobar does not intersect the vapor-pressure curves of solutions of the potassium silicates, nor of those of caustic soda and caustic potash, these solutions will not attain saturation under the conditions we have assumed, and the salts will remain

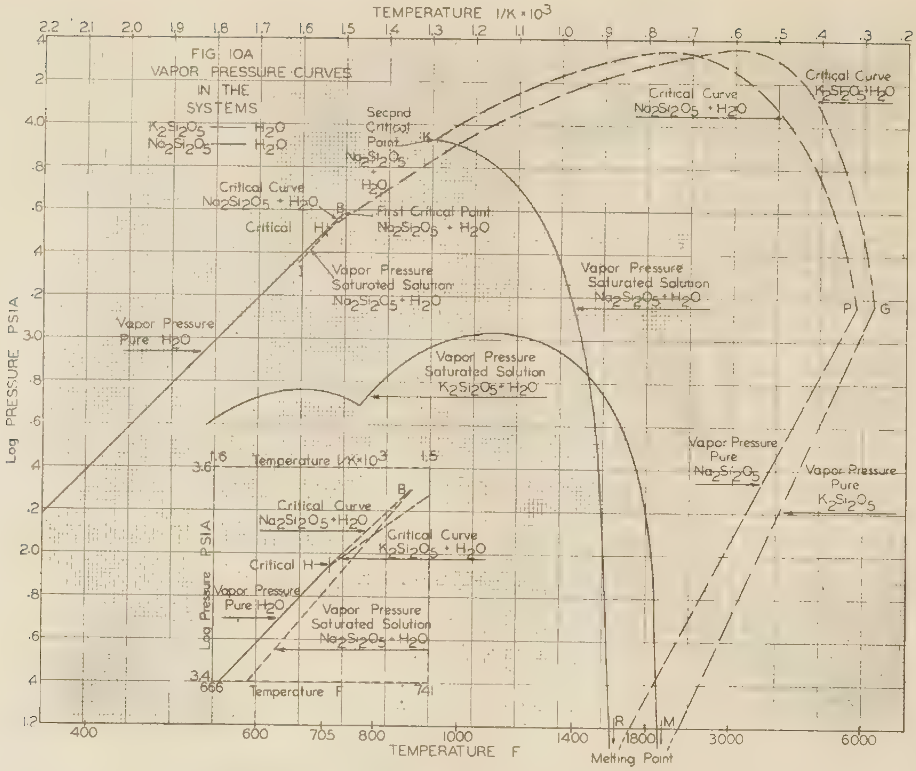

Fig. 10(a) Vapor-Pressure Curves in the Systems $NaSiO_5$-H_2O and $K_2Si_2O_5$-H_2O

in solution even as they pass from the superheater at the full temperature of 1000 F.

The intersections of the 1400-psia isobar with the descending branches of the various saturation curves shown, occur well above the superheat temperature of 1000 F, except in the case of the mixture KCl + NaBr. In that case, the saturation curve reaches its maximum vapor-pressure value at approximately 914 F and, thereafter, as its equilibrium vapor pressure becomes lower with increasing temperature (also increasing concentration), intersects the 1400-psia isobar at C_6, 970 F, and at 1000 F lies well below it. This means that at 970 F, the potassium chloride and sodium bromide must redissolve, and at 1000 F, will pass from the superheater in solution. If the superheat temperature was a little above 1400 F, all solo salts noted in Fig. 10 would redissolve and pass from the superheater in solution. In case of mixtures, evaporation to dryness might never occur, but if it did, redissolution would be almost certain to occur, and very likely at temperatures considerably lower than 1400 F. The mixture of potassium chloride and sodium bromide is the only one on which data are available at present. On the descending, as well as on the ascending branch of its saturation curve, its marked difference in values from those of the saturated solo-solution curves of its components emphasizes the need of data of this sort for mixtures of salts which will be representative of those characterizing boiler waters. Experimental technique is extremely difficult, but Partridge and Kaufman of our laboratories are drawing plans for the work.

The vapor-pressure curves of saturated solutions thus far discussed, and presented in Fig. 10, are of salts whose solubilities continuously increase with rise of temperature. The descending branch of each of the vapor-pressure curves is approaching closely to the melting temperature of the dry salt as it intersects the base line (pressure = 10 psia) and attains that temperature somewhat below at a pressure which is the vapor pressure of the pure salt, and a concentration of 100 per cent dry salt.[7]

If, now, the temperature of the melted salt is raised, the pressure created by the pure vapor of the salt increases. In the case of potassium silicate, Fig. 10(a), this follows a line such as MG,[8] originating at the melting point and attaining critical state at G (assuming no decomposition of salt or vapor), just as water attains its critical state at H. The line connecting G and H, and representing the lowest vapor pressures of critical conditions for mixtures of water and potassium disilicate, is known as their critical, or plait-point curve. The critical curves of sodium chloride and potassium chloride for a short distance beyond H have been determined by Schröer (A, VIII). The significant point about this curve is that throughout its length, it lies well above the vapor-pressure curve of saturated solutions of potassium disilicate, no intersection of the saturation vapor-pressure curve and of the plait-point curve occurring. This is a general property of the plait-point curves of substances whose solubilities increase with increasing temperature in both branches of the saturation vapor-pressure curve. Between the saturation vapor-pressure curve and the plait-point curve of any substance lie the vapor-pressure curves of its unsaturated solutions, critical state

[7] The melting point under these conditions will be very slightly different from the melting point usually cited in the handbooks, a value obtained under ordinary atmospheric conditions.

[8] The broken-line curves MG, RP, GH and the two segments PK and BH are hypothetical only, definite points not having been established. K has been tentatively fixed by Morey. Definite location of B is not established other than that it lies slightly above the critical temperature of water. One point of BI has been established by Morey (private communication) at 662 F, but the general form of these curves of decreasing solubility along the ascending branch has been established by the data on Na_2CO_3 and Na_2SO_4 (A, III, 2, 19). The incongruent solubility relationship of $Na_2Si_2O_5$ probably begins shortly below I.

of the solutions occurring at their points of tangency to the plait-point curve (40). At or above the critical point of the solution, there is no discontinuity of solubility, the substance dissolved in the water remaining in solution in the gaseous phase, and solubility increasing with both temperature and pressure, but especially the latter (41, 20, 42).

Now let us consider the salts whose solubilities are retrograde in the range of boiler-water temperatures. In keeping with their low solubilities, their Δt_s values are low, and at 1400 psia (Table 5), evaporation to dryness will occur almost immediately upon entry into the superheater. The question of moment to anyone interested in boiler waters is: Will this same condition continue, as temperature is increased well above 705 F in the superheater, or will changes occur resulting eventually in a descending branch of the vapor-pressure curve, and therefore greater solubility?

In regard to critical characteristics, the data on sodium disilicate by Morey and Ingerson (14, 15) are alone available. Line *IB* represents the saturation vapor-pressure curve of sodium disilicate. It passes close to the critical point of water *H* and, slightly beyond at *B* intersects the critical or plait-point curve joining *H* with *P*, the critical point of sodium disilicate, and passes through points *B* and *K*, but with a break in the curve over the interval *BK*. Beginning at point *B*, the intersection with the critical curve, the sodium disilicate exists only in solid form, or dissolved in the fluid (gaseous) phase; the coexistence of liquid and vapor is impossible.

Morey and Ingerson (21) state: "When the critical curve is intersected in one point by the solubility curve (as at *B*), it must also be intersected in a second point."

In searching for this second point, they (15) started with sodium disilicate at its melting point just below *R*, Fig. 10(*a*), and by controlling as desired the pressure of steam with which they surrounded it, found vapor pressure to increase with falling temperature along the line *RK*. Finally near *K* (8500–8700 psia, 932 F), the second point of intersection of the saturation vapor-pressure curve with the critical curve has been located tentatively. From *K*, the descending branch of the vapor-pressure curve extends unbroken through *R* to the melting point of the salt, saturation concentration being high and increasing with temperature increase.

The break *BK* between the first and second intersections of the saturation vapor-pressure curve and the critical or plait-point curve is a little-explored region, of which Morey (14) states:

"Numerous experiments have been made at temperatures ranging from 375 to 500 C (707 to 932 F) and at pressures up to 16,500 psia and with mixtures ranging in composition from sodium metasilicate to the eutectic between sodium disilicate and quartz. These studies are not complete, but they all indicate that under the conditions of experiment the percentage of sodium oxide and silica in the vapor becomes large. Some results as yet unconfirmed indicate over 50 per cent by weight of solid material in solution in the vapor.

"The conditions of these experiments are more drastic than those encountered in steam boilers and superheaters, but, nevertheless, the very considerable power of steam under these more drastic conditions to transport ordinarily nonvolatile components will be paralleled by a smaller transport power per unit of steam at the lower pressures."

For the theory of these relationships, in the temperature range above critical for water, which characterize the salts of retrograde solubility, the reader is referred particularly to Morey and Ingerson (21) and to Schröer (20).

The very great difference in behavior in the region above 705 F of sodium disilicate, representative of the inorganic salts, with retrograde solubilities approaching zero at the critical temperature of water from those of steadily increasing solubility is apparent from Fig. 10(*a*). While no data on the former are available, other than those of Morey and Ingerson on sodium disilicate, it seems practically certain that they, like it, must have a descending branch to their saturation vapor-pressure curves, and along that branch must increase in solubility with rising temperature.[9] The physical characteristics of such solutions, for instance fluidity, are unknown. Where the curves of mixtures of salts, some with retrograde and some with continuously increasing solubilities, as boiler-water salts, would lie on the diagram, and whether their temperatures on the descending branches of the curves would be low enough to be significant as regards conditions in the superheater, we can only surmise until pertinent data have been obtained.

Once the essential data are available to establish Δt-function curves, such as those in Figs. 4, 10, and 10(*a*), for boiler-water compositions, an analysis of the dilute over-all boiler water, expressed in mol fractions of the components, will make feasible estimation of concentrations and of the condition of the components in the concentrating-film boiler water, or of carry-over in the superheater at different values of Δt, and likewise at any point in the turbine of which the temperature and pressure are known. We may illustrate the principle by the following example:

In Keevil's (6) work, from which the data of the KCl-NaBr mixture were obtained, the concentration of the saturated solutions was not measured. Kaufman and Morris in our laboratory made a saturated solution by agitating water with solid KCl and NaBr and found the boiling point at atmospheric pressure to be 245 F, the composition of the solution being expressed by 0.044 mol fraction of KCl, 0.029 of KBr, and 0.117 of NaBr, a total of 0.19 mol fraction or 56.4 per cent. At intersection C_5, Fig. 10, P_s = 1400 psia or boiler pressure; t_s = 874 F; and P_{H_2O}, the vapor pressure of pure water at t_s, read from the extension beyond critical of the H_2O vapor-pressure curve (hypothetical, but of use in the calculations) is 8040 psia. The ratio P_s/P_{H_2C} = 0.174. Now, let us assume, in the absence of data on mixtures, that the deviation curve of the Halides, Fig. 4, applies to their mixtures as well as to their solo solutions; this may or may not be true but is as near the truth as we can get until more data are available. The value of 0.174 on the ordinate P_s/P_{H_2O} corresponds to X_2 = 0.5 mol fraction on the abscissa. The evalua-

[9] The data of Goranson on albite, $NaAlSi_3O_8$, and orthoclase, $KAlSi_3O_8$, indicate that the complex silicates, such as analcite, $NaAlSi_2O_6.2H_2O$, and leucite, $KAlSi_2O_6$, belong in this class (44, 45).

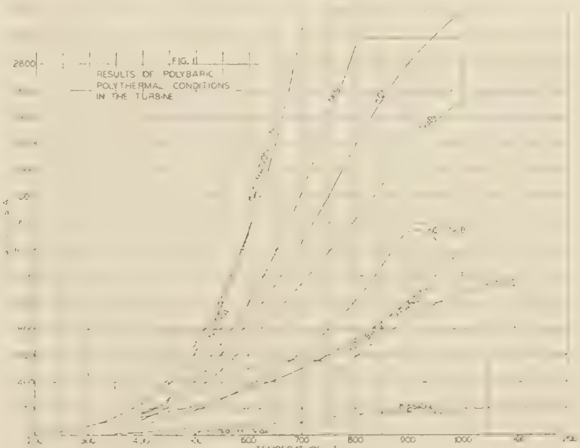

FIG. 11 RESULTS OF POLYBARIC-POLYTHERMAL CONDITIONS IN TURBINE

tion of mol fraction into per cent involves uncertainty because of lack of knowledge of the relative proportions of K, Na, Cl, and Br in the saturated solution at 874 F. However, as the solubility curves of the salts involved are similar in form and in so far as we know do not form double salts, we shall assume that their proportionate concentrations at 874 F, and 1400 psia, remain essentially what Kaufman and Morris found them at 245 F and 14.7 psia. On that basis, the molar concentration of the saturated solution at 874 F is estimated to be 0.116 of KCl, 0.076 of KBr, 0.308 of NaBr, 0.500 H_2O, which correspond to 14.8 per cent KCl, 15.5 per cent KBr, and 54.3 per cent NaBr, a total of 84.6 per cent. The outside limits of variation would be represented by 80.6 per cent if the solution contained only KCl, and 85.2 per cent, if only NaBr. When data become available to obviate the uncertainties involved in the assumptions necessarily made in this case, the method will give a clear picture of the condition of any salts transported as carry-over in the steam, if the temperature and pressure of the latter are known.

Returning to Fig. 10, we note that for a superheat temperature of 850 F, and a boiler pressure of 600 psia, only potassium metasilicate and the alkalies lie below the isobar, and would leave the superheater in solution.[10]

Presumably in a great majority of cases any carry-over of boiler water in the steam, attaining even a semblance of equilibrium at the temperature of the superheater, has become a magma (that is, a thin paste of solution with precipitated salts dispersed therein) by the time it enters the turbine. Once their Δt-function properties are known, it will be possible to prescribe and maintain in the over-all boiler water, proportional relationships among its components which, as concentration occurs, will result in magmas of least disadvantage both to superheater and turbine.

Role of Silica in the Turbine

The magmas, as they leave the superheater, are the product of equilibrium corresponding to its high temperature and boiler pressure. As they enter the turbine and experience lowering of temperature and pressure, they must change in seeking to maintain equilibrium under the new conditions. What happens to the magmas[11] of solo solutions, and of the one known mixture of potassium chloride and sodium bromide, is shown in Fig. 11, developed by Kaufman, for pressure of 1400 psia and total temperature of 1000 F. An ideal adiabatic expansion curve has been used for purposes of illustration, but the conclusions drawn will hold equally well for operating conditions. All curves falling within the area bounded by the saturation line and the adiabatic expansion curve represent undissolved or solid condition of the different substances concerned in this example; all curves falling below the adiabatic expansion curve represent a dissolved condition of the respective substances. Thus at all pressures and temperatures, potassium metasilicate and potassium or sodium hydroxide enter the turbine and continue on through in dissolved form. Sodium sulphate, chloride, and bromide, and potassium chloride, enter in solid form, and remain thus until the region of condensation or close thereto is reached, finally dissolving. The sodium di- and metasilicates, because of low solubility and consequent small Δt_s values like sodium sulphate, almost certainly enter and remain in solid form until the lower-temperature region of the turbine is reached. Apparently potassium disilicate enters the turbine under consideration in dissolved form, begins crystallizing at about 730 F and 560 psia, and then at 536 F enters the region of its incongruent solubility relationship with any moisture, a relationship discussed fully later. The mixture of potassium chloride and sodium bromide enters in solution, then saturates and crystallizes at 984 F and 1330 psia, but redissolves at about 563 F and 300 psia, where its saturation curve crosses the adiabatic expansion curve, considerably above the region of condensation.[12]

With the establishment of Δt-function curves for boiler-water salt mixtures, it will be possible to relate the composition and condition of magmas entering the turbine to their tendency for adherence (47, 48), and hold this minimum by establishing appropriate relationships in the dilute over-all boiler water. However, because of certain intrinsic properties of the silicates, enough is already known to point the way toward rendering them least obnoxious in any carry-over.

In the turbine, above the temperature range approximately 500 F, silica found in any deposition is probably combined in the form of sodium silicate, and thus is water-soluble at the temperatures usual in washing turbines; below the 500 F range, it is uncombined and insoluble, usually being in the form of quartz down to perhaps 250 F, and in the amorphous form thereafter. This quite signal temperature boundary between sodium silicate on the high, and uncombined silica on the low side, leads to the conclusion that there must be definite chemical characteristics of the silicates which cause this condition to exist. What are these characteristics?

In studying the silicates to answer this question, we must perforce use the potassium salts as our examples, as sufficient data on the sodium salts are not of record. One conclusion, however, may be inferred, i.e., that the sodium silicates, because of their retrograde solubility, and consequent very small Δt_s value, will at least equally, and perhaps more strongly, exemplify the characteristics brought out than will the potassium salts.

Morey (14, 27) states: "A large number of compounds become decomposed by, and are incongruently soluble in, water at elevated temperatures." In a solution in which the ratio of silica to potash is that of the disilicate ($2SiO_2$:$1K_2O$) and the temperature is lowered to below 536 F, "the disilicate becomes decom-

[10] As pointed out by Hall (43), the passage of superheated steam over solid sodium chloride in determinations of its solubility in the steam may result in dissolving the sodium chloride with formation of a strong solution, thus largely changing the conditions of the experiment. In Spillner's (18) work, for example, with total steam temperature of 765 F, below 2600 psia (183 kg per cm^2) the sodium chloride would remain solid as long as it retained its purity; at 2600 psia, and 765 F, it would dissolve to form a solution of 0.214 mol fraction or 47 per cent, the solution becoming more dilute over the remaining range of experimental pressure terminating at 3980 psia (280 kg per cm^2). In his experiments on potassium chloride, dissolution of pure solid salt would occur at 1930 psia (135 kg per cm^2) and 765 F, the solution formed having a concentration of 0.302 mol fraction or 64.2 per cent, and dilution occurring with increased pressure. In Straub's (19) experiments at 1540 psia, the sodium chloride would be dissolved up to the temperature 667 F, and solid above that temperature; at 2200 psia, it would be dissolved up to 732 F, and solid above. It is apparent from the curve of KCl + NaBr (Fig. 10) that, if the sodium chloride should collect any impurity from its surroundings during the experiment, the maximum temperature at which it would remain in dissolved form at the pressure level of the experiment would be higher than for the pure salt.

[11] Perhaps better in this case, "equilibrium products" since the example deals only with solo solutions, and one mixture of two components only.

[12] What would happen in the case of a mixture such as that of sodium sulphate and sodium hydroxide? At high temperatures of the superheater, the solubility of sodium sulphate in sodium-hydroxide solutions is probably quite high (the data of Schroeder et al. (8) lead to this conclusion). With temperature falling rapidly in the turbine, solid sodium sulphate would almost certainly form in the magma. In this connection, it is notable that sodium sulphate readily crystallizes as deposit in high-pressure turbines (43). Sodium silicate in the magma would probably duplicate the performance of sodium sulphate (43). In the latter case, the separation of sodium silicate may result in a remanent magma with higher $Si{:}O_2/Na_2O$ ratio. Silica enrichment of this type has played a large part in the differentiation of molten magmas in nature, with granite, which is high in silica, forming in the latter stages (46).

posed by water, and crystallizes only from solutions which contain an excess of potassium oxide. If a solution containing potash and silica in the disilicate ratio is cooled and crystallized below 536 F, the compound separating is not potassium disilicate but potassium tetrasilicate ($K_2O.4SiO_2.H_2O$), and below this temperature potassium disilicate is incongruently soluble in water.... Potassium tetrasilicate becomes decomposed by water at temperatures below 680 F, with quartz becoming the stable phase."

In the light of these facts, we can visualize the chain of events started by the exceptional property of the disilicate of crystallizing from solution in the turbine beginning at about 730 F, as just noted, (Fig. 11). Farther on as the crystals thus formed redissolve at lower temperatures, it seems probable that the relationship of incongruent solubility between water and disilicate develops, with formation of tetrasilicate, which in turn and by the same mechanism splits off silica. The high solubility and high Δt_s value of potassium metasilicate insure its presence in solution throughout the turbine, and by excess of alkali over silica development of the incongruent solubility relationship with water is avoided. The disilicate of sodium, with retrograde solubility curve and small Δt_s value, is very likely to be already crystallized in the magmas entering the turbine from the superheater, but as a region of lower temperature is reached, at which its solubility is higher, decomposition by water presumably proceeds, thus directly forming silica without the interposition of any tetrasilicate. The course of sodium metasilicate in the turbine is probably similar, but more easily controlled than that of the disilicate.

Also significant is the abrupt ending of the solubility curves of potassium disilicate and metasilicate in Fig. 5, the cause thereof being the beginning at the temperature indicated of their incongruent solubility relationship with water, as evidenced by their decomposition thereby.

One may also read the story from the curves in Fig. 8. Starting for instance with a solution in which the ratio of SiO_2/K_2O is 2 (that of the disilicate), we find that the vertical co-ordinate representing this ratio crosses the saturation isotherm of 1112 F at 0.645 mol fraction, and well within the field of stability of $K_2Si_2O_5$, so that any crystals separating from solution will be potassium disilicate, which has high solubility. Following the same co-ordinate downward, we find it crossing the saturation isotherm of 680 F at 0.495 mol fraction, and still well within the field of disilicate stability. But a little lower, at a mol fraction value of 0.385, this co-ordinate crosses the boundary line separating the field of potassium-disilicate stability from that of potassium tetrasilicate. From that point on, down to the very small mol fraction or solubility value of the intersection of this co-ordinate with the 392 F saturation isotherm, any crystals separating will be potassium tetrasilicate, which in this temperature range has the property of decomposition by water with formation of silica. Thus, a ratio of silica to alkali of even a little less than 2 (Fig. 8), normally as one of its inherent properties, splits off silica in the lower temperature region of the turbine. As the ratio of silica to alkali increases, the effect becomes more pronounced.

How are these conclusions affected, if Spillner's work (18) on solvent action of steam is finally found correct? While there has been considerable discussion of the vaporization of boiler-water salts with the steam, and their transport in this form into the turbine, the point is not settled. Straub (19) has been working on it and has made an excellent report. Morey's work (14) indicates that any volatilization of the sodium silicates into the steam will occur as salts and not as silica, if the SiO_2/(alkali) ratio in the boiler water is appropriately controlled by the boiler-water alkalinity. In any event, in the turbine, condensation from vapor phase to solid phase or solution must precede deposition as solid phase. As any silicates thus condense from the vapor phase, they will follow immediately the course of reaction just described, and will likewise equally respond to conditions directed toward controlling the course of reaction.

For the application of these facts to the control of silica deposition in the turbine, we can reduce them to the formula of appropriately controlling the ratio SiO_2/(alkali) in the over-all boiler water, thus obviating conditions favoring any development of the relationship of incongruent solubility between water and the silicates in the turbine, and also thereby establishing conditions unfavorable to any separation of insoluble silica, and in fact favorable to the dissolution of silica already deposited.

With these facts in mind, it is not surprising that there has been an epidemic of silica deposition in turbines. Every step taken in the direction of very low alkalinities in the boiler water, usually as an aid in combating the ravages of corrosion by the bonded oxygen thereof, raised the ratio of SiO_2/Na_2O in the over-all boiler water, and thus abetted the certainty and extent of a relation of incongruent solubility between any silicate, and any moisture or any solution developing in the lower-temperature range of the turbine. Very likely boiler waters have been made too pure. The absence of other soluble salts to exercise their influence in the Δt function of the concentrating-film boiler water, thus restraining an otherwise exaggerated rise in caustic concentration, has led to the necessity of so far decreasing boiler-water alkalinity that the SiO_2/(alkali) ratio has risen to values highly favorable to silica deposition in the turbine. It should be noted that it is not the absolute value of the silica in the boiler water, but the value of the ratio of silica to alkali that is the governing factor. Quite evidently, the practical answer in operation is to build the dilute over-all boiler water to meet the requirements, as expressed both by the Δt function and by the SiO_2/(alkali) ratio, and so to maintain it.

Protection From Embrittlement in the Light of the Δt Function

If caustic embrittlement is caused by strong caustic-alkali solutions, resulting from strongly concentrating boiler water at vapor-water interfaces, application of the Δt function to the problem of protection therefrom leads to significant results. If the evaporation necessary to concentrate the boiler water to high causticity occurs at boiler pressures or thereabouts, and at a surface such as A, Fig. 1, subject to higher temperature than corresponds to the steam pressure, concentration of the boiler water will proceed according to the Δt relationship, and the rapid increase of caustic concentration with increase of Δt, unless restrained by other suitable salts in the boiler water, is illustrated by the curve in Fig. 7.

The type of embrittlement (25) produced by reaction of the boiler water on a tube surface such as A, Fig. 1, is illustrated in Fig. 13, in which the damage resulted, we believe, from a concentrating-film boiler water due to relatively large Δt value. Under such conditions of boiler pressure and Δt values, any protective action of sulphate must be that of scale formation or plugging action, since its solubility is of such small mol fraction even with its increment of solubility in caustic solution, that its aid in deterring concentration of caustic to meet the Δt requirement is negligible for all practical purposes, unless perchance Δt is enough so that the temperature environment is that corresponding to the descending branch of the vapor-pressure curve of sodium sulphate, or its mixtures with caustic soda and other salts (see sodium disilicate, Fig. 10(a), and the discussion concerning it). The usual absence of spheroidization of the cementite in embrittled metal disputes the existence of the necessary temperature, unless the descending branch of the curve is shifted to much lower temperature by reason of the mixture of salts. Sodium phosphate, as used by Schroeder and Berk (24) and by Whirl

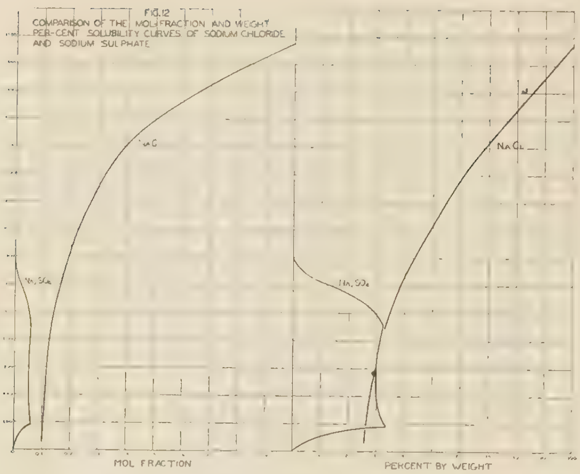

Fig. 12 Comparison of Mol-Fraction and Per-Cent-by-Weight Solubility Curves of Sodium Chloride and Sodium Sulphate

Fig. 13 Embrittlement Due to Δt-Function Conditions on a Generating Tube Surface

and Purcell (13), holds any concentration of alkalinity in check, because purposely the over-all boiler-water alkalinity is of captive type based on sodium phosphate.

On the other hand, sodium or potassium chloride, and the potassium silicates, with their relatively high mol-fraction solubilities, and correspondingly large values of Δt_s, if maintained in appropriate concentration in the over-all boiler water in relation to caustic, can thus be made to restrain the caustic concentrations within limits recognized as safe. Examples of protection thus obtained, though not purposely, are found in the boilers of ocean-going vessels, in which embrittlement has only infrequently been recognized (37, 38), and whose boiler waters contain considerable chloride, resulting mainly from condenser leakage. Tighter condensers on the newer ships may alter this record unless appropriate protective measures are taken. The ratio of chloride along with sulphate, recommended by Straub (22), falls in the same category of protective measures. The potassium silicates are not only of high Δt_s value, but combine as well with that part of the captive alkalinity which must be credited to them in the over-all boiler water.

If the evaporation of the boiler water in concentrating the caustic occurs at high temperature but with escape of the steam readily so that pressure at the water-vapor interface is low, then about the only salts that would help in the protection from embrittlement by curbing alkalinity are the phosphates and potassium silicates on the basis of captive alkalinity, and potassium metasilicate by reason of its high solubility. Any effect of the other salts is apparently a scale-forming or plugging action.

If the evaporation is at a cooler temperature, the results are practically the same as for the high-temperature low-pressure conditions. It is interesting to note from Fig. 12, that sodium chloride for all conditions of temperature and pressure has a higher mol-fraction solubility than sodium sulphate, and is thus of more value in curbing caustic concentration. The same is true of potassium chloride. Perhaps scale-forming or plugging action of the sodium sulphate enhances its protective action against embrittlement. In view of its very small Δt_s value, that seems to be the only logical reason for any advantage in the different levels of the ratios of sodium sulphate to alkalinity in relation to pressure which have been so widely used for the purpose of protection from embrittlement.

On the Technique of Substituting Potassium for Sodium

In condensing plants supplied with evaporated make-up water, the feedwater is practically devoid of sodium save for small amounts derived from condenser leakage and evaporator carry-over, up to the point at which treating chemicals in the form of sodium salts are added. Use of potassium chemicals in the treatment in place of sodium chemicals will thus result in a boiler water in which the ratio of potassium to sodium is quite high, and controlling.

In those plants in which the make-up is treated with zeolite softeners of various types, substitution of potassium chloride for sodium chloride in the process of regeneration of the zeolite effects the result that the softener may be run to substitute potassium for sodium, as well as for calcium and magnesium. Work done in our laboratories shows that this is readily feasible and economical. This applies to the zeolite softeners run on the sodium cycle. On the other hand, if the zeolite is run on the acid cycle, neutralization of the treated water with potassium hydroxide or other alkaline potassium salts will supply a feedwater which in many instances will contain the required amount of potassium to establish satisfactory potassium equilibrium in the boiler water. If the zeolitic operation includes use both of the sodium and the acid cycles, then regeneration with potassium chloride will change the sodium cycle to a potassium cycle, and the neutralization of the water from the acid cycle with potassium hydroxide or with the water from the potassium cycle will result in a feedwater of sufficient potassium concentration.

If the softening operation is of the lime-soda type, the substitution of potash for soda ash will increase considerably the ratio of potassium to sodium in the water. In coagulation processes, the substitution of potassium for sodium salts will also aid in building the potassium concentration in the water.

In general, the requisite supply of potassium salts in the boiler water for potassium equilibrium will be accomplished if throughout the treating processes, the sodium chemicals now in use are replaced by corresponding potassium salts. In the case of some raw waters relatively low in sodium content, it may be possible to build a sufficient potassium concentration in the boiler water for the potassium equilibrium, by the addition of potassium treating chemicals appropriately chosen, and supplied in amount to maintain in the boiler water, as determined by analysis thereof,

the essential preponderance of potassium over sodium in the ratio K/Na. In a great majority of the cases, however, it will be advantageous to effect substitution of potassium for sodium in the feedwater by one of the methods of treatment as indicated. For the boiler water itself, in order to establish the relationships demanded on the basis of the Δt function, and the required ratio of SiO_2/K_2O for the prevention of any incongruent-solubility relationship in the turbine, the potassium chemicals of required type may be added as direct treatment.

Tiger (23) has recently proposed that the anion-cation exchangers, which produce water well-nigh equivalent to distilled water in so far as most of the dissolved salts are concerned, could well be used for make-up purposes in condensing plants in place of evaporators. He states, however, that silica is not removed in the process and suggests, therefore, that silica-removal equipment be supplied for preliminary treatment to remove silica before the water goes to the exchanger. The vast difference that is wrought by maintenance of potassium equilibrium on the boiler water, and substitution of potassium for sodium therein, is well illustrated by the fact that retention of silica in the water from the anion-cation exchangers may very well be advantageous for the purpose it serves in the boiler water, and in any event can be readily taken care of in the boiler water merely by proper attention to the demands, according to the Δt function, and the appropriate SiO_2/K_2O ratio. If Tiger's proposal to use the anion-cation exchangers will provide practically the equivalent of distilled water, except for a little silica present, and if it will fit into the heat balance of the plant, we see no reason at all to shy away from the scheme.

Conclusion

Starting with substitution in the boiler water of the potassium equilibrium for the universal sodium equilibrium, we have been led to give consideration to a number of problems concerning the boiler-water and steam cycle. In this discussion, we have included those relating to magnesium deposits, hide-out of boiler-water salts, corrosion by the bonded oxygen of the boiler water, silica in boiler and turbine, equilibria developed in the superheater in any carry-over of boiler water, and protection from embrittlement.

The problem of magnesium deposits involves equilibrium under the isobaric-isothermal conditions of the over-all boiler water, and has been included because of the usefulness of boiler-water silica, and the potassium equilibrium in its solution.

There is nothing fortuitous in this grouping of the other problems. Each one derives its origin from that fundamental and ubiquitous phenomenon of the boiler-water and steam cycle, the steam-water interface, where the manifold variations of isobaric-polythermal and polybaric-polythermal conditions are spurred on by the mighty forces set free in the boiler furnace. In systematizing the results which must ensue from the complex network of interrelations of temperature, pressure, and concentration under these conditions, the concept of the concentrating-film boiler water and the application of the Δt function have proved most useful, likewise the relationship of incongruent solubility between water and the silicates. We believe that we would have been considerably dismayed by the conclusions to which we were inevitably led, had not the tools of potassium equilibrium and, in that environment, the utility of silica, been at hand. Many more data are still needed, but we feel confident that this logical approach to the problems of the steam-water interfaces will give to the engineers those cleaner surfaces for the boiler-tube raceway which will bring calculated and actual circulation into better agreement.

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Discussion

R. C. Corey.[13] Sixteen years ago, with the completion of a co-operative investigation by the Hagan Corporation and the Bureau of Mines, the author and his associates published a research bulletin,[14] in which the physicochemical basis for scale formation and prevention in steam boilers was presented. To the present day this bulletin is a sound fundamental guide to the principles of water-conditioning.

In the intervening years, the intensity factors, temperature and pressure, and the unit-heat-absorption rate of boilers have increased considerably to provide greater over-all thermal efficiency. It was inevitable that, as the operating temperatures and pressures increased, the sequence of events within a steaming boiler, from the standpoint of chemical reactions, should approach those of a geological nature. Witness, for example, the occurrence in boilers of analcite, acmite, and a host of other complex solid phases that are found as natural constituents in the earth's crust where they were formed as the result of high temperatures and pressures. The tools of the petrographer and the mineralogist, the polarizing microscope, and the X-ray diffraction apparatus, now are familiar equipment in water-conditioning technology.

The new problems presented by modern boiler-operating practice have become manifold. Hide-out, turbine-blade deposits, excessive metal temperatures as the result of resistance to heat transfer offered by unusual solid deposits, and corrosion of steel surfaces by alkali have required that new ideas be developed of the events occurring at the metal-water and metal-vapor interfaces in the boiler. In an effort to find an answer to the questions presented by these phenomena, the author has taken the initiative and applied the logical reasoning of physical chemistry, not the classical concepts found in our physical-chemistry textbooks, but data that have heretofore been of interest to a relatively small group concerned with the study of phase equilibria at high temperatures and pressures; a field of investigation in which the experimental procedures are very difficult. The principles presented in the present paper are not contradictory to those expressed 16 years ago, as the latter still are valid for the isobaric-isothermal conditions of the over-all boiler water. Rather, they have been developed with the realization that isobaric-polythermal and polybaric-polythermal conditions between the steaming surfaces and the turbine discharge can

[13] Research and Development Department, Combustion Engineering Company, New York, N. Y.

[14] "A Physico-Chemical Study of Scale Formation and Boiler-Water Conditioning," by R. E. Hall, G. W. Smith, H. A. Jackson, J. A. Robb, H. S. Karch, and E. A. Hertzell, Bulletin No. 24, Carnegie Institute of Technology, 1927.

produce changes in the solid and liquid phases that are markedly different from those with which we have been familiar heretofore.

The writer was greatly interested in the author's statement that, when the potassium treatment was initiated at a station in which the sodium treatment still was in effect, the hide-out of salts became worse. The following may offer a possible explanation for this condition: Sodium and potassium sulphate will, either when fused together or when crystallized from an aqueous solution of the two salts, form a solid phase known as glaserite (or aphthitalite). The literature is not definite whether this phase is a double salt or a mixed crystal, i.e., a solid solution of limited range. The most recent view[15] is that at room temperature it is a solid solution of limited range of Na_2SO_4 in α-K_2SO_4 and is given the nominal formula $K_3Na(SO_4)_2$, indicating content of three mols of K_2SO_4 to one of Na_2SO_4. The factors applying most directly to the present case are its solubility and vapor pressure at high temperatures, and these data unfortunately are not available. Therefore the author may find it advisable to carry out such determinations in order to determine if excessive hide-out or other undesirable characteristics, associated with low solubility and Δt_s values, are apt to occur when changing from sodium to potassium treatment or where condenser leakage introduces sodium ion.

With reference to the use of silica to precipitate magnesium silicate in lieu of magnesium hydroxide or phosphate, is this contradictory to the principles outlined in the previous bulletin?[14] The writer would guess that present-day practice results in a form of magnesium silicate, identified in high pressure boilers as serpentine, which differs in physical properties from that formed at lower temperatures.

Considering the extreme importance of the implications of Fig. 10 of the paper, in which the vapor pressure of saturated solutions of several salts is plotted against temperature, and the events at various isobars are described, it is believed that greater clarity of certain statements, particularly those referring to the descending branches of the vapor-pressure curves, would obtain if this material were introduced by a generalized statement of the characteristics of plots of this kind. Although vapor-pressure curves are familiar to many, it is not always evident that they are a special form of phase diagram.

With reference to Fig. 13 of the paper, described as an example of a tube embrittled by a relatively large Δt value in the presence of sodium sulphate and sodium hydroxide, it is not clear that the steel is embrittled in the usual sense of the word; rather it appears to resemble the type of attack which might occur as the result of contact with concentrated alkali at high temperature.

Data indicating the potential solubility of iron in caustic solutions at a concentration, that the author has shown may be easily attained on evaporative surfaces with sufficiently high Δt values, have been developed by Scholder and Weber,[16] who found a boiling 50 per cent NaOH solution to contain 5000 ppm of iron at equilibrium.

G. C. Daniels.[17] Most of the utilities treat their feedwater; some of them because of necessity, others through force of habit or fear of consequences if they did not follow the general practice.

The author admits that there is no problem with dissolved oxygen in the feedwater since it can readily be and is removed by various methods in practically all plants. Yet, the feeding of sodium sulphite is generally continued, and as it hides away in the boiler, more is added. When the boiler water is treated, the pH is maintained in the region of 10.5 to avoid corrosion which nevertheless takes place under steam bubbles where high concentrations and high causticity are reached, as pointed out by the author.

In contrast to the difficulties which have been encountered, when using boiler-water conditioning with its attendant high boiler concentrations, carry-over, and turbine-blade deposits, is the extreme simplicity of using no treatment whatsoever either to absorb any dissolved oxygen that might possibly be present or to maintain any predetermined pH. When pure water is fed into the boiler, the pH generally is found to be between 8 and 8.5. A very thin film of oxide is formed on the internal boiler surfaces which adequately protects the metal and is not dissolved, as is the case if caustic is present, and no concentrations of salts can form under the steam bubbles which are also very much smaller when no dissolved solids are present. The only tube failures that can result from the use of pure water will be those caused from inadequate boiler circulation or flame impingement.

When using no boiler-water conditioning, only three conditions must be observed, all external to the boiler, as follows:

1 The occluded oxygen in the feedwater must be kept below 0.05 ml per l.

2 The operation of the evaporator must be such as to keep the carry-over at a minimum.

3 Condenser leakage must be eliminated. This is not difficult to do with the instruments now available for detecting leaks immediately, and with the divided waterbox condenser, the leaks can readily be located and fixed without shutting down.

Our experience with no boiler-water treatment dates back to 1925 and covers many makes of boilers in the 400- and 900-psi-pressure class. Carry-over and turbine-blade deposits are practically nil or entirely absent. The total solids in the boiler water are frequently below 10 ppm after months of operation. No sludge or iron-oxide accumulations are found in these boilers, and the iron content in the boiler water is a very small fraction of one ppm.

Assuming that the author's theories are correct and that the potassium salts will ameliorate the conditions now encountered with sodium salts, would it not be better to determine in the individual plants whether it is at all necessary to condition the boiler water?

In referring to the experiment on the elevation of the boiling point of water by the addition of salts, the author gives the impression that the steam issuing from the flask is at 212 F instead of being superheated to the temperature of the liquid. The fact that the temperature of the vapor as shown by the thermometer immediately above the liquid is the same temperature as the boiling liquid might be accounted for by the radiation from the surface of the liquid in the flask, which is carefully insulated above the water line.

L. E. Hankison[18] and M. D. Baker.[19] Phosphate in the form of sodium phosphate was introduced for boiler-water conditioning into the 350-psi boilers at Springdale Station in 1923. Results obtained with this treatment have been satisfactory to the extent that some boilers erected in 1924 and 1925 have not been completely turbined since their installation. Clean internal surfaces were maintained by washing the boilers with a stream of water when they were down for overhauling. This type of cleaning did not require the removal of hand caps and,

[15] "A New Group of Isomorphous Compounds," by M. A. Bredig, *Journal of the American Chemical Society*, vol. 63, 1941, p. 2533.

[16] "Über anionisches Eisen," by R. Scholder and H. Weber, *Angewandte Chemie*, vol. 49, 1936, pp. 255–259.

[17] Mechanical Engineer, The Commonwealth & Southern Corp., Jackson, Mich. Mem. A.S.M.E.

[18] Superintendent Efficiency Department, West Penn Power Company, Pittsburgh, Pa. Mem. A.S.M.E.

[19] West Penn Power Company, Pittsburgh, Pa.

consequently, some tubes in the boilers have never been inspected.

After 12 or 15 years of operation there was some loss in capacity of these boilers but inspections of the lower tubes disclosed no adhering scale or deposit. Because these tubes where the greatest evaporation occurred were clean, no inspections were made of the upper bank of tubes. When the 1350-psi boilers were installed in 1937, six 350-psi boilers, erected prior to 1923, were removed. An inspection made of all tubes in one of these boilers disclosed considerable analcite scale (sodium-aluminum silicate) in the tubes of the upper bank. This scale was found in spots and was possibly $^1/_4$ to $^3/_8$ in. thick. While the scale was quite prevalent in the upper bank of tubes, it is questionable whether the entire loss in load on the boilers could be attributed to scale.

After the 1350-psi boilers had been placed in continuous service, analcite scale was found in the generating-section tubes. In an endeavor to eliminate this scale, a steel tank was substituted for the concrete make-up-water storage tank from which silica was being dissolved. Following the use of the steel tank, the amount of silica as SiO_2 in the high-pressure-boiler water was reduced from about 20 ppm to 10 ppm per 1000-ppm concentration of boiler water. While this reduction in silica appeared to be beneficial, it was still necessary to turbine the generating tubes periodically for the removal of silica scale. This scale would form under the loose deposits which accumulated in the tubes in the generating section of the boilers. It was on the bottom of the tubes in patches varying from 1 to 3 in. length and $^1/_8$ to $^3/_8$ in. thickness. The formation of this scale could not be predicted as there were periods of 2 or 3 months' operation, following which an inspection showed no scale formation. At another inspection of the same boiler, sometimes after only 14 days' operation, there would be scale in 50 per cent of the tubes. There have been times when it required 4 hr of turbining to remove a patch of scale from one tube. The boilers were never operated for more than 6 months without requiring turbining in some of the tubes. An analysis of the hard scale that was found is as follows:

	Per cent
Sulphur trioxide (SO_3)	2.5
Carbon dioxide (CO_2)	0.0
Phosphorus pentoxide (P_2O_5)	8.9
Silica (SiO_2)	13.3
Iron oxide (Fe_3O_4)	45.9
Aluminum oxide (Al_2O_3)	10.7
Calcium oxide (CaO)	5.2
Magnesium oxide (MgO)	1.5
Sodium oxide (Na_2)	9.2
Copper (Cu)	1.1
Net ignition loss (calculated)	2.9

Trouble with scale of this nature emphasized the need for some correction in treatment. The problem was discussed with the the author, and upon his recommendation on August 25, 1942, potassium salts were substituted for the sodium salts used for boiler-water treatment. All boilers in the Springdale Station, high and low pressure, were placed under the potassium treatment.

After about 2 months' of treatment, leakage occurred around several handhole plates and at tube rolls. Within the next 6 weeks, the leakage had become so pronounced that there was no question as to the immediate cause for such a large amount of trouble. The potassium treatment was dissolving any silica that had deposited around the gaskets and at the tube rolls where it had been serving as a seal. The gaskets used were monel-clad asbestos, and their failure was caused by the cracking of the metal due to expansion and contraction during the time they were in service. There was no chemical attack on the monel. The potassium-treated water would dissolve the silica scale that was sealing the cracks and permit leakage to occur. This type of trouble was eliminated after regasketing the entire boiler completely, and rerolling all leaking tubes. With potassium treatment, any "sweating" or leakage of a gasket or roll on a hydrostatic test will not take up when the boiler is in service. Gasket-failure leakage had started prior to the use of potassium treatment but the change precipitated the trouble. Within a period of 2 months gasket leakage occurred that might otherwise have been extended over a period of several years.

About the time that gasket leakage was being corrected, large amounts of loose scale were noticed in the boilers. The scale found was mostly magnetic iron oxide containing less than 2 per cent of silica as SiO_2. Scale loosened from the evaporating surfaces of the boiler was found in all the headers and was most abundant about the fifth or sixth month after starting the potassium treatment. In some headers, there was sufficient scale to stop circulation and to cause failure of one tube. Following this failure, all headers were thoroughly cleaned and the loose scale removed. Also, all tubes were turbined to remove any material that might still be adhering to their internal surfaces. An analysis of this scale is as follows:

	Per cent
Sulphur trioxide (SO_3)	Trace
Carbon dioxide (CO_2)	Negative
Phosphorus pentoxide (P_2O_5)	10.7
Silica (SiO_2)	1.8
Iron oxide (Fe_3O_4)	60.3
Aluminum oxide (Al_2O_3)	0.4
Calcium oxide (CaO)	7.3
Magnesium oxide (MgO)	6.2
Copper (Cu)	8.1
Total ignition gain	2.3
Net ignition loss	2.2

This iron-oxide scale could have been caused by either of the following reasons:

1 The silica being dissolved from the acmite (sodium-iron-silica) scale that was present in the boilers when the treatment was started.

2 The hydroxide alkalinity (50 to 100 ppm OH) was concentrating at points of high evaporation and attacking the metal.

It is doubtful if scale was formed by the caustic attack, but the presence of large amounts of loose magnetic iron oxide in sludge deposits could readily be attributed to the high alkalinity. The pH value of the boiler water before and after the use of potassium was maintained in excess of 11. The maintenance of this pH value with potassium produced no hard scale in the sludge or on he evaporating surfaces.

A change in treatment was made to reduce and possibly eliminate caustic attack on the metal. This change was the reduction of the hydroxide content to 10 ppm with a pH value maintained at 10.5 or lower. Potassium chloride was added so that at all times the chloride (Cl) was considerably in excess of the hydroxide (OH). Normally the ratio is 5 of Cl to 1 of OH. The effect of the reduction in alkalinity was soon noticed in the characteristics of the boilers deposits. The sludge in the boilers instead of being soft and fine contained many little scale-like particles. Also, the boiler tubes instead of being clean had the appearance of dried sludge adhering to the entire surface of the tubes. No attack could be detected on the evaporating surface and the color of the deposits changed from black to light gray. In an endeavor to stop the formation of the scale-like particles in the sludge and on the evaporating surfaces, the alkalinity was restored to its former values but the chloride-hydroxide ratio was maintained. Inspections made since the increase in alkalinities have shown no attack on the evaporating surfaces, light-gray deposits in the boilers, and no adhering scale.

In the 1350-psi boilers, the potassium treatment has changed

the characteristics of sludge deposits so that at the present time no sticky sludge is adhering to the drum or tube surfaces. The drums are clean and the tubes are coated with a thin film of light-gray deposit throughout their length. This coating can be easily and completely removed with a stream of water. No adhering scale is found after washing. This condition may result from a change in characteristics of the magnesium compounds present in the sludge. Previously, large amounts of magnesium phosphate and magnesium hydroxide were noted when making microscopic examination. Now magnesium silicate is present. When using sodium treatment, the cleaning schedule was that each third time a boiler was removed from service, regardless of the time interval between, or the first outage after 3 months of service, the tubes in the generating section had to be washed and those containing scale turbined. Recently, with potassium treatment, the boiler had seven outages, due to gasket and roll leakage, during a 4-month period without cleaning the internal surfaces. The tubes were then washed but their condition was such that the boiler could safely have been put in service without cleaning. After 15 months of potassium treatment, there has been no need for turbining tubes to remove silica scale that was being formed.

Following the discovery of silica-bearing scale in the 1350-psi boilers a complete inspection of all tubes in some of the 350-psi boilers was made. The inspections showed thin layers of scale under loose sludge deposits in the upper bank of tubes where there is little circulation, and sludge accumulates. Several of the boilers were turbined to remove this scale. No boilers have been turbined since the introduction of potassium treatment, and the scale has disappeared from the boilers that were not turbined.

Since potassium treatment was started, no turbines have been opened, so the condition of the blades is not known. The reduction in capacity now existing may be restored by water-washing or spindle adjustment.

Max Hecht.[20] Within a span of some 40 years, high-pressure steam generation has increased tenfold since the initial operation at 250 psi at the old Fish Street Station. Throughout this period, the general trend in so far as it concerns the purity of the boiler water has been to generate steam from relatively dilute boiler waters. Credit for this trend should be accepted by the mechanical engineers who demonstrated that pure water could be produced by distillation, and that one of the gases contributing to corrosion, dissolved oxygen, could be removed mechanically by deaeration. Credit for the application of distilled water for make-up in stationary plants is given to the late Prof. C. F. Hirshfeld, The Detroit Edison Company, for his pioneering work at the old Delray Station, and for deaeration, to the researches of J. R. McDermet of the Elliott Company, which resulted in the commercial utilization of the vacuum-type deaerator. Both of these developments came in the period, 1910–1920.

The concept of generation of steam from relatively dilute boiler water has persisted, marked by design changes in the equipment to meet the demands of higher steam pressures and temperatures, and the availability of new chemical processes for conditioning the boiler feedwater. New chemical processes and apparatus have been made commercially available for the primary conditioning of boiler-feed make-up, and numerous chemical reagents are used for the secondary conditioning of the boiler water. In the early 20's, boiler manufacturers supplied equipment of liberal design, and such units were operated at not too excessive ratings. Steam pressures were relatively moderate and steam temperatures were not unduly high.

The problems and difficulties of deposits on boiler tubes, in superheaters, and on turbine blades, and of corrosion now being encountered, were present in the earlier operations, either entirely masked, or if observed were of less major significance than they are today.

During the past 25 years, the water technologist has drawn heavily from the knowledge of the physical chemist, and in the paper under discussion, from the studies of the geophysicist. An important and notable contribution was made by George W. Morey, geophysical laboratory, Carnegie Institution of Washington, Washington, D. C., in his paper, "The Solubility of Solids in Water Vapor." This formed the principal paper of the Round Table Discussion on Action of Water Vapor at High Temperatures and Pressures, presented at the 1942 meeting of the American Society for Testing Materials, Proceedings of the A.S.T.M., vol. 42, 1942, pages 980–988. The proper utilization of this knowledge has permitted the water technologist more accurately to analyze conditions brought to his attention, and so make the proper recommendations to ameliorate the troubles that the operator is encountering. It is apparent that in the past the thought in treatment has been directed toward the main body of the boiler water. The investigations of F. G. Straub, pointing toward improvements in ratios for embrittlement protection and turbine-blade deposits are, in direction, a departure from this concept. Lack of adequate solution of problems encountered in the steam-and-water cycle of the present-day high-pressure-and-temperature stations demands a new approach, and this, it is believed, has been made by considering the behavior of water at the steam-water interface.

It is apparent that the operator has been demanding a fluid for use in evaporative processes, which has had removed from it all of the deposit-forming and corrosive characteristics, both in the liquid and vapor state. This implies either the complete removal of the solid phases or their conversion to a form that is not troublesome throughout the steam-and-water cycle, as well as the total elimination of all of the gases contributing to corrosion. In spite of the stepwise progress on separate phases of the general problem made in the past, to meet this demand, new and unpredictable water problems have confronted the operators. Are the present concepts in this paper adequate to overcome these difficulties?

The author has made an impressive and convincing presentation, based upon scientific information now available. His analysis requires detailed study to appreciate the valid theoretical considerations advanced. The operators may find adequate proof of the theoretical applications proposed, presumably from the short experience presented in the operating discussions, and possibly by trial in their own plants. The writer is frank to admit that it will require some time to digest the author's valuable presentation. There come to mind some minor items which may be of interest.

In the writer's opinion, "hide-out" is a self-aggrevating condition. It is most likely to begin in tubes with poor circulation. The deposition of salt further impedes the already restricted flow. The composition of the water must therefore be such that t_e is never reached, even in the tubes having the poorest circulation.

It has been observed that, when boilers are initially put into service, with no preparation other than the customary boiling-out, a considerable amount of iron oxide is found in the drums. This has been attributed to the mechanical removal of mill scale from the drums and tubes during the shakedown operation. Kaufman and Marcy removed iron oxide from bombs containing 5 per cent caustic soda at 470 F but found no attack on the iron. It might be of interest that in the same temperature range, with low alkalinities (pH 8. to 9.5), hydrogen has been detected in steam from boilers even though iron-oxide sludge was not a problem. Oxygen was also found simultaneously with the hydrogen. It is therefore not necessary to have com-

[20] Pittsburgh 17, Pa.

plete absence of dissolved oxygen in order to secure hydrogen evolution or attack of the iron by water.

The values reported[21] are as follows:

	Oxygen, cc per l	Hydrogen, cc per l
Boiler feedwater	0.20	0.0380
Boiler water (drum)	0.05	Not detectable
Steam superheater outlet	0.22	0.0648

C. E. Kaufman.[22] In the intelligent application to boiler-water treatment of many of the principles enumerated by the

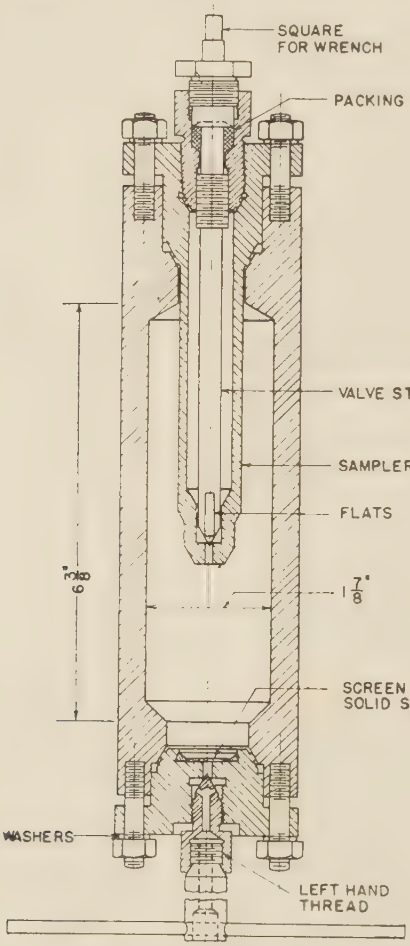

Fig. 14 Solubility Bomb

author, fundamental data on solubilities and vapor pressures at relatively high temperatures are required. The author has marshaled pertinent material available from the literature but gaps are evident, especially in the data relating to aqueous solutions of potassium compounds, and in the almost completely neglected field of salt mixtures in water.

We are endeavoring currently to fill in some of the missing facts. The experimental technique at elevated temperatures and pressures is somewhat involved, and fixing of quantitative values requires repeated trials. Therefore, at this time, we shall merely mention briefly qualitative behavior.

The solubility bomb in Fig. 14 of this discussion, of the type used by Schroeder, Berk, Partridge, and Gabriel[23] at the U. S. Bureau of Mines, is being employed in a suitable thermostat. Saturated solution, representing the equilibrium existing at a fixed experimental temperature, is taken through the valved

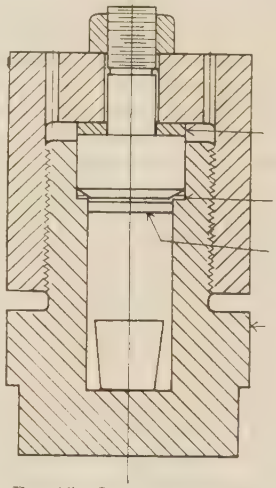

Fig. 15 Quenching Bomb

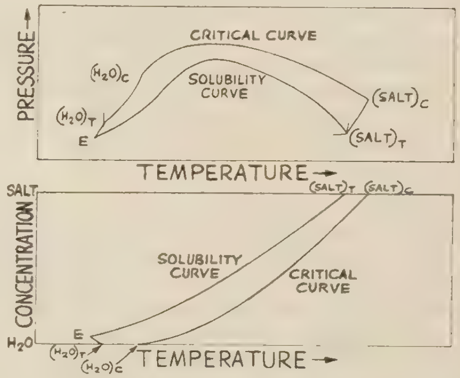

Fig. 16 Nonintersecting Critical and Solubility Curves

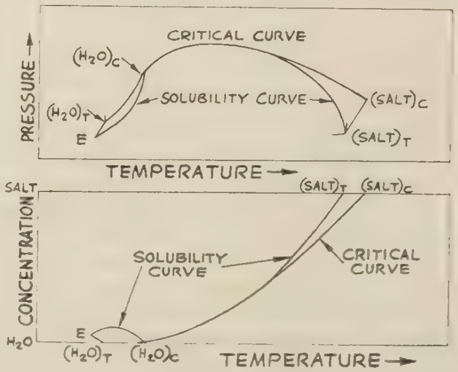

Fig. 17 Intersecting Critical and Solubility Curves

sampler, while the solid phase can be isolated by blowing down excess liquid through the bottom valve. In other experiments,

[21] "Corrosion, Its Causes and Prevention," by F. N. Speller, second edition, McGraw-Hill Book Company, Inc., New York, N. Y., 1935, Tables LXV and LXVI, pp. 424, 425.

[22] Research Engineer, Hall Laboratories, Inc., Pittsburgh, Pa.

[23] "Solubility Equilibria of Sodium Sulfate at Temperatures of 150 to 350°," by W. C. Schroeder, A. A. Berk, E. P. Partridge, and A. Gabriel, *Journal of the American Chemical Society*, vol. 57, 1935, p. 1539; vol. 58, 1936, p. 843; vol. 59, 1937, p. 1783.

the sampling head can be removed and connection made to a pressure gage. In a preliminary investigation with materials of glass-forming properties, the quenching bomb of Morey,[24] Fig. 15, is being utilized.

As the author has mentioned, aqueous solutions of individual salts, and perhaps mixtures, can be classified in two ways, subject to certain variations:

1 Those whose solubility or vapor-pressure curves do not intersect the critical curve.

2 Those with intersections.

Fig. 16 of this discussion illustrates the first type with continuous solubility extending up to the melting point of the pure salt. Sodium chloride, potassium chloride, and the potassium silicates behave in this manner. Fig. 17 shows typical plots for salts of the second type with the solubility curve interrupted by the critical curve. A number of substances of this class are normally found in boiler waters: sodium sulphate, sodium phosphate, and the sodium silicates, for example. Moreover, these salts show retrograde solubility in the temperature range before the first intersection with the critical curve and are assumed to have very low solubilities at the critical temperature of water.

The salts of constantly increasing solubility will naturally be of advantage in the boiler water, as explained in the paper, because their ΔT_s values are in general much greater than those of the retrograde type for the pressures corresponding to boiler operation.

We have then the following preliminary information: In confirmation of values in the literature, potassium sulphate appears to resemble sodium sulphate; solubility becomes retrograde well below the critical point of water, and ΔT_s is probably never very appreciable in any part of the range up to the intersection of the solubility curve with the critical curve at a point somewhat above the critical temperature of pure water.

On the other hand, tripotassium phosphate, from the evidence of both solubility and vapor-pressure measurements, has the properties of the salts shown in Fig. 16, a continuously increasing solubility curve which does not intersect the critical curve and large values of ΔT_s.

What does this mean in practice? For one thing, tripotassium phosphate is quite unlike trisodium phosphate, which follows the curves in Fig. 17. Trisodium phosphate is one of the worst offenders as far as hide-out is concerned in hard-pressed boilers. With tripotassium phosphate, any tendency to hide out should be entirely curbed or drastically minimized. Again, if a solution containing free alkali starts to concentrate on an overheated surface, trisodium phosphate cannot concentrate with the caustic to blunt in some measure the severity of attack, whereas tripotassium phosphate can do so. In brief, tripotassium phosphate is as advantageously unlike trisodium phosphate as the potassium silicates are advantageously unlike the sodium silicates.

Later on, it is expected that we shall have additional pressure-temperature-concentration data of a more comprehensive character, including information on the complex mixtures which represent the practical working media in all boilers.

N. B. Keevil.[25] Solubilities and vapor pressures of aqueous solutions at high temperatures are of considerable importance in interpreting the phenomena of boiler-water chemistry and in conditioning water for steam generation, as well as being essential to the understanding of hydrothermal processes responsible for ore-desposition in the earth's crust. Experimental data for soluble salts are meager, being limited to work by Morey at the geophysical laboratory, and the writer under the auspices of the

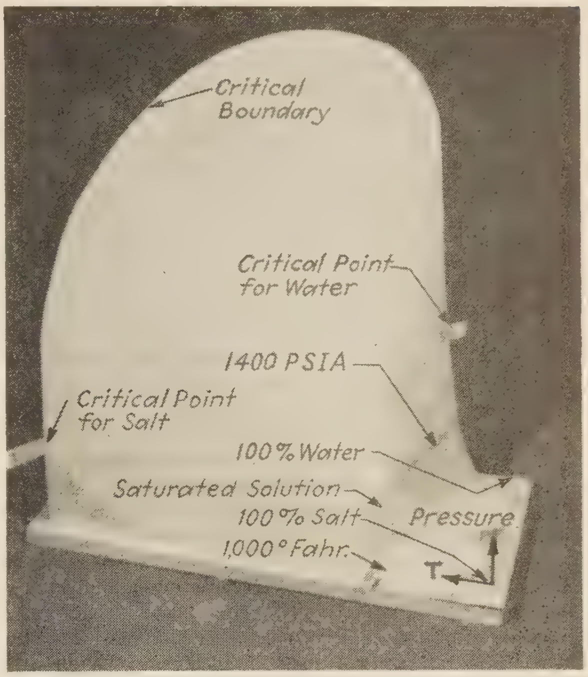

Fig. 18 Pressure-Temperature-Composition Model for an Aqueous Solution

(Data for NaI-H_2O; maximum is close to that of aqueous solutions of KCl-NaBr-H_2O and a little higher than $K_2Si_2O_5$-H_2O.)

Harvard Geophysics Committee. This work has revealed important differences in the properties of salts of potassium and sodium at the temperatures encountered industrially in boilers and in steam cycle, the halides and silicates of potassium being more soluble, and their solutions having lower vapor pressures than their sodium analogues.

The application of these differences to water-conditioning for steam generation has been developed by the author, by substituting potassium equilibria for convential sodium equilibria. Theoretically this should reduce many of the common difficulties of scale, hide-out, and sludge formation, and good practical results have been reported. The lowering of vapor pressure sufficiently to prevent precipitation is controlled largely by mol fraction, as well illustrated by the three-dimensional model for pressure P, temperature T, and composition X, shown in Fig. 18 of this discussion, and the results on various halides. This model, with pressure as the vertical axis, illustrates the remarkable effect of dissolved molecules in reducing the escaping tendency of water. In the two-component system shown in Fig. 18, the 1400-psia isobar intersects the P-T-X surface with resulting boiling and precipitation. Additional components in solution leading to a greater mol fraction would reduce the range of vapor pressure sufficiently to prevent precipitation and permit salts to remain in solution when they pass from the superheater at the full temperature of 1000 F.

The difficulty in predicting the exact conditions in the boiler and turbine is that experimental research has been of necessity, largely confined to solo solutions, while those met in practice are multicomponent systems. However, one result obtained by the writer for a two-component system has proved of considerable value in showing a marked lowering of vapor pressure below the equilibrium pressures exhibited by the pure saturated solutions of soluble salts.

There is some indication that in the competition for a place in

[24] "The Ternary System H_2O-K_2SiO_3-SiO_2," by G. W. Morey, *Journal of the American Chemical Society*, vol. 39, 1917, pp. 1173–1229.

[25] Professor, University of Toronto, Toronto, Canada.

a complex solution, the most soluble component may be disproportionately high in the final solution, so that in the K^+, Na^+, Cl^-, Br^- solution studied by the writer the composition at 874 F might have mol fractions 0.23 KBr, 0.20 NaBr, 0.06 KCl, and 0.01 NaCl rather than those extrapolated by the author from the data at 245 F. In this connection, the use of potassium may be preferable to other ions which might be considered for their greater effect in reducing vapor pressure, since the solubilities of $K_2Si_2O_5$, KBr, KCl (and NaBr) are fairly close in the neighborhood of 900 F. Furthermore, in a system containing these components, it is reasonably certain that the range of vapor pressures would be lowered enough for the materials to leave the superheater in solution.

Although critical phenomena are not ordinarily of importance in boiler problems, it should be borne in mind that there is evidence that the phenomena in the critical region are not as simple as is generally believed.

I. G. McChesney.[26] The author's work is definitely a forward step in methods of boiler-water treatment. His elaboration on the process of scale formation and control of solubilities within the concentrating film is clear and well supported. Substitution of a potassium equilibrium for the usual sodium equilibrium as a means of avoiding saturation in the concentrating film is a new and practical attack on analcite-scale formations.

The use of phosphates with sodium equilibrium, low boiler-water concentrations, and reduced alkalinities has brought trouble with analcite scales and turbine deposits.

High Boiler-Water Concentrations. The writer will cite an instance of successful water treatment in a modern high-pressure plant where the procedure closely follows an alternate method of water-conditioning suggested by the author. This plant uses from 40 to 60 per cent make-up. Lime-soda treatment of river water is followed with an aftertreatment of phosphate. Oxygen is efficiently removed by deaeration and an excess of sodium sulphite of approximately 5 ppm is used. Excess phosphates are in the order of 2 to 5 ppm.

Early in the operation of the plant, steam-washer tests were made with boiler-water concentrations up to 6800 ppm. In the final steam-washer design (tray type), priming was not appreciably increased until concentrations in excess of 5000 ppm were reached. Boiler operation at 3000 ppm was standardized after careful consideration of effect of concentration on priming over a range from 1800 ppm to 6800 ppm.

Operation with 3000 ppm total solids in the boiler water permitted an alkalinity of 11.5 without evidence of channeling corrosion. Over several years, corrosion and iron-oxide sludge have not been serious.

High concentrations and steam-washing have effectively prevented silica-scale formation on turbine blading. This is pointed to as experience in one of the makeshifts suggested by the author. His method is superior in many ways and shows good promise of overcoming many of the high-pressure feedwater problems.

Turbine Deposits. Deposition of quartz scales on the 3rd to 8th turbine stages indicates condensation of silicates. The powdery silica deposit in the 13th to 16th stages is likely formed from the repeated boiling to dryness which occurs near the dew point in the turbine expansion. In one Midwest plant this deposit in the later stages of the turbine was accompanied by embrittlement and failure of turbine blades. Maintaining the proper SiO_2/K_2O relation will likely prove effective in preventing both scale and embrittlement.

Messrs. L. E. Hankison and M. D. Baker should be commended for their task in pioneering and developing power-plant experience by the author's method.

[26] Lieutenant, U.S.N.R., Barberton, Ohio. Jun. A.S.M.E.

D. S. McKinney.[27] The troubles that accompany steam generation in high-pressure boilers may logically be divided into two groups, as follows:

1 Those that arise due to faulty design of the boiler and can be eliminated by changing the design.

2 Those that arise primarily due to the chemical properties of the water and the materials dissolved in it.

In the writer's opinion, the problems of "hide-out," corrosion due to bonded oxygen in water, and, to a considerable extent, carry-over, belong to group 1. Proper design should eliminate "hide-out" and bonded-oxygen corrosion and should considerably reduce carry-over. Operators should insist upon and be willing to pay for properly designed boilers, in order that these difficulties may not plague them for the entire life of the boiler.

Chemical treatments to alleviate troubles falling in group 1 do not eliminate the cause of the trouble and should be resorted to only to make existing boilers usable; even in these cases the operator and the manufacturer should make every effort to correct the design in situ in order to eliminate the difficulty.

The problems of magnesium scale, silica in the boiler, and, to some extent, carry-over, belong to group 2. The use of potassium salts, as proposed by the author, is a logical solution of these problems. The advantageous properties of the potassium silicates over those of the corresponding sodium silicates should be especially valuable.

The reasons which the writer has for classifying "hide-out," corrosion due to bonded oxygen, and to a considerable extent, carry-over, as due to faulty design of the boiler, are as follows: The mechanism given by the author for explaining hide-out is undoubtedly correct. However, he does not go quite far enough. The conditions necessary for maintaining the hide-out are not explained.

Consider the hide-out of sodium sulphate. The author gives its solubility at a temperature corresponding to 1800 psi as 0.0095 mol fraction and at 1400 psi as 20 per cent. These figures, expressed in parts per million, are: at 1800 psi, 70,000 ppm, and at 1400 psi, 200,000 ppm. Assuming that the bulk boiler water contains 1000 ppm of Na_2SO_4, then 98.6 per cent of the water must evaporate to reach 70,000 ppm, and 99.5 per cent to reach 200,000 ppm. If evaporation proceeds beyond these figures (at the respective pressures and temperatures), solid Na_2SO_4 will deposit. However, if the evaporating steam bubble breaks away, and a new supply of fresh boiler water arrives, the solid Na_2SO_4 will immediately begin to redissolve. Hence in order that Na_2SO_4 form scale of appreciable thickness, the rate of supply of fresh water to the tube must not exceed the rate of evaporation by more than about 2 per cent at 1400 psi, and about 0.5 per cent at 1800 psi. In other words, nearly complete evaporation must occur in one pass through the tube. Such a state of affairs can be caused only by poor circulation, or by excessive rates of heat transfer; or by both, since excessive heat transfer may cause poor circulation.

It is very likely that such a state of affairs is confined to relatively few tubes or parts of tubes in the boiler. The solution to the problem is to reduce the heat transfer by relocation of the tubes or to provide sufficient circulation to allow them to handle the heat transfer demanded.

Regarding corrosion due to bonded oxygen, and assuming the author's mechanism to be correct, it is necessary that very high concentrations of caustic be produced by the same mechanism that causes hide-out, in order to dissolve or peptize the pro-

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tecting film of iron oxide from the tube wall. Obviously, the solution of the problem is the same as in the case of hide-out. If his mechanism is not correct, and the attack is due merely to steam-binding with attendant overheating of the tube, the solution is again one of improving circulation by allowing more ready release of steam.

Finally, turning to the problem of carry-over, it appears reasonable that if the tremendous concentrations previously mentioned are permitted in the steam-generating tubes, a portion of this very concentrated water will be carried along as a mist with the escaping steam. Hence an appreciable amount of boiler solids will be carried into the superheater, even though the amount of liquid water accompanying the solids is very low. This condition, too, can be improved by improving circulation or reducing the heat-transfer rate as just suggested. In addition, the modern high-pressure boiler still suffers from many of the same defects as older boilers. Provisions are poor for separating steam from water as it comes from the steam-generating tubes. Much splashing must occur in the drums. Redesign to secure smoother steam release from the water in the drums without splashing and droplet formation, together with adequate provision for release of droplets carried with the steam, should greatly reduce carry-over troubles.

A. C. Pasini.[28] This paper offers a theory for boiler treatment which throws some light on results that have puzzled and worried power-plant managers. In this field there has been confusion and misunderstanding.

A little over a year ago, we had occasion to discontinue sodium-sulphite treatment so that a study could be made to determine where the oxygen was entering the feedwater system. (This treatment was stopped because reducing agents such as sulphite affected the accuracy of the determination of dissolved oxygen.) We were told that if the proper sodium sulphate to sodium hydroxide ratio were maintained by the addition of sodium sulphate, no trouble from carry-over should result. In less than 2 months the first-stage pressure on our turbines increased more than 10 per cent, while the plant heat rate went up 200 Btu per kwhr. Considering that the plant output was in excess of 5,000,000 kwhr daily, the loss was substantial. Further inquiry disclosed that, in other plants where the proper ratio had been maintained and with very little carry-over, both sodium sulphite and sodium sulphate were used.

Another theory which gained prominence was and is known as "total phosphate control." It is stated that proper water-conditioning will result. But this means carrying high residuals. We feel that the lower the residuals carried in the boiler water, the less chance there is for carry-over.

We have silica and iron-oxide deposits amounting to an increase in turbine first-stage pressure of approximately 2 per cent per year. On 3-year inspections we remove the deposits by hand. It is a dirty and tedious job. We want to eliminate this work.

The author states that the silica deposits will be held to a minimum or eliminated, but the residuals to be carried will be higher than ever. Are we going to get into more trouble from some other source? The chemists will have to search diligently.

In the paper, it is stated that chloride concentrations for years prevented boiler troubles on ocean-going vessels. It is unfortunate that this significant fact was not recognized sooner and applied to land boilers. What was the drawback?

If tube temperatures are 200 to 300 F above the temperature of saturation and also above the temperature of superheat, what effect has this on Δt and on the action of the potassium salts?

Nevertheless, the theory now advanced at least gives to power-plant managers an engineering appraisal of the phenomena of boiler-water equilibrium. Dr. Hall is to be congratulated.

W. C. Schroeder[29] and A. A. Berk.[30,31] Over a period of several years, and particularly since the advent of the waterwall furnace with high rates of heat transfer, there has been a disquieting but rather vague feeling that boiler water was not always acting as a dilute solution. The hiding-out of boiler-water salts, the formation of silica scale, and the appearance of intercrystalline cracks in boiler tubes all indicated concentrations much greater than normal.

The author has made a frontal assault upon this problem, proposing (*a*) what appears to be a reasonable explanation of how the dilute solution can concentrate, at least momentarily, at tube surfaces, and (*b*) the beginning of a sound analysis of the effect of this concentration upon the deposition of salts from the boiler water.

Existence of dilute and concentrated boiler water makes it necessary to analyze the effect of the various salts under both conditions. For example, Fig. 6 of the paper shows that enough sodium hydroxide to give a pH of about 11.5 (measured at room temperature) corresponds to a minimum attack upon the steel. When this dilute solution concentrates, however, the caustic will soon reach proportions that will cause the solution to attack the steel more vigorously than it would if water alone were present. In brief, the caustic added to minimize corrosion in the dilute boiler water accelerates corrosion in the concentrated boiler water. Presumably then, if the maintenance of a pH above 7 in a boiler water with caustic is still believed to be desirable, it is because (1) the area of metal exposed to the concentrated solution may be reduced to zero by proper boiler design and operation, (2) the concentration reached is not high enough to cause corrosion at a rate appreciably greater than that caused by water alone, or (3) the time during which the concentrated solution can act at a given area may be too short to cause damage.

Considering only the chemicals added for feedwater treatment, two procedures are available to eliminate the action of the concentrated caustic. The first is to operate the boiler with distilled water alone. Theoretically, this would cause general corrosion over the entire boiler surface at a rate slightly in excess of that for a boiler water at a pH of 10 to 12, but it would cause less corrosion at points of concentration. This scheme has been tried and is reported to be successful in some boilers at present. It has been unsuccessful in others, and in any event it demands that evaporators and condensers must function especially well to prevent entrance of extraneous solids. Deaeration also must be very good, or oxygen corrosion will occur.

The alternative method would be to substitute an alkaline salt, such as sodium or potassium phosphate, for caustic. This makes it possible to control the pH at the desired level, but at points on the tube surfaces that are very hot the solution cannot attain a high concentration of caustic that will rapidly attack the metal. Instead, there will be only a deposit or concentrated solution of phosphate salt. There is strong reason to believe that iron or steel is much less soluble in these salts than in caustic and, consequently, less corrosion of the tube surface would be anticipated. Maintenance of alkalinity with sodium phosphate is now being tried in several boilers operating in the range from 400 to 2000 psi. So far, these tests appear to have been quite successful. On the other hand, the corrosivity of concentrated

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[30] Associate Chemist, Bureau of Mines, Washington, D. C.

[31] Discussion published by permission of the Director, Bureau of Mines, U. S. Dept. of Interior.

solutions of sodium or potassium silicates or chlorides on steel is as yet unknown.

The author states that the decreasing solubility of sodium silicate with increase in temperature will tend to promote the deposition of insoluble silica at relatively hot surfaces. On the other hand, if the salt is potassium silicate instead of sodium silicate, the solubility characteristics are reversed, and in contact with high-pressure steam it becomes difficult if not impossible to reach temperatures high enough to drive all the water out of the solid and to deposit potassium silicate in a form in which it cannot rapidly redissolve when the surface is again wet. If the effect of the other salts that may be present in a boiler water on the solubility characteristcs of the sodium and potassium silicates at high temperatures were known, a major tool for the solution of boiler-water problems would be available. The solubility and vapor-pressure data for the potassium and sodium salts should be extended as rapidly as possible, at least to the critical temperature of water. If sodium silicate is insoluble under conditions that may exist in steam-generating tubes and may incongruently form silica scale, it is also necessary to know the limiting ratio of sodium to potassium at which this effect no longer occurs.

On the basis of solubility and vapor-pressure data, the author offers a brief explanation of how chlorides can raise the boiling temperature of the solution on an overheated surface to such a degree that the concentration of caustic required to cause embrittlement-cracking could not be attained. How effective these salts are in preventing such concentrated caustic solutions will depend upon the temperature reached at the overheated surface. With respect to the embrittlement detector, where the difference between the operating temperature in the boiler and the atmospheric boiling temperature is available to concentrate the solution, it has been found that ratios of sodium chloride to caustic of 20 to 1 had no effect in preventing cracking. This statement applies to tests conducted at pressures as low as 210 psi.

In embrittlement-testing equipment developed by Straub, where concentration of the solution occurs by loss of water vapor toward the full steam pressure, the effect of this factor may be more important. It appears also to offer a partial explanation of the different results obtained with mixtures of chloride with sodium sulphate or R_2O_3 in this unit, as compared with the results obtained in the embrittlement detector. It would seem therefore that maintaining a high chloride concentration would tend to prevent intercrystalline cracking of tubes that overheat at high ratings rather than of boiler seams and tube ends which generally do not receive direct heat.

For protecting riveted seams, the author recommends "captive" alkalinity, in which the alkalinity of the water is captured in the form of phosphates and silicates as the water is concentrated. The embrittlement detector has shown this method to be successful with phosphate alkalinity, and it has been proved that concentrated solutions of trisodium phosphate in the absence of sodium hydroxide will not cause embrittlement cracking. It has not yet been shown that concentrated solutions of potassium or sodium silicate, which may be much more alkaline, will similarly not cause embrittlement. Embrittlement-detector tests with sodium or potassium silicates are now being carried out.

Turning from the boiler, the chemist is still awaiting enough data to determine how salts are carried from the boiler water to the superheater and then into the turbine. Carry-over from the boiler water undoubtedly is a major factor, but at the high steam pressures now employed some salts may be soluble in steam. The factors that influence carry-over are not clear, and a bare beginning has been made in studying solubility in steam. In the superheater, the higher temperatures would tend to increase solubility of some salts in steam, but at the same time the lower specific gravity of the steam would tend to decrease their solubility.[32] The effect of these factors doubtless will have to be determined for each salt involved. The situation in the turbine is equally as involved, and few data are available to help in analyzing it.

Since the specific gravity of steam, or, in other words, the close packing of the molecules, appears to be the largest factor in its ability to act as a solvent for salts, it may be possible to determine the conditions under which the boiler salts could dissolve and where they might deposit in the turbine. Spillner[33] compared observed volumes of vapors at their critical points with values calculated from pressure-temperature-volume relationships and concluded that to act as a solvent a vapor must show a certain minimum departure from the gas laws. The crowding of steam molecules at high pressures can be estimated from the specific gravity and the Maxwell distribution law. It is quite probable that a mathematician could tell us at what special arrangement the molecules will be so far apart as to be able no longer to attract salt molecules sufficiently strongly to hold them in solution.

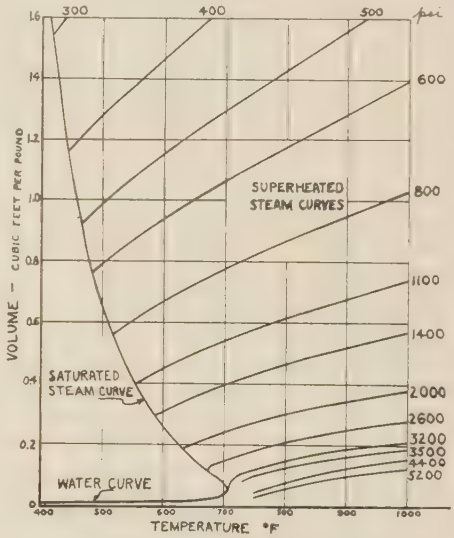

Fig. 19 Volume of Superheated Steam

As a start in this direction, Fig. 19 of this discussion compares the volume occupied by 1 lb of steam at different temperatures and pressures with the volume occupied by 1 lb of water in equilibrium with saturated steam. Saturated steam expands as it is superheated, approaching a perfect gas at lower pressures and higher temperatures. Of special significance is the region around the critical point. At about 650 F, the specific gravity of water begins to decrease rapidly. Saturated steam formed at this temperature shows a similar unusually high coefficient of expansion for a very small region on the curve. At the critical point, this effect is much more pronounced, and the specific gravity of critical steam is reduced by half during a rise in temperature of about 15 deg F. This behavior may be very significant, as the attraction of steam molecules for each other has been shown to increase very rapidly as the distance between them is decreased.[34]

[32] "The Hydrothermal Solubility of Silica," by G. C. Kennedy, Economic Geology, vol. 34, no. 1, 1944, pp. 25–32.

[33] "High-Pressure Steam as a Solvent," by F. Spillner, *Die Chemische Fabrik*, vol. 13, 1940, pp. 405–424.

[34] "Steam Research Program, Part 3," by L. B. Smith and F. G. Keyes, Proceedings American Academy of Arts and Sciences, vol. 69, 1935, pp. 285–314.

It is believed that such attraction controls the degree of solubility of salts in steam.

The author has called attention to the possible marked effect substitution of potassium salts for sodium salts may have upon turbine deposits. As yet it is difficult to associate the separation of silica as an insoluble phase with the appearance of compact crystalline masses on turbine blades. How do they get there?

Also, what is the boiler operator to do while the chemist is unwinding this complex solubility-vapor-pressure-temperature equilibrium? He can rely upon chemical treatment to correct one set of difficulties, but this may accelerate others or move the trouble from the boiler to the superheater or turbine. At the same time, a method that has been applied successfully in one plant will not necessarily work equally well in another, for factors affecting the results may be overlooked owing to lack of data or failure to understand an involved but important chemical effect

It must also be realized that the analysis of solubility and vapor-pressure equilibrium, as applied to boiler operation, has so far been tried for relatively simple solutions containing one or two compounds. Actually, most boiler waters contain half a dozen salts or compounds, each of which may alter drastically the relationships involved in the mixture. The treatment of so complex a solution to determine the conditions it can produce in a boiler, superheater, or turbine can be extremely involved, assuming that all the necessary data are available.

These statements are not intended to discourage the accumulation or full application of solubility and vapor-pressure data to the modern high-pressure boilers. Instead, they indicate that the work must be pursued vigorously and without delay to determine just how far valuable application can be made. While this is being done, there is also the chance that test methods or devices will appear or be developed that, together with a better understanding of the physical chemistry of the processes, will allow direct determination of the action of a given boiler water, or the steam from it, in boiler, superheater, and turbine. Such a development may furnish a shortcut that the boiler operator can readily interpret to test the effect of any specific set of conditions. It may also place a greater burden upon the manufacturers of boilers and auxiliary equipment to avoid trouble created primarily by design characteristics.

While the boiler operator is awaiting this further work, the one thing he can do to relieve difficulty is to reduce to a minimum the salts carried into the system through condenser leakage and evaporator carry-over. If they are at a minimum in the boiler water, it is reasonable to assume that this condition will favor the best operation of boiler, turbine, and superheater. The boiler operator who does not exercise the greatest care in this regard is missing the best opportunity for preventing trouble. Assuming that this does not overcome all his difficulties, he is then in a position to apply a minimum of chemical treatment to help solve any remaining problems.

F. G. Straub.[35] It is rather difficult to discuss this paper since the author attempts to cover the entire field of water treatment from steam-blanketed tubes, superheaters, and embrittlement through turbine-blade deposits, and offers one method of approach to solve all these difficulties experienced by the power-plant industry. As he points out, the data available are few and incomplete. Thus in Fig. 5 of the paper, he compares the solubility of sodium and potassium di- and metasilicates. The sodium salts have only two points on the curve over a range of 450 F. In a previous theory advanced by the author, he based his reasoning on the increase of solubility with increasing temperature curve of calcium carbonate, as compared to the decrease in solubility with increase in temperature of calcium sulphate. A short time later, Partridge showed that both salts decreased in solubility with increase in temperature. The writer would suggest that more data be made available before such definite conclusions can be drawn.

The real importance of the suggested potassium substitution for sodium will depend upon the results obtained in actual operation. Here again, caution should be the watchword in drawing conclusions. Often, several changes are made at one time and it becomes difficult to point out definitely which change actually caused the beneficial results obtained.

In 1936, the writer[36] called attention to the fact that sodium hydroxide would not produce a saturated solution in superheated steams, while sodium carbonate and sulphate would form saturated solutions and precipitate the dry salts under similar conditions. In 1939, we showed[37] that in steam-blanketed areas the corrosion (or as the author refers to it "embrittlement") could be controlled by chemical treatment. However, we found that if the metal temperature increased above a certain point, the chemical treatment was no longer effective. These results indicated that it would be possible to retard tube corrosion and resulting deposits as long as the tube temperature could be controlled and kept below a certain value. The author's recommended treatment should be beneficial in boilers where the steam-blanketing is such that the metal temperature is below this limiting temperature; however, if this temperature increases, the treatment will fail. It is believed that the ultimate solution of this type of trouble will involve proper circulation and wetting of the tube.

The author states that sodium hydroxide is not guilty of hide-out propensities due to its high Δt_s value. The writer knows of many boilers where sodium hydroxide is guilty of hide-out.

The statement is made to the effect that the silica deposition in the turbines is the result of the mechanical carrying of the silicates in the boiler water into the steam. A. Splittgerber[38] calls attention to the fact that dilute solutions of the silicates will hydrolyze at higher temperatures and liberate appreciable amounts of silicic acid. The silicic acid will be present in the steam even with the absence of mechanical carry-over. The deposition of the silica from the silicic acid at the lower temperatures and pressures in the turbine is a much more logical explanation of the silica deposition than that of the author.

W. L. Webb.[39] Regardless of whether the theories presented in this paper are borne out entirely in practice, this attack of the problem stimulates productive thinking and will lead the way to fundamental improvement in control technique.

Our experience with potassium treatment has been confined to its use in two boilers rated 750,000 lb per hr, 1350 psi, 950 F steam temperature at the Beech Bottom Power Company's Windsor Station since early in April, 1943. These 3-drum bent-tube tangentially-fired wet-bottom boilers are two of four boilers of approximately the same capacity which supply steam to two 60,000-kw topping turbines exhausting at 230 psi, 550 F to six 30,000-kw condensing units.

The two 1350-psi boilers under test present a number of problems considered susceptible of solution by potassium treatment. These include the following:

[35] University of Illinois, Urbana, Ill.

[36] "The Cause and Prevention of Steam Turbine Blade Deposits," by F. G. Straub, Bulletin 282, Engineering Experiment Station, University of Illinois, 1936.

[37] "Corrosion in Partially Dry Steam-Generating Tubes," by F. G. Straub and E. E. Nelson, *Mechanical Engineering*, vol. 61, 1939, pp. 199–202.

[38] "Die Flüchtigkeit der Kieselsäure," by A. Splittgerber, *Archiv. für Wärmewirtschaft*, vol. 22, 1941, p. 66.

[39] Engineering Department, American Gas and Electric Service Corporation, New York, N. Y.

1 Sludge accumulation on the steam scrubbers and driers and downtake and riser tubes.

2 Deposition of sodium-aluminum-silicate scale on heating surfaces.

3 Water-insoluble deposits in low-pressure turbines.

The sludge-and-scale problem comes about primarily from excessive condenser leakage, bringing into the system raw Ohio River water containing hardness, silica, and alumina.

Between late 1941, when the boilers were first put into service, and the time potassium treatment was started, much trouble was experienced with furnace-tube wastage. In the wastage zones, many tubes blistered, and a number developed leaks in longitudinal cracks usually located in slight bulges. On inspection these tubes showed thin hard deposits of sodium-aluminum-silicate scale. Several other tube failures occurred which were attributed to design faults.

same boiler, on May 3. These failures were in wastage areas and were repaired by window-welding. What bearing, if any, potassium treatment may have on furnace-tube wastage rates is not known.

Contrary to expectations, no leakage of gaskets in contact with boiler water has occurred. Flexitallic hand-hole gaskets are used throughout.

Riser Tube Deposits. Typical analyses of the more important constituents of deposits turbined from wall bifurcated tubes under the two treatments are shown in columns 1, 2, 3, and 4, of Table 6 of this discussion.

It was concluded that sodium-aluminum-silicate deposits not only were not forming but also that previously formed deposits were being removed by the potassium chemicals. Furthermore, at least some of the silica was being precipitated as magnesium-silicate sludge.

TABLE 6 WALL-TUBE DEPOSITS

	Sodium treatment		Potassium treatment		
Column no.	1	2	3	4	5
Date of sampling	4/5/43	3/14/43	7/24/43	6/27/43	10/10/43
Boiler no.	84	82	84	82	82
Bifurcated-tube nos.	5–6 n. wall	117–118 w. wall	5–6 n. wall	117–118 w. wall	Hard scale at failure[a]
Deposit removed:					
Grams	344	209	73	168	..
Grams per linear foot	5.5	3.3	1.2	2.7	..
Analysis, per cent:					
Sulphur trioxide (SO_3)	Neg.	...	Neg.	0.5	1.8
Carbon dioxide (CO_2)	Neg.	...	Neg.	Neg.	Neg.
Phosphorus pentoxide (P_2O_5)	7.4	13.9	19.4	17.8	7.6
Silica (SiO_2)	11.6	11.6	2.8	4.1	29.5
Iron oxide (Fe_3O_4)	53.3	20.7	30.4	26.7	24.8
Aluminum oxide (Al_2O_3)	4.2	7.0	Neg.	2.4	12.6
Calcium oxide (CaO)	6.6	8.6	15.3	15.3	1.0
Magnesium oxide (MgO)	3.6	3.6	6.4	7.1	2.2
Sodium oxide (Na_2O)	3.8	6.3	...	...	7.6
Potassium oxide (K_2O)	...	...	Neg.	...	10.1
Copper (Cu)	3.5	15.2	22.1	18.2	Neg.
Net ignition loss (calc.)	4.2	11.3	3.8	8.9	1.6
Total	98.2	98.2	100.2	101.0	98.8

[a] Hard scale taken from area of failure of Boiler 82 north wall tube after removal of soft sludge.

Inasmuch as a number of new aspects developed during October and November, 1943, in the boilers under potassium treatment, the discussion of our experience is divided into two parts:

First Six Months of Experience With Potassium Treatment

Under potassium treatment from April to September, 1943, inclusive, only two boiler-tube failures occurred, both in the

TABLE 7 SLUDGE DEPOSITS

	Sodium treatment		Potassium treatment	
Date of sampling	4/2/43	4/2/43	6/27/43	6/27/43
Sample source	Turbulent drum	Steam washer	Turbulent drum	Steam washer
Analysis, per cent:				
Sulphur trioxide (SO_3)	Neg.	Neg.	0.5	0.5 to 1.0
Carbon dioxide (CO_2)	Neg.	Neg.	Neg.	Neg.
Phosphorus pentoxide (P_2O_5)	10 to 12	30	32.4	33.0
Silica (SiO_2)	1.4	0.3	0.1	0.5
Iron oxides (Fe_3O_4 & Fe_2O_3)	35.5	15	20.7	17.0
Aluminum oxide (Al_2O_3)	...	...	Nil	...
Calcium oxide (CaO)	10 to 12	40 or more	28.1	32.0
Magnesium oxide (MgO)	2 to 3	7–8	10.2	11.0
Ignition loss	3.2	2	6.7	6.0
Copper (Cu)	33.6	Slight	...	...

TABLE 8 TURBINE-WASHING SUMMARY

	Unit no. 2	Unit no. 4	Unit no. 6
	Under sodium treatment		
Date of wash	5/24/42	5/10/42	5/17/42
Capacity restored	3000 kw	4000 kw	3000 kw
Date of wash	6/27/42[a]	11/1/42	1/17/43
Capacity restored	1000 kw	4000 kw	1500 kw
	Under potassium treatment		
Date of wash	7/11/43[b]	5/30/43	[c]
Capacity restored	1500 kw	3000 kw	...

[a] Washed just before overhaul to determine effectiveness of cleaning method.

[b] Part of previously observed capacity loss was found due to faulty governor mechanism.

[c] Unit has lost about 2000 kw capacity and is shortly to be washed.

Sludge Deposits. With substantially the same condenser leakage under the two treatments, the drum internal and visible tube surfaces were appreciably cleaner under the potassium treatment than under the sodium treatment, following operating periods of comparable steam output. This has been particularly true of the steam scrubbers and downcomer tubes, Table 7 indicating typical analyses of these sludge deposits.

TABLE 9 BOILER-WATER ANALYSES AND TREATMENT

	Sodium treatment	Potassium treatment: First month	Potassium treatment: Subsequently
Boiler water conditions:			
OH, ppm	15	5 to 15	15
PO_4, ppm	40	25 to 50	40
SO_4, ppm	150	50 to 150	150
Cl, ppm	15	50 to 100	100
Na_2SO_3, ppm	5 to 10	0	0
SiO_2, ppm	10	7 to 40	8
Total Solids, ppm	400	350 to 700	600
pH	10.8	10.0 to 10.8	10.8
SiO_2/K_2O, epm	...	0.5 to 2.0	0.5
K/Na, epm	...	1 to 2	2
Average condenser leakage (per cent of feedwater)	0.1	0.1	0.1
Average blowdown (per cent of feedwater)	0.2	0.2	0.2
Treating chemicals (Average per million pounds of feedwater):			
NaOH, lb	0.15	...	...
Na_2HPO_4, lb	0.2	...	...
Na_2SO_3, lb	0.2	...	...
KOH, lb	...	0.2	0.5
KCl, lb	...	0.3	0.5
KH_2PO_4, lb	...	0.3	...
H_3PO_4 (75 per cent), lb	...	...	0.4
Average chemical cost (per million pounds of feedwater), cents	2.6	7.6	10.0

Turbine Deposits. The relative rates of build-up of deposits in low-pressure turbines under the two treatments are confused

somewhat by the fact that these units receive some steam from sodium-treated low-pressure boilers.

The deposits became troublesome during the first half of 1942, during which time condenser leakage was unusually high and the cooling water high in suspended matter. Caustic-washing of turbines was started soon thereafter. Condenser leakage was greatly reduced but deposits continued to build up.

The washing dates of the turbines receiving steam from the potassium-treated boilers and approximate capacities regained by washing are indicated in Table 8 of this discussion.

Since July, 1943, stage-pressure data of units Nos. 2, 4, and 6 have been obtained and plotted. The curves for unit No. 6 only show progressive increases in pressures with time. Further experience is needed to show definitely the effect of potassium versus sodium on deposit build-up especially since units Nos. 1, 3, and 5, receiving steam only from sodium-treated boilers, now show slower rates of accumulation of deposits.

Boiler-Water Conditions. Typical boiler-water conditions, and chemical feed rates and costs under the two treatments are shown in Table 9.

Governed principally by changing rates of condenser leakage, the boiler-water sampling and analysis schedules are normally the same under the two treatments. Determinations ordinarily are made for OH and PO_4 five times and the other usual constituents once per 24 hr per boiler.

Conclusions. Based upon the first 6 months of experience with potassium treatment in the two 1350-psi boilers, the following tentative conclusions are drawn:

1 Indications point to complete elimination of sodium-aluminum-silicate scale and presumably all silica scales from heating surfaces.

2 All surfaces of the boiler circuits and of the steam scrubber are freer of sludge deposits than when sodium treatment was used.

3 Indications point to a reduced rate of build-up of low-pressure-turbine deposits under potassium treatment than under sodium treatment.

Eighth and Ninth Month of Experience With Potassium Treatment

During an outage of boiler 82 on October 10, one tube in the furnace wastage area of the north wall was found to be leaking, the failure being in the form of a series of longitudinal cracks in a small bulge. The 12-ft section of this tube, removed for internal examination, revealed the presence of thin hard scale on the fire side and some slight internal corrosion in the scaled areas. About half a dozen small bulges in other tubes in the wastage area in this wall were found and about half that number were observed in the east wall, a condition not previously experienced under potassium treatment. To what extent the formation of bulges can be attributed to old scale laid down under sodium treatment is not known.

The study of the thin, hard scale in the failed tube is as yet incomplete but X-ray analyses show the absence of analcite. The scale appears to be some form of potassium-aluminum silicate. Its chemical analysis is indicated in column 5 of Table 6 of this discussion.

During the November 21 outage of boiler 84, no bulges were observed on any of the wall tubes. Sludge deposits, however, were somewhat heavier than usual, due perhaps to a greater than normal time interval between shutdown and draining. The following quantities of deposits were turbined from selected bifurcated tubes of the north wall:

Tubes Nos. 5 and 6—3 g (previously turbined 7/24/43)
Tubes Nos. 17 and 18—385 g
Tubes Nos. 43 and 44—333 g

Analyses of the foregoing deposits have not yet been made.

Lacking details concerning the October-November run with potassium treatment, no conclusions are being drawn. Consideration, however, is being given to the following:

(*a*) Raising the silica in the boiler water by feeding potassium silicate to assist in precipitating magnesium as magnesium silicate for the purpose of reducing the tendency of the sludge to adhere to boiler surfaces.

(*b*) Removal of existing deposits from one of the high-pressure boilers by acid-cleaning to determine whether scales containing alumina and silicon are formed under potassium treatment.

It is possible that the scales, recently observed, were initially deposited under sodium treatment as analcite, and under potassium treatment they are being converted to a potassium-sodium-aluminum-silicate complex through zeolitic action.

Potassium treatment like all other pioneering must be modified to fit new conditions as they arise in practice. That in some respects it should meet with apparent failure is no cause for discouragement. The writer is confident that as needed vapor-pressure and solubility data become available, and the theories derived thereform are tempered by field application, boiler deposits will cease to be the problem they are today.

S. F. Whirl.[40] A new mode of thinking in the field of boiler-water treatment has been born. In diagnosing the water-steam problems encountered in high-pressure high-temperature operation, the author has utilized sound physicochemical reasoning instead of trial-and-error methods. Substitution of potassium for sodium salts is an outgrowth of this innovation. The approach is so fundamental and logical that we naturally ask ourselves the question, why did not someone conceive and apply it years ago?

Perhaps the main reason for the delay in attacking these problems in the present scientific manner has been a lack of apparent data. The author is to be complimented for the compilation and utilization of the data presented in the paper, particularly Morey's data on the behavior of silicates at high temperatures and pressures, presumably of interest only to the geophysicist and the glass manufacturer. Too often in seeking a practical solution of troubles, we overlook the findings of chemists in other fields of endeavor.

As repeatedly pointed out in the paper, the data supporting many of his contentions are very meager, and considerable research will be necessary to attain a thorough understanding of the behavior of the solids in the boiler, superheater, and turbine.

The available data deal chiefly with "solo" solutions, those in which only one chemical is dissolved in water. As other substances are added to these solutions, as in the case of actual boiler waters, the original data no longer apply, and therefore theoretical consideration of such a complex system as boiler water may lead to erroneous conclusions. This is aptly illustrated by the behavior of KCl-NaBr mixtures as compared to the behavior of these salts individually, shown in Fig. 11 of the paper.

Iron-Oxide Sludge. Corrosion of boiler metal by water is a natural oxidation process whereby iron tends to seek its most stable form under the conditions of its environment. It will continue as long as metal and water are in contact. All we can hope for is to reduce the intensity of the reaction to a negligible value by chemical treatment which favors the development and maintenance of a protective iron-oxide film on the metal surface. In the absence of dissolved oxygen in the boiler water, the reaction is fundamentally the capture of oxygen from the H_2O molecule by the iron with an unleashing of a corresponding amount of hydrogen. In fact, the amount of molecular hydrogen

[40] Chief Chemist, Duquesne Light Company, Pittsburgh, Pa.

in the steam is an indication of the intensity of the corrosion. Since caustic indirectly promotes the corrosion process by destroying the protective barrier, any efforts to reduce its concentration at the evaporative surfaces will be a step in the right direction, whether it be by the addition of other salts having high Δt_s values or by co-ordinated phosphate—pH control, as suggested by Purcell and Whirl.[41,42,43]

The presence of iron-oxide sludge in the boiler, however, does not necessarily indicate wastage of boiler metal. Iron oxide is continuously being formed throughout the entire system and is transported from the economizer, deaerator, heaters, condensers, pipe lines, etc., into the catch-all, the boiler. We must bear these facts in mind and not be misled on finding accumulations of iron oxide in the boiler at inspections, especially on starting up new units which are inclined to contain considerable iron oxide as a result of atmospheric rusting.

Boiler Water Carry-Over and Deposits in Steam System. In spite of improvements in steam quality in recent years, troubles with turbine deposits continue. This is not surprising, however, when one considers that even with steam containing less than 1 ppm of solids, the amount of chemicals entering a turbine over a period of a year is a matter of tons, not of pounds. The important item in turbine operation is whether or not these chemicals "stick and stay stuck." The extent to which operation is affected depends upon the amount of deposit and its location. Such factors as the nature of the solids carried over, temperature, pressure, velocity, pH of the steam, and turbine operation evidently all play a part. The author has supplied us with a very important piece of this complex jig-saw puzzle. Since silica is the chief offender in high-pressure operation, he suggests a method of boiler-water treatment designed to keep the silica in solution as it passes through the turbine. It must be recognized, however, that the supporting data deal only with solutions, and the effect of other salts may completely alter the behavior of the silica. This can be determined only by experimental investigation.

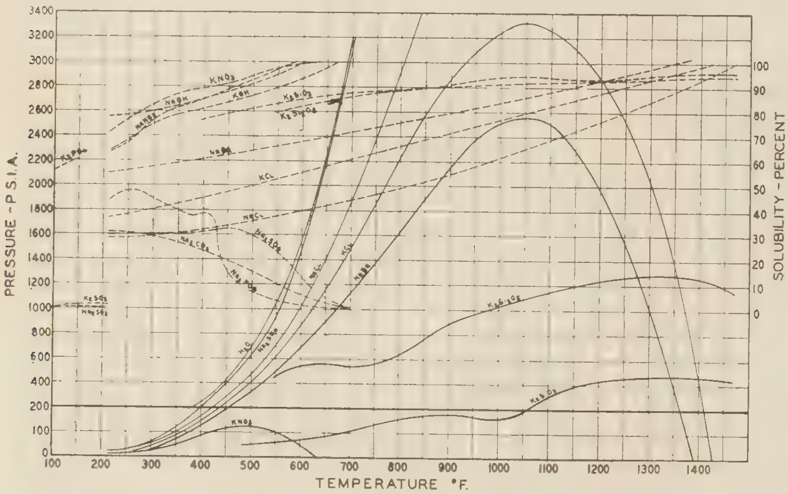

Fig. 20 Solubility-Temperature-Vapor Pressure Relations for Water Solutions of Various Salts

Is keeping the silica in solution the complete answer, though? Let us consider a droplet of boiler water carried over with the steam. By the time it has passed through the superheater and entered the turbine, certainly it will be a concentrated solution of some chemicals. Some salts will precipitate and their behavior is a matter of conjecture. Will they settle out, or will they remain with the concentrated droplet? Most important, however, will the final droplet be a liquid which will pass on through the turbine or will it be a viscous, pasty, wet mass which will adhere to the blading?

To give an idea of the extent of the solubility of different salts under superheater and turbine conditions, some of the solubility and pressure-temperature data in the appendix have been plotted in Fig. 20 of this discussion. By means of these curves the weight per cent of the salt in solution is given directly for various pressure-temperature conditions. It is interesting to note that at 1000 psi and 900 F, a number of the solutions are richer in chemicals than in water; for instance, 70 per cent KCl, 30 per cent H_2O; 100 per cent KOH; 91 per cent K_2SiO_3, 9 per cent H_2O. We would naturally expect a mixture of these to be quite viscous.

There are many questions concerning the behavior of silica which cannot be readily explained. The silica concentration in the saturated steam at one of our plants operating at 900 psi has been found, by careful evaporation studies, to be more than tenfold that which would be expected. Is this indicated selective carry-over due to the solubility of silica in the vapor or to mechanical carry-over, promoted by the tendency of silicates to concentrate at the steam-water interface?

Another question is prompted by the E.E.I. investigation of several years ago,[44] and the comprehensive study now under way. A number of cases are on record where no silica difficulties were encountered in the low-pressure turbines until after topping units were installed. In general, the quality of the steam has been improved. In some instances, the treatment in the high-pressure and the low-pressure boilers has been essentially the same. The tendency has been to carry lower alkalinities in the high-pressure boilers, but there are a few cases where the reverse is true. There is one principal difference, however, that is, the higher superheat temperature to which the droplets of boiler water are subjected in the high-pressure superheater. Would we expect this factor to have any particular influence on the behavior of silica in the turbine, using sodium treatment in the boiler water? A phase-rule study similar to that presented in Fig. 8 of the paper, but including all of the salts present, would most certainly determine the conditions under which the various constitutents would precipitate from solution. Whether or not and where they may "stick and stay stuck" are different problems.

The same phase-rule study would apply to deposition of solids in the boiler and might correlate many experiences which now seem contradictory. As an example, after several months'

[41] "Embrittlement of Boiler Steel—Experiences With the Schroeder Detector," by T. E. Purcell and S. F. Whirl, Trans. A.S.M.E., vol. 64, 1942, pp. 397–402.

[42] "Protection Against Caustic Embrittlement by Co-ordinated Phosphate-pH Control," by S. F. Whirl and T. E. Purcell, Third Annual Water Conference, Engineering Society of Western Pennsylvania, 1942, pp. 45–60.

[43] "Protection Against Caustic Embrittlement by Co-Ordinated Phosphate-pH Control," by T. E. Purcell and S. F. Whirl, Trans. Electrochemical Society, vol. 83, 1943, pp. 279–295.

[44] "Power Station Chemistry," Edison Electric Institute, Publication No. E 10, 1937

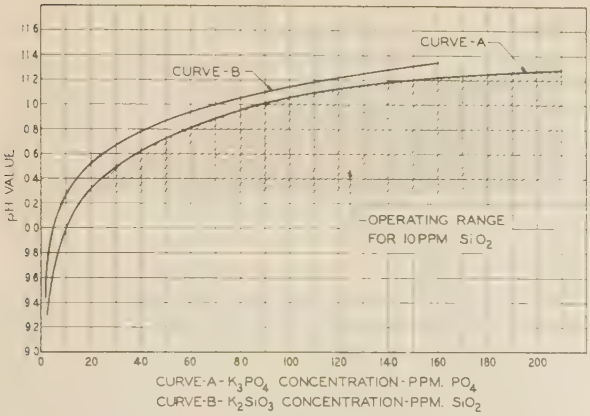

Fig. 21 pH Values of Water Solutions of Tripotassium Phosphate and Potassium Metasilicate

operation with co-ordinated phosphate-pH control in two 300-psi B&W cross-drum boilers, the silica-rope scale, always found tightly adhering to the sides of the top rows of generating and upper steam-circulating tubes, was observed to be loosening, and many pieces, several inches in length, were flaked off and lying on the bottoms of the tubes. This observation is offered for what it may be worth and not as a recommendation of a method of treatment for removing silica.

Caustic Embrittlement. At the last two Annual Meetings of this Society, the corrective measures offered for the prevention of caustic embrittlement, when the boiler water was treated with sodium salts, included inhibitors and our co-ordinated phosphate-pH control of alkalinity,[41,42,43] the latter being especially suited to high-pressure operation. That the principle of our method might apply to other salts of weak acids and strong bases was fully recognized. The author terms this application "captive alkalinity." However, in utilizing the potassium silicates for this purpose, as suggested by the author, the question logically arises, can these salts in themselves promote embrittlement, especially the potassium metasilicate, because of its high solubility and high Δt_s value? The reason why pure trisodium phosphate will not cause embrittlement is evidently due to its low Δt_s value with consequent insufficient alkalinity in its saturated solution to promote this type of attack. We would expect much higher alkalinities from a saturated potassium-metasilicate solution.[45]

Since no two boilers have the same characteristics with respect to concentrating-film boiling at localized areas, and since the addition of other salts like KCl with high Δt_s values are not effective when the concentration is produced by evaporation to atmosphere, the only assured protection against caustic embrittlement is the elimination of caustic as such from the boiler water. The method of treatment to be described is offered as a practical means of protection against caustic embrittlement and, at the same time, a possible means of holding the silica in check.

Additional experimental data will be necessary to determine its effectiveness with respect to the latter. In Fig. 21 of this discussion are plotted separately the pH values of solutions of K_3PO_4 and K_2SiO_3. It is suggested that the pH of the boiler water be maintained below the phosphate-pH curve and above the pH value corresponding to the silica concentration in the boiler water, assuming the silica to be present as K_2SiO_3. The operating region for 10 ppm SiO_2 in the boiler water is indicated in Fig. 21 of this discussion.

[45] "Silicate of Soda in the Building Industry," by J. A. Vail, *Industrial and Engineering Chemistry*, vol. 27, 1935, pp. 888–893.

It is realized that, in the light of the data presented in Fig. 8 of the paper, this method may not appear logical with respect to preventing silica deposits. However, it should be pointed out that these data deal only with silicates and caustic potash; we do not know what effect the phosphates will exert on the equilibrium, especially when the pH is below that corresponding to the K_3PO_4 concentration. We would expect the Δt_s values of the phosphates to be higher with lower pH, and as the Δt of the solution increases and the vapor pressure decreases, what effects will the phosphates have on the silicate equilibrium? The nature of the phosphate salts which separate will also have a bearing on the ultimate behavior of the silica. For the complete picture, we must go still further and consider what happens to the concentrated boiler-water droplets as dilution occurs with lowering temperature in the turbine in an environment of various dry salts, having varying degrees of solubility at the different temperatures.

Potassium treatment was started in the 900-psi boilers of our Frank R. Phillips Station on August 1, 1943. The boiler-water condition is summarized in Table 10 of this discussion.

TABLE 10 AVERAGE ANALYSIS OF BOILER WATER; NO. 1 BOILER, F. R. PHILLIPS POWER STATION; AUGUST 1 TO NOVEMBER 20, 1943

pH: Determined	10.59
From phosphate-pH curve	10.86
Sp. Cond., micromhos	625
Dissolved salts, ppm	374
Phosphate, PO_4, ppm	87
Sulphate, SO_4, ppm	138
Sulphite, SO_3, ppm	11
Chloride, Cl, ppm	10
Hydroxide, OH, ppm[a]	0
Silica, SiO_2, ppm	8

[a] By modified Winkler method.[41]

Since inspections have not been made in the interim, no information can be offered concerning the effect of the treatment on boiler, superheater, and turbine deposits.

An embrittlement-detector test, however, was terminated at the end of 58 days instead of our usual 90 days, for the purpose of submitting the information in this discussion. Neither the hot- nor the cold-rolled specimen cracked. It is noteworthy that the deposit accumulations on the specimens were much softer and more readily dissolved in water than those of any previous test, using sodium treatment in the boiler water.

It is apparent that this paper opens a new and wide field of research in boiler-water treatment, and we may be certain that the method of approach will serve to guide others in future study.

Author's Closure

The constructive character of the discussion is outstanding. Only a few points need be given further consideration.

Mr. Corey's impression regarding higher hide-out by potassium than by sodium salts must be reversed. *Less* hide-out occurs with potassium than with sodium salts.

Mr. Daniels, who completely eschews any treatment of the boiler water, ably supports his position by the excellence of results in his plants. By maintaining his condensers tight, and avoiding carry-over from his evaporators, he is in position to say "Go thou and do likewise" to anyone objecting to his methods, because these sources may provide impurities in the boiler water. Many, however, conservatively sanction the added insurance embodied in a second defense line maintained in the boiler water, no matter how impregnable they may consider their primary defenses to be. As pointed out in the section on "The Iron Oxide Sludge Problem," the low pH value of untreated boiler water "provides optimum conditions for silica deposition in boiler and turbine." Also, where salt water is used in condensers, in addi-

tion to more severe problems of corrosion, the operator is confronted with a hundredfold or more increase of dissolved impurities over that derived from the same per cent of inleakage of typical fresh water.

Professor Keevil emphasizes the dearth of solubility and vapor-pressure data on multicomponent systems at temperatures and pressures such as those met in boilers and turbines. Continuance of the work discussed by Mr. Kaufman will in time result in obtaining such data, and probably with results as unexpected as the totally unlike solubilities of sodium and potassium phosphate.

It is pleasing to note that Dr. Schroeder and Mr. Berk, in their discussion, by use of the principles set forth in the section on "Protection From Embrittlement in the Light of the Δt Function" arrive at a logical explanation of the differences in results obtained by use of the embrittlement-testing equipment developed by Straub, on the one hand, and, on the other, by use of the embrittlement detector of Schroeder and Berk.

Professor Straub notes in relation to the tentative solubility curve of sodium meta- or disilicate (Fig. 5) that there are "only two points on the curve over a range of 450 F," and that he "would suggest that more data be made available before such definite conclusions can be drawn." Relative to this solubility, Dr. George W. Morey, of the Geophysical Laboratory, Carnegie Institution, Washington, D. C., states (14): "Experiments at comparatively low temperatures have shown that the solubility of sodium metasilicate is strongly retrograde. For example, at 100 C, the saturated solution of sodium metasilicate in water contains about 50 per cent of the anhydrous compound, but at 350 C, the solubility is measured only in tenths of a per cent." The broken curve of Fig. 5 for sodium metasilicate has been drawn in accordance with these data.

Professor Straub states, "The statement is made to the effect that the silica deposition in the turbines is the result of the mechanical carrying of the silicates in the boiler water into the steam." On the contrary, both in this paper (Section on "Role of Silica in the Turbine") and in the A.S.T.M. Round-Table Discussion of 1942 (43), the author has taken the position that the question of mechanical transfer, on the one hand, or transfer by vaporization, on the other, is an open one whose answer must await further data.

Professor Straub states, "We showed that in steam-blanketed areas the corrosion (or as the author refers to it 'embrittlement') could be controlled by chemical treatment." Reference to the section of the paper entitled, "The Iron Oxide Sludge Problem: Corrosion by the Bonded Oxygen of the Boiler Water," fails to disclose any mention of embrittlement in connection with this type of corrosion. However, in some cases, a type of embrittlement develops under such conditions, as noted in the section "Protection From Embrittlement, etc."

As brought out in Mr. Webb's discussion, and particularly by analysis No. 5, Table 6, potassium aluminum silicate, analogous to sodium aluminum silicate, will form under severe conditions. The conclusion is thus inescapable that with either sodium or potassium conditioning of the boiler water, the concentration of aluminum therein must be held to the minimum possible.

The author is very appreciative of the time and thought which has been given by the several contributors in the development of this discussion.

Theoretical Regenerative-Steam-Cycle Heat Rates

By A. M. Selvey,[1] Pittsburg, Calif., and P. H. Knowlton,[2] Schenectady, N. Y.

The calculation of heat rates for a theoretical steam cycle with an infinite number of heaters regenerating feedwater to throttle saturation temperature is facilitated by the development of a simple tabular-integration method of computation which takes into account boiler-feed-pump work. A table of theoretical heat rates is presented for throttle-steam conditions, ranging from 300 to 3200 psia, and from saturation temperature to 1200 F, and for an absolute exhaust pressure of 1 in. of mercury. Means are provided for calculating heat rates at other exhaust pressures within the wet-steam region. These heat rates serve two purposes, i.e., (1) as a standard of power-plant performance; and (2), as a first step in estimating regenerative-cycle economy at unfamiliar steam conditions. The derivation of factors necessary to complete economic estimates, however, must await a more favorable opportunity. In the meantime, existing plant records can be made to provide approximate factors as substitutes for those it is hoped will be computed and published later. The economy of using superheated steam, integral moisture extraction, etc., are discussed briefly. An Appendix provides full directions for using or adapting the tabular-integration method for steam-cycle studies.

FOR several years a need has existed for a simple method of calculating theoretical regenerative-steam-cycle heat rates to serve as a standard of comparison for steam-power-plant performance. Such a standard has been set for the simple Rankine cycle by publication at various times of theoretical steam rates.[3] This paper presents a similar set of tables for a hypothetical regenerative cycle as described hereinafter.

Many expositions have been made by various writers of means for determining the performance of regenerative cycles, but it appears to the present authors that such performance is best obtained in two main steps, namely, (1) the determination of theoretical heat rates with zero engine and heat losses; and (2) the use of factors of "efficiency" for the various actual pieces of the working-cycle apparatus, to reduce the theoretical heat rates to practical cases. Information concerning step (1), namely, the production of the necessary tables of theoretical performance, is contained in this paper, with a description of the method and a set of tables showing the resulting heat rates, which are proposed as an acceptable standard, depending only upon the properties of steam, and therefore not subject to change unless, as appears unlikely, the basic steam tables are revised.

Step (2), the delineation of factors of actual performance, will be concerned with evaluating the effects of such losses as occur in actual apparatus—engines, boilers, feed heaters, pumps, etc., and in their arrangement. Work on step (2), however, has been postponed because of prior demands by the war effort. It is hoped that the work will be executed later. In the meantime, with the theoretical heat rates completed, there seemed good reason for making this material available for use in engineering studies. Estimating plant heat rates for higher steam conditions than are now in general use can be accomplished with reasonable accuracy by computing a series of proportionality factors or ratios from the performance of an actual plant and applying them in turn to the new conditions. This procedure should prove of some use until time is available for work on step (2).

Numerous interesting problems arose while work was proceeding on this study. Neither time nor space, however, will permit more than one or two of these secondary topics being introduced and discussed. Accordingly, this paper is confined to defining and describing the chosen theoretical regenerative cycle, discussing the mathematical procedure for computing heat rates, some brief mention of possible applications for the data, and some views on two recent proposals concerning the regenerative cycle. An Appendix demonstrates two computation forms, a short and a long, which may be adapted as desired to suit various modifications of the regenerative cycle.

The Theoretical Regenerative Cycle

A clear picture of the theoretical regenerative cycle considered in this paper is essential as several writers have proposed and discussed different modifications which also can be classified strictly as regenerative cycles. In addition, engineers still lack a universally acceptable steam-cycle terminology and, consequently, have resorted to naming cycles as they personally feel justified. Under such circumstances, definitions, descriptions, and diagrams need to be complete.

Definition. The theoretical regenerative steam cycle is viewed herein as one in which steam, expanded isentropically (at constant entropy) through a prime mover, in this case a turbine, is extracted (bled) for heating feedwater to throttle saturation temperature in an infinite number of progressive steps of zero terminal difference, each assisted by an attendant boiler feed pump of 100 per cent efficiency powered from the prime mover.

Other terms sometimes used instead of "theoretical," in speaking of such a cycle, are "ideal," "perfect," "hypothetical," etc. The term "ideal" is used often to indicate 100 per cent mechanical efficiency, reversibility, and possibly the idea of the cycle being "imaginary." The designation "ideal cycle," however, has been applied also to regenerative cycles of less than an infinite number of feedwater heaters and to a cycle in which only saturated steam and wet steam are bled for feedwater regeneration. Whatever the cycle is called, a more lengthy description is desirable.

Description. The heat rates provided by this paper are for a regenerative steam cycle of the following component parts:

1 A turbine, having no mechanical losses, through which steam is expanded at constant entropy (engine efficiency is 100 per cent).

2 A regenerative feedwater system having an infinite number

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[1] Chief Engineer, Shell Point Plant, Shell Chemical Division, Shell Union Oil Corporation; formerly Engineer, Engineering Division, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

[2] Engineer, Turbine Engineering Department, General Electric Company. Mem. A.S.M.E.

[3] See, for example, "Theoretical Steam-Rate Tables," by J. H. Keenan and F. G. Keyes, published by The American Society of Mechanical Engineers, New York, N. Y., 1938.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

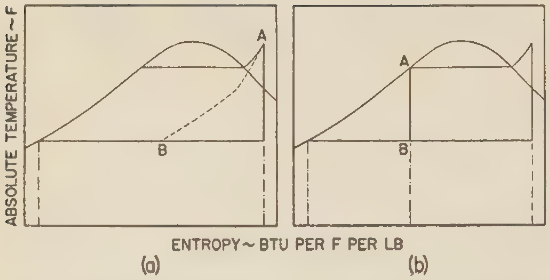

Fig. 1 Temperature-Entropy Diagram for Theoretical Regenerative Steam Cycle, Operating Between 1000 Psia 700 F and 1 In. Hg Steam Conditions

(*a*, Conventional diagram. *b*, Diagram based on construction method described in reference 5.)

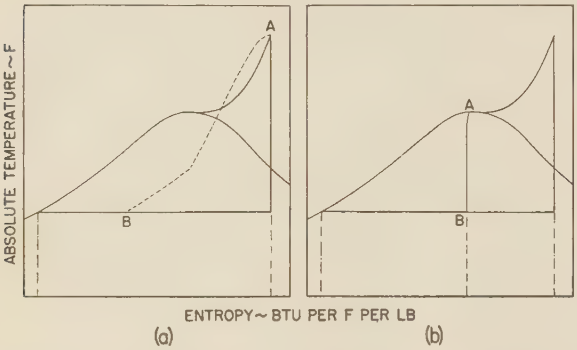

Fig. 2 Temperature-Entropy Diagram for Theoretical Regenerative Steam Cycle, Operating Between 3200 Psia 1200 F and 1 In. Hg Steam Conditions

(*a*, Conventional diagram. *b*, Diagram based on same construction method as used for Fig. 1*b*.)

of bled-steam heaters heating to the saturation temperature corresponding to the throttle-steam pressure, and with zero terminal difference between the saturation temperature of the bled steam and the temperature of the feedwater leaving the heater, even when superheat is present in bled steam,[4] and an attendant feed pump of 100 per cent efficiency with each heater to step up the feedwater pressure to the level of the next higher heater.

3 An electric generator of 100 per cent efficiency, which supplies, without line loss, the boiler feed pumps.

4 A steam generator of 100 per cent efficiency, in which blowdown, soot-blowing losses, etc., are zero.

Since energy required to drive fans, fuel equipment, general services, etc., is considered zero in a theoretical cycle, auxiliary-power usage other than that required for the boiler feed pumps, as described in item (2), is not included.

Diagrams. A conventional diagram for the theoretical regenerative cycle is shown in Fig. 1(*a*) for throttle-steam conditions of 1000 psia 700 F and 1 in. Hg exhaust pressure. The dotted line *AB*, representing the reduction in weight of steam passing through the turbine, is not an expansion line. The area below the dotted line is not significant. The chief advantage of the diagram is that it appears familiar.

In attempting to make a diagram in which areas are significant and bear the correct relationship of cycle efficiency, recourse was taken to a method of construction described fully by Barnard, Ellenwood, and Hirshfeld.[5] This method was used to produce Fig. 1(*b*) which is drawn for same steam conditions as Fig. 1(*a*). In Fig. 1(*b*) the area below the saturation line, bounded by condensate entropy and line *AB*, represents heat given up to feedwater.

With high-pressure and high-temperature throttle steam conditions, such as 3200 psia 1200 F, these diagrams, however, undergo unusual changes as are indicated in Figs. 2(*a*) and 2(*b*). Parallelism between line *AB* and the isentropic expansion line, the apparently meritorious feature of Fig. 1(*b*), no longer extends over the whole length of *AB*, in Fig. 2(*b*), but exists only up to the temperature level where the expansion line crosses the saturated-vapor line. Above that point, line *AB* shows to a more or less pronounced degree the effect of bleeding superheated steam. Curvature of line *AB* in Figs. 1(*b*) and 2(*b*) actually will exist whenever superheated steam is bled, but it is only noticeable on a small-scale diagram with throttle steam relatively highly superheated. Even then, curvature is apparently so slight that casual inspection of Fig. 2(*b*) might lead to the contention that a simple formula could be developed for calculating theoretical regenerative-cycle heat rates even for top throttle conditions. The significance of this curvature in line *AB* of Fig. 2(*b*) is not easily apparent, and it is hoped that the discussion will bring forth an explanation.

The closest approximation by "short-cut" formula would obtain if a straight line could be drawn vertically through point *B*. But the location of *B* cannot be found without recourse to the integration method to be described later. Accordingly, no labor can be saved in this case. The next approximation would be based on the assumption that the straight line passed vertically through known point, *A*. A short-cut formula for this case would be

$$\text{Theoretical heat rate}^{6}\ \text{(inaccurate)} = \frac{3412.75\,(H_t - h_f)}{H_t - h_f - T_e(S_t - S_f)}$$

[4] In actual practice, where a desuperheating zone is provided in a feedwater heater, a temperature crossover or negative terminal difference may exist where superheated steam is being bled. Since the number of degrees crossover varies with heater design and the degrees of superheat, an acceptable allowance for this effect feasibly cannot be included in a theoretical study.

[5] "Heat Power Engineering," by W. N. Barnard, F. O. Ellenwood, and C. F. Hirshfeld, John Wiley & Sons, Inc., New York, N. Y., third edition, part 1, sec. 217, 1926, pp. 353–356.

[6] An approximately similar incorrect equation will be found in connection with Fig. 712, of "Steam Power Plant Engineering," by F. G. Gebhardt, sixth edition, John Wiley & Sons, Inc., New York, N. Y., 1925.

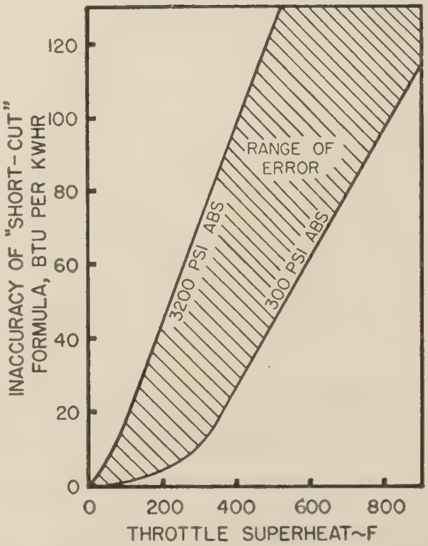

Fig. 3 A Simple or "Short-Cut" Formula for Determining Approximate Theoretical Regenerative-Cycle Heat Rates Produces Results Which Are Too Low

(The diagram demonstrates how this inaccuracy increases with the amount of superheat in the throttle steam.)

where subscript t refers to throttle, f to saturated feedwater, and e to turbine-exhaust conditions. This formula is incorrect because it fails to take into account the sizable vertical strip bounded on one side by the line AB and on the other by a vertical through A. Heat rates computed by the short-cut formula are too low. The extent of the error is shown in Fig. 3.

The theoretical regenerative cycle, as defined and described, has been considered commonly as representing the best economy actually approachable in practice and, on this score, has been viewed by many as a standard of attainment. A recent suggestion, to be described later, of a novel but probably impractical regenerative cycle having Carnot efficiency does not seem to detract from the already established regenerative-cycle standard. The fact that a simple computation method, which will be described next, has been developed and heat rates calculated for a wide range of steam pressures and temperatures should minimize objections on the grounds of complication to the continued use of the theoretical regenerative cycle as an acceptable standard.

Development of Computation Methods

The Ernest L. Robinson equations for calculating the theoretical weight and energy value of bled steam appear in the Appendix of his 1923 paper.[7] The heat balance for an infinitesimal heater is

$$dw(H - h) = (1 + w)dh$$

which, when integrated, gives

$$1 + w = e^{\int_{h_2}^{h_1} \frac{dh}{H-h}}$$

where $(1 + w)$ is the pounds of steam at any point in the turbine. In the 1923 paper,[7] with a maximum steam condition of 1000 psia 800 F, integration was performed graphically. (The nomenclature used appears in the Appendix.)

Twenty years later, attention is focused on much higher steam conditions. There appears a definite need to provide theoretical regenerative-cycle heat rates for a wide range of throttle pressure and temperature up to 3200 psia and 1200 F. Because the effect of pump work on cycle efficiency is appreciable at high feedwater pressure, and because of the larger number of heat rates that would have to be computed to cover the greater range of steam conditions, Robinson's equation[7] had to be modified and adapted to a tabular method of computation.

Step-by-Step Integration. If $dh/(H - h)$ were to be integrated graphically, the quantity $1/(H - h)$ would be plotted against h as shown in Fig. 4. The product $dh/(H - h)$ is the area of an infinitesimal strip below the curve AB. The integral of $dh/(H - h)$, by the Robinson equation, is the whole area $ABCD$. Some energy, however, is added by the boiler feed pump. For each strip, dh, representing heat added by bled steam, there is an attendant strip, dF, representing the boiler-feed-pump work. Since the steam flow in the turbine depends upon the amount of steam bled, the enthalpy added by the pump in each step must be subtracted from the gross gain in enthalpy.[8]

In order to calculate accurately rather than to planimeter the area below the curve, it is necessary that sufficiently narrow strips be so chosen that the section of curve bounding the top of each strip is essentially a straight line. This provision permits determining the area of the strip by taking the equivalent height as the average of the two sides. About 35 steps from throttle to condenser for steam pressures of 2000 to 3200 psia have proved adequate. (An extra 20 steps were found unnecessary.) The steps, however, must be chosen so that the area of each strip, i.e., $dh/(H - h)$, is the same. An initial selection with constant increments of h may be proportioned to give constant values of $dh/(H - h)$ on a second or third trial. The method is described in the Appendix under "Short-Form Computation." With throttle pressure of 1000 psia or less, 25 steps may be chosen, and equal values of dh are satisfactory.

Checking the Method. Only one independent check on the method is available. A theoretical regenerative cycle supplied with saturated steam will have Carnot-cycle efficiency. Checks made on this basis have proved the method to be reliable.

[7] "The Margins of Possible Improvement in the Central-Station Steam Plant," by Ernest L Robinson, Trans. A.S.M.E., vol. 45, 1923, p. 644.

[8] The work done by the perfect boiler feed pump should be read from Fig. 3, p. 75, of the Keenan and Keyes "Thermodynamic Properties of Steam," John Wiley & Sons, Inc., N. Y., 1936. In the tabular-integration method, however, steps are so small that Fig. 3 cannot be read accurately enough. Since the change in specific volume of compressed water is insignificant on three scores, namely, (1) the change in feedwater temperature during compression is small, (2) the change in pressure is small, and (3) the effects of the foregoing oppose each other, the simple equation $F = Nv/5.4$ proved satisfactory. This form substitutes for the correction

$$-\frac{1}{J}\int_{r-1}^{r} v_w dP$$

recently suggested by Dr. W. J. Kearton; see *Journal* and Proceedings of The Institution of Mechanical Engineers, vol. 147, 1942, pp. 114–115, comment on paper by H. S. Horsman, reference 9.

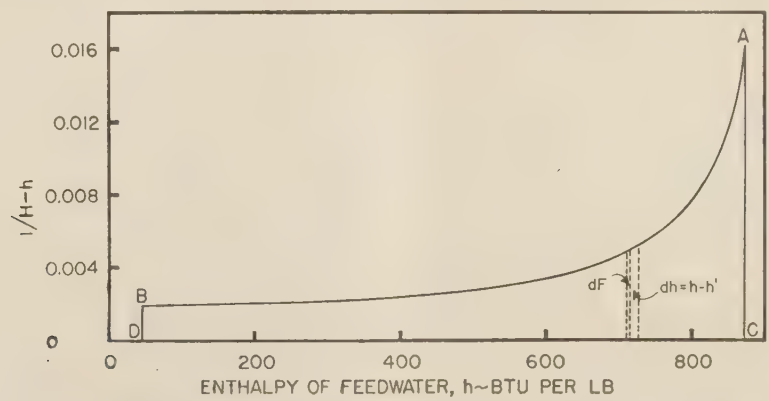

Fig. 4 Graphical Representation of Robinson's Equation $1 + w = e^{\int_{h_2}^{h_1} \frac{dh}{H-h}}$ Determining Weight of Steam Bled in a Theoretical Regenerative Cycle

(Example is for 3200 psia saturated steam.)

Application to Modified Regenerative Cycles. The step-by-step integration can be used to compute heat rates of other theoretical cycles such as the moisture-extraction cycle, the isothermal compression regenerative cycle, etc. It may be applied also to theoretical cycles where dry-stage efficiencies less than 100 per cent have been taken into account. In some cases, the second Robinson equation

$$E = \int_{h_2}^{h_1} \frac{H_1 - H}{H - h} (1 + w) dh$$

will have to be used. The method of handling this equation is explained in the Appendix under "Long-Form Computation."

To those interested in studying theoretical cycles, the tabular-integration method should prove a useful tool. The theoretical regenerative-steam-cycle heat-rate table, given in this paper, is a typical application of the method.

Theoretical Heat-Rate Tables

The production of a theoretical heat-rate table is a straightforward enough operation, although some ingenuity is required to minimize the necessary work to be done. Applying the short-form method, described in the foregoing discussion and illustrated in the Appendix, heat rates were calculated for a number of "key points," where throttle pressure and temperature are represented by round numbers. Temperature intervals were taken every 100 F; pressure intervals varied from 100 to 300 psia to suit circumstances.

Heat rates thus computed were plotted on a large curve sheet. A cross-plot also was used. The combination of the main and cross-plots established the principal connecting links between key points. Work sheets consisting of large skeleton tables were made up from the heat-rate readings given by the curve sheet. The skeleton was adjusted slightly where obviously necessary and eventually filled in mathematically by a system of uniform progressive differences. The application of this system to the data on the work sheets minimized errors of interpolation.

Table 1 gives theoretical regenerative-cycle heat rates at 1 in. Hg abs exhaust pressure for steam pressures from 300 to 3200 psia

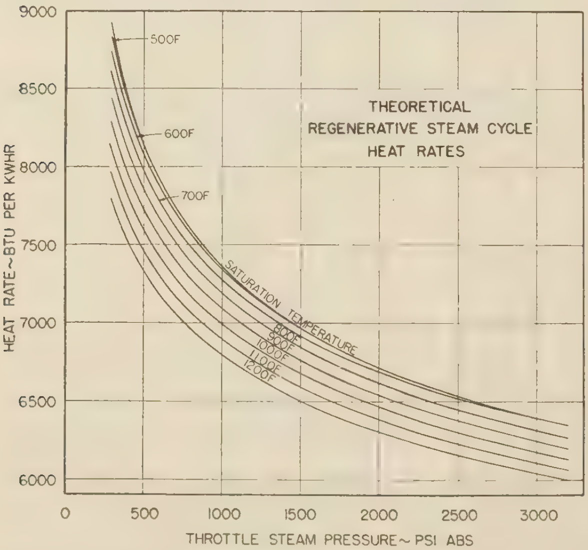

Fig. 5(a) Theoretical Regenerative-Steam-Cycle Heat Rates

(Throttle steam conditions as noted; exhaust pressure = 1 in. Hg abs. Infinite number of bled-steam feedwater heaters and attendant boiler feed pumps. Zero terminal difference. Feedwater heated to throttle saturation temperature.)

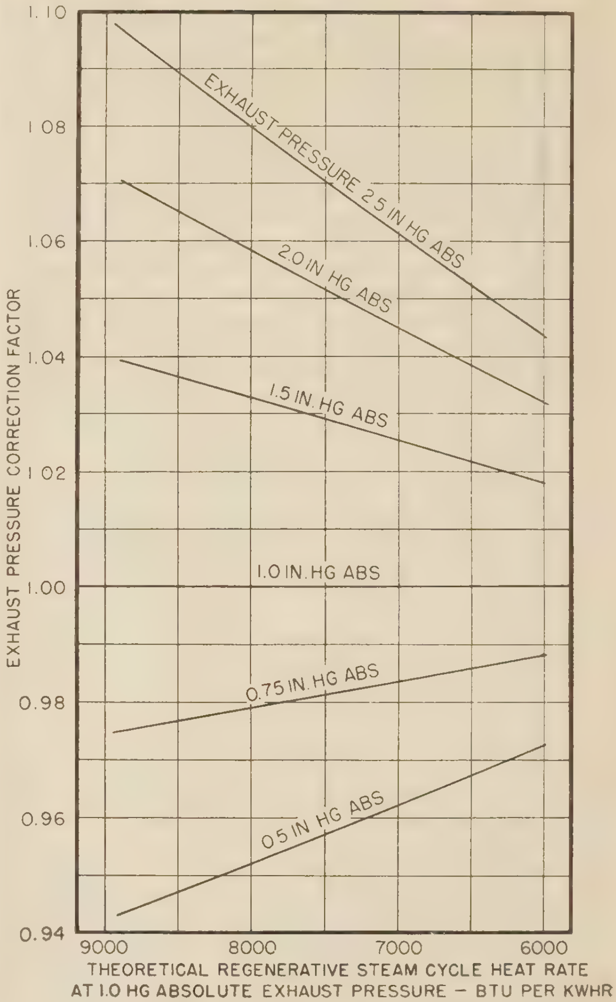

Fig. 5(b) Exhaust-Pressure Correction Factor

(To obtain the heat rate, HR_T, at a particular exhaust pressure within the wet-steam region: [1] Read the heat rate, HR_0, from Table 1 or Fig. 5a for 1 in. Hg abs exhaust pressure; [2] find the correction factor, F, from Fig. 5b corresponding to HR_0 and the new exhaust pressure; [3] multiply HR_0 by F to get HR_T. See revised Fig. 5(b) on page 511.)

and from saturation temperature to 1200 F. The selection of pressures and temperatures for which heat rates are given was based (1) on the steps used in the 1936 steam tables and (2) on the need to insure that proportional interpolation between readings would be accurate. The table provides a means of determining quickly the heat rate of any theoretical regenerative-cycle power plant.[9] Theoretical heat rates at 1 in. Hg abs exhaust pressure are shown also in Fig. 5(a).

Exhaust-pressure corrections are based upon the fact that, within the wet-steam region, the theoretical regenerative cycle has Carnot efficiency, from which it follows that, with constant throttle conditions, the heat rejected is proportional to the abso-

[9] A comprehensive paper titled, "The Regenerative Cycle: An Efficiency Basis Having Special Reference to the Number of Feedwater Heating Stages," by H. S. Horsman, was published in the *Journal* and Proceedings of The Institution of Mechanical Engineers, vol. 146, 1941, pp. 5–19. In the foregoing paper, however, feedwater temperature is limited to a maximum of 460 F, corresponding to 467 psia saturation pressure, and hence provides little opportunity to check data given here in Table 1, where final feedwater temperatures may be up to 705 F.

lute temperature of the exhaust. The generation of A Btu of useful work (say 1 kwhr or 3412.75 Btu) requires a heat input of HR_0 (the theoretical heat rate in Btu per kwhr given in Table 1), followed by the rejection of $(HR_0 - A)$ Btu at T_0 exhaust temperature. If the exhaust temperature is reduced by ΔT to T, then additional useful work for the same heat input is obtained, amounting to

$$\frac{\Delta T}{T_0}(HR_0 - A)$$

As the heat rates of the cycle at constant steam conditions, but varying exhaust temperatures, are inversely proportional to the amounts of useful work generated, a heat-rate correction factor F may be obtained from the equation

$$F = \frac{HR_T}{HR_0} = \frac{A}{A + \frac{\Delta T}{T_0}(HR_0 - A)} = \frac{1}{1 + \frac{\Delta T}{T_0}\left(\frac{HR_0}{A} - 1\right)}$$

$$= \frac{T_0}{T_0 + \Delta T\left(\frac{HR_0}{A} - 1\right)}$$

Substituting $T_0 - T$ for ΔT, cycle efficiency η for A/HR_0

$$F = \frac{\eta T_0}{T_0 - T(1 - \eta)} = \frac{538.72\eta}{538.72 - T(1 - \eta)}$$

Where T = new exhaust temperature in absolute degrees F, and η = 3412.75/(theoretical heat rate) from Table 1, or Fig. 5(*a*). The corrected heat rate, for exhaust pressure other than 1 in. Hg abs, is obtained from the equation

$$HR_T = F \times HR_0$$

Values of F for various exhaust pressures are given in Fig. 5(*b*). The alteration in heat rate, $D = HR_T - HR_0$, to be added to the heat rate found in Table 1, is obtained from the equation

$$D = \frac{HR_0(T - 538.72)(1 - \eta)}{538.72 - T(1 - \eta)}$$

When the value of D is negative, as is the case when the exhaust pressure is less than 1 in. Hg abs, D is to be subtracted from the heat rate given in Table 1.

Possible Application of Data

Two ways in which the heat-rate table might be used are (1) as a comparative standard of power-plant design and operation; and (2) as an aid in estimating plant performance with steam conditions somewhat above the current level.

In recent years, several technical papers have been written proposing this or that standard for use as a bogey for particular plant installations. The percentage ratio of actual to theoretical heat rate for any plant, however, combines inefficiencies both of design and operation. To gage plant operation alone, additional factors must be included in the standard.

These additional factors are required also where power-plant performance is to be estimated. For estimating purposes, modifying factors would be applied to the theoretical heat rate. For instance, turbine efficiency would change the heat rate so much. The use of four instead of an infinite number of heaters would result in an increase in heat rate. Similarly, modifying factors would be required for boiler efficiency, auxiliary power, exhaust pressure, etc. With these factors available, it would be possible to compute operating efficiency and estimate performance exactly.

These modifying factors, however, have not been computed yet, as prior demands on the authors' time by the war effort prevented any further extensive work being done on the study. In addition, it is felt that the factors will be extensive enough to warrant being the subject of a second or companion paper. The method of computing factors has not definitely crystallized but it is viewed as requiring about three steps, as follows:

1 Turbine engine efficiencies would be obtained from a paper by Warren and Knowlton.[10]

2 A number of heat-balance calculations would be made for typical feedwater-heater arrangements[11] using a range of terminal difference and final feedwater temperature, and taking into account condensate- and feedwater-pump efficiency.

3 Power-plant operating data for different steam conditions would be analyzed for as many stations as obtainable, to determine the effect on plant efficiency of boiler performance and auxiliary services.

Data would be tabulated ultimately as "efficiency" factors to show what increase in turbine and plant heat rate was due to each particular item of power-plant heat balance and operation. It is hoped that, with the publication of these theoretical cycle heat rates, others may be encouraged to take a hand in supplying needed performance data, in determining some of the factors, and in making them available to the engineering profession.

In the meantime, the data here presented can be helpful if a certain amount of ingenuity is used. In making estimates, existing plant and turbine-performance data usually will provide sufficient information for calculating approximate modifying factors which in turn may be applied directly or after adjustment to the theoretical heat rate at the steam conditions under investigation. Plant performance can be reported as a percentage of the theoretical figure. By so doing, the advantage one plant has over another through using higher steam pressure and temperature is eliminated from the performance comparison. Undoubtedly, other uses in addition to these two will come to mind.

An example of the relationship between theoretical-cycle and actual plant heat rates is shown in Table 2. Although in the cases cited, plants B and C differ in economy by 700 Btu per kwhr, they are operating almost equally effectively, as indicated by the similarity in plant-design and performance factors. To demonstrate roughly how modifying factors might be applied to theoretical heat rates, Table 2 shows a typical but incomplete breakdown of the over-all factors into a list of items and attendant factors which affect plant performance.

These factors were obtained from turbogenerator heat-balance data and actual plant performance of three different stations. In the normal course of events, the companion paper will contain tables of factors covering a complete pressure-temperature range of steam and feedwater conditions for typical power-plant design based upon available up-to-date information on turbine and station performance. Subsequently, if a marked change in power-plant design occurs, supplementary factors would be published to bring the series again up to date.

Lacking time now to make heat-balance computations and analyze plant records, the authors provide Table 2 solely to illustrate how factors might apply in the cases cited for three different power stations. A discussion of methods used to obtain factors will be incorporated in the companion paper. Items, it is hoped, will be subdivided eventually to indicate the effect on economy of the more detailed phases of plant design and operation. In the meantime, correlations between theoretical cycle and plant heat

[10] "Relative 'Engine Efficiencies' Realizable From Large Modern Steam-Turbine-Generator Units," by G. B. Warren and P. H. Knowlton, Trans. A.S.M.E., vol. 63, 1941, pp. 125–135.

[11] As, for instance, any "standardized" cycles which may be the outcome of the "Preferred Standards for Steam Turbine Generators," proposed Nov. 3, 1938, by the Subcommittee on Standardization of the National Defense Power Committee, and subsequent work thereon instigated by the A.S.M.E.

rates of existing power stations will help engineers in predicting heat rates for somewhat extended throttle conditions.

Other Regenerative Cycles

At least two modifications of the regenerative cycle proposed in recent years merit comment. One suggestion is for moisture extraction in the wet stages, and the other, for isothermal compression of superheated bled steam.

Moisture Extraction. Within the wet-steam region, the theoretical regenerative cycle has Carnot efficiency without requiring 100 per cent moisture withdrawal, as proposed by Dr. W. M. Meijer,[12] since expansion is assumed to take place isentropically. This thoughtful proposal of an integral moisture-withdrawal provision, however, merits serious attention on two practical scores, i.e., (1) moisture causes a definite loss of efficiency in an actual turbine attributable to bucket-impingement shock and supersaturation effect, and (2) the possibility of bucket erosion by moisture in the lower stages appreciably restricts turbine designers in the selection of turbine-throttle conditions.

Moisture withdrawal of 100 per cent might be difficult to attain without occasioning other serious thermodynamic losses. Fortunately, such effectiveness is not essential. For instance, if 12 per cent moisture were deemed the maximum permissible practical limit, and if in any stage not more than 2 to 3 per cent moisture were produced, then it would be necessary to withdraw only 2 to 3 per cent moisture to keep within the 12 per cent limit. Efficiency of withdrawal then need be only 15 to 25 per cent in each stage to permit steam expansion to continue to low exhaust conditions which otherwise might be equivalent to 20 per cent moisture or better. Hence provision for moisture withdrawal at even this low efficiency would be sufficient to overcome present objections on the score of bucket erosion to using high throttle steam pressures without attendant high superheated-steam temperatures. There remains, however, sufficient reason for the present high superheat temperatures, on the grounds of improved cycle efficiency as well as improved turbine efficiency, as compared to what is possible with the saturated-steam cycle.

With the application of superheat, the efficiency of the theoretical regenerative steam cycle does increase although not to the extent of the Carnot cycle. The use of superheat, however, is advantageous, and particularly so where superheated steam is bled for regeneration. Even where an arbitrary restriction is placed on the regenerative cycle that only dry-saturated and wet steam may be bled, the provision of superheated steam to the cycle is still advantageous. Fig. 6 shows the relation of theoretical heat rates versus throttle steam temperature for the two foregoing cycles, namely, regeneration only by saturated and wet steam,[13] and regeneration up to throttle saturation temperature using bled superheated steam. In either case, a reduction in heat rate with increased superheat is noticeable.

Isothermal Compression. While, with the use of superheated steam, the theoretical regenerative-cycle efficiency does increase, it falls below the corresponding Carnot efficiency because it is not a wholly reversible cycle. James F. Field[14] points out that "the ideal engine would therefore require additional elements to compress this superheated bled steam isothermally to the corresponding saturation temperature and pressure before actually mixing with feedwater." This cycle can be studied by means of the tabular-integration method. A typical example, taking into account all auxiliary power usage, showed the cycle to have Carnot efficiency.

The choice of a system for gaging power-plant performance cannot be made arbitrarily. At present, the economy of American power plants is reported in terms of British thermal units per kilowatthour. British plants use a scale of percentage efficiency. In both cases, a plant of high steam conditions would be recorded as having a better economy than one with lower steam conditions. The latter, however, may have been operated more creditably and deserves proper recognition. By relating plant performance to some standard, whether it be the theoretical regenerative-cycle heat rate, the Carnot cycle, or even the basic efficiency,[15] a somewhat better gage can be obtained of what advantage operating personnel is taking of the facilities at their command. Before a universal gage is established, however, engineers will have to agree among themselves as to what standard they will adopt, how closely such a standard needs to match the steam cycles for which it is established, and what allowance will be made for uncontrollable losses attendant on the number of feedwater heaters installed, load distribution, etc.

Conclusion

This paper has described a tabular-integration method for computing theoretical heat rates, and demonstrated its use by presenting a table of heat rates for a theoretical regenerative steam cycle. It is hoped that this table will prove helpful in at least three ways:

[12] "The Extraction of Condensate From Expanding Steam," by Dr. W. M. Meijer, Trans. Institution of Naval Architects, vol. 81, 1939, pp. 36–58; see also *Engineering*, vol. 147, April 7, 1939, p. 147.

[13] A Typical formula for computing the efficiency of wet-steam-regeneration cycle may be found in "High Pressure, Reheating, and Regenerating for Steam Power Plants," by C. F. Hirshfeld and F. O. Ellenwood, Trans. A.S.M.E., vol. 45, 1923, p. 663.

[14] "A Suggested Basis of Comparison for the Efficiency of Steam Turbo-Generators and Steam-Electric Generating Stations," *Journal* of The Institution of Civil Engineers, vol. 10, 1938, pp. 241–253.

[15] The use of basic efficiency was described by Sir Leonard Pearce, C.B.E., D.Sc., in his Thomas Hawksley Lecture on "A Review of Forty Years' Development in Mechanical Engineering Plant for Power Stations," *Journal* of The Institution of Mechanical Engineers, vol. 142, 1940, pp. 305–363. In an appendix under "Operating Efficiency," Sir Leonard discusses the use of some major factors modifying theoretical or basic plant efficiency.

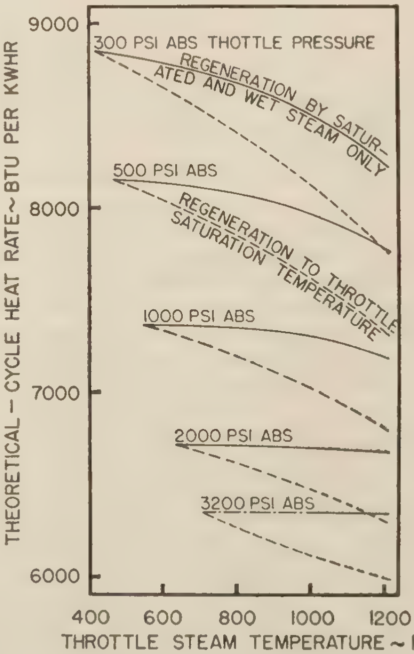

Fig. 6 Reduction in Theoretical Regenerative-Cycle Heat Rates With Increased Superheat in Throttle Steam

1 In comparing design and operation of different power plants operating at different steam conditions.

2 In estimating the approximate performance of power plants.

3 In stimulating others to undertake preparation of tables of modifying factors for use in making exact performance estimates.

Since the tabular-integration method of calculating heat rates can be adapted readily, it is thought that it will prove useful to others in studying new cycles.

Acknowledgment

The authors wish to express their appreciation for the assistance of everyone concerned with the preparation of this paper. We are indebted to Mr. Sabin Crocker whose stimulating interest and hearty encouragement helped on the work; to Mr. Max W. Benjamin and Mr. E. L. Robinson for timely and helpful suggestions; to the General Electric Company for providing the means for computing "key-point" heat rates; and to The Detroit Edison Company for making the study possible.

Appendix

Nomenclature

The following nomenclature is used in the first part of the paper and in the Appendix:

H_1 = enthalpy of throttle steam, Btu per lb

H = enthalpy of steam at any stage in the prime mover, Btu per lb

h = enthalpy of saturated feedwater, corresponding to the pressure in the selected stage of the prime mover, Btu per lb

$F = Nv/5.4$ = energy added by the perfect boiler feed pump, Btu per lb

N = increase in feedwater pressure in the pump, psi

v = specific volume of feedwater at pump suction, cu ft per lb

$h' = h + F$ = enthalpy of feedwater leaving the boiler feed pump after isentropic compression, and entering the next higher heater, Btu per lb

$dh = h - h'$ = heat added in heater by bled steam, Btu per lb of feedwater

$1 + w$ = weight of steam at any turbine stage, lb
= weight of feedwater returning at any stage, lb

dw = weight of steam bled at any turbine stage, lb

$_e$ = Naperian logarithmic base

Short Form for Computing Theoretical Regenerative-Cycle Heat Rates

In Table 3, an example is carried through according to the short form for computing theoretical regenerative-cycle heat rates. The conditions stated are throttle-steam pressure = 3200 psia; throttle-steam temperature = 1200 F; throttle-steam entropy = 1.5745.

To complete the computations of Table 3, by short form, the following steps must be taken:

(a)

$$1 + w = e^{\sum_{\text{avg}} \frac{dh}{(H-h)}}$$

Add up column (9) = 0.9773690
Convert to logarithmic base 10
0.9773690 × 0.43428 = 0.4244518
Find antilog = 2.6574

(b) Heat input to cycle = heat supplied at throttle less heat returned to steam generator

$= (1 + w)(H_1 - h_1)$
$= 2.6574 \times 697.5$ from column (7)
$= 1853.54$ Btu per lb of steam at condenser

(c) Heat rejected to condenser

$= 1(H_e - h_c)$
$= 1 \times 799.14$ Btu from column (7)

(d) Heat converted to energy and delivered at electric-generator terminals = item (b) — item (c) = 1853.54 — 799.14 = 1054.40 Btu

(e) Cycle efficiency = output/input = 1054.40/1853.54 = 0.56886

(f) Theoretical heat rate = 3412.75/cycle efficiency = 5999 Btu per kwhr.

Computations by Long Form

For the same example, the long form for computing regenerative-cycle heat rates and boiler-feed-pump work is demonstrated in Table 4, which is a continuation of the short form.

To complete the computations by long form, the following steps must be taken:

(a) Rankine cycle output = heat at throttle minus heat at exhaust
= first minus last reading, column (2)
= 1569.9 — 846.19
= 723.71 Btu

(b) Extra output by regenerative cycle = E = 359.64 Btu

(c) Total prime-mover output (a) + (b) = 1083.35 Btu

(d) Less boiler-feed-pump work = 28.99 Btu from column (14)

(e) Net regenerative-cycle output = 1054.36 Btu

(f) Heat rejected = 799.14 Btu from column (7)

(g) Cycle input = 1853.50 Btu

(h) Cycle efficiency = 1054.36/1853.50 = 0.56885

(i) Theoretical cycle heat rate = 3412.75/0.56885 = 5999 Btu per kwhr

Note: Investigation of the isothermal-compression regenerative cycle, for example, would require the use of the long form of computation. Calculations of the work of compressing bled steam would be handled in a manner similar to that used for the boiler feed pump. Additional columns of calculations, however, would be required to suit the case being studied.

TABLE 1 THEORETICAL REGENERATIVE-STEAM-CYCLE HEAT RATES, BTU PER KWHR (1 IN. HG ABS EXHAUST PRESSURE)

Throttle pressure, psia	*Saturation temperature, deg F*	Throttle temperature: Saturation	420F	440F	460F	480F	500F	520F	540F	560F	580F	600F	620F	640F	660F	680F	700F	720F	740F	760F	780F	800F
300	*417.3*	8850	8850	8846	8838	8828	8813	8796	8777	8756	8732	8708	8683	8657	8630	8602	8573	8543	8512	8481	8450	8418
310	*420.4*	8802		8799	8792	8783	8768	8751	8732	8711	8688	8664	8639	8614	8588	8561	8532	8502	8472	8442	8411	8379
320	*423.3*	8756		8753	8747	8739	8725	8709	8690	8669	8646	8622	8598	8573	8548	8521	8493	8464	8435	8405	8374	8342
330	*426.2*	8712		8710	8704	8696	8683	8667	8649	8628	8605	8582	8558	8534	8509	8483	8455	8427	8398	8368	8337	8306
340	*429.0*	8670		8668	8662	8654	8642	8626	8609	8589	8567	8544	8520	8496	8472	8446	8418	8390	8362	8333	8308	8272
350	*431.7*	8629		8628	8622	8615	8603	8588	8571	8551	8530	8507	8484	8460	8436	8410	8382	8355	8327	8299	8270	8239
360	*434.4*	8589		8588	8583	8576	8565	8550	8533	8514	8494	8472	8449	8425	8401	8375	8348	8321	8294	8266	8237	8207
370	*437.0*	8551		8550	8546	8539	8528	8514	8498	8479	8459	8438	8415	8391	8367	8341	8315	8288	8261	8234	8205	8176
380	*439.6*	8515		8515	8511	8504	8493	8479	8463	8445	8426	8405	8383	8359	8335	8309	8283	8257	8230	8203	8174	8145
390	*442.1*	8479			8476	8470	8460	8447	8431	8413	8394	8373	8351	8327	8303	8278	8252	8226	8199	8172	8144	8116
400	*444.6*	8445			8443	8438	8428	8416	8400	8382	8363	8342	8320	8297	8273	8248	8222	8196	8169	8142	8115	8088
410	*447.0*	8411			8409	8406	8397	8385	8370	8352	8333	8312	8290	8267	8244	8219	8193	8167	8141	8115	8088	8061
420	*449.4*	8379			8378	8376	8367	8355	8340	8322	8303	8282	8261	8238	8215	8190	8165	8140	8114	8088	8062	8035
430	*451.7*	8348			8347	8346	8338	8326	8311	8293	8274	8253	8232	8210	8187	8163	8138	8113	8088	8062	8036	8010
440	*454.0*	8318			8317	8316	8309	8297	8282	8265	8246	8225	8204	8182	8160	8136	8112	8087	8062	8037	8012	7986
450	*456.3*	8289			8289	8287	8281	8269	8255	8238	8219	8198	8177	8156	8134	8111	8087	8063	8038	8013	7988	7962
460	*458.5*	8260			8260	8259	8253	8242	8228	8212	8193	8172	8152	8131	8109	8086	8062	8038	8014	7990	7965	7939
470	*460.7*	8232				8231	8226	8215	8201	8186	8167	8147	8127	8106	8084	8061	8038	8015	7991	7967	7942	7916
480	*462.8*	8205				8205	8200	8189	8175	8160	8142	8123	8103	8082	8060	8038	8016	7992	7969	7945	7920	7894
490	*464.9*	8179				8179	8175	8164	8151	8136	8119	8100	8080	8059	8038	8016	7994	7970	7947	7923	7898	7873
500	*467.0*	8154				8154	8150	8141	8128	8114	8097	8078	8058	8037	8016	7994	7972	7949	7926	7902	7877	7852
520	*471.1*	8106				8106	8102	8094	8082	8069	8053	8035	8016	7995	7974	7952	7930	7907	7884	7861	7838	7812
540	*475.0*	8059				8059	8056	8048	8037	8024	8009	7992	7973	7953	7933	7912	7890	7868	7845	7822	7799	7775
560	*478.8*	8015				8015	8012	8005	7994	7981	7967	7951	7933	7914	7895	7874	7852	7830	7808	7785	7762	7738
580	*482.6*	7972					7971	7964	7954	7942	7928	7912	7894	7875	7856	7836	7815	7793	7771	7749	7726	7702
600	*486.2*	7931					7931	7925	7916	7904	7890	7875	7858	7839	7820	7800	7780	7759	7737	7715	7692	7669
620	*489.8*	7892					7892	7888	7880	7869	7855	7840	7823	7804	7785	7765	7745	7724	7703	7681	7659	7637
640	*493.2*	7855					7855	7851	7844	7833	7820	7806	7789	7771	7752	7732	7712	7691	7670	7649	7628	7606
660	*496.6*	7819					7819	7816	7809	7798	7786	7773	7757	7739	7720	7700	7680	7660	7640	7619	7598	7576
680	*499.9*	7784					7784	7782	7776	7766	7754	7740	7725	7708	7689	7670	7650	7630	7610	7590	7569	7546
700	*503.1*	7751						7749	7744	7735	7724	7711	7695	7678	7660	7641	7621	7601	7582	7562	7542	7521
720	*506.2*	7720						7718	7713	7705	7694	7681	7665	7649	7631	7612	7593	7574	7555	7535	7515	7494
740	*509.3*	7689						7688	7684	7676	7665	7652	7637	7621	7603	7585	7566	7547	7528	7509	7489	7468
760	*512.4*	7659						7659	7656	7648	7637	7624	7609	7593	7576	7558	7539	7520	7501	7482	7463	7443
780	*515.3*	7630						7630	7628	7621	7610	7597	7582	7566	7549	7531	7513	7495	7477	7458	7439	7419
800	*518.2*	7601						7601	7600	7594	7584	7571	7556	7540	7523	7506	7488	7470	7452	7434	7415	7395
820	*521.1*	7575							7574	7568	7558	7546	7532	7516	7499	7482	7464	7446	7428	7410	7391	7372
840	*523.9*	7549							7548	7543	7534	7522	7508	7492	7476	7459	7441	7423	7405	7387	7369	7350
860	*526.6*	7523							7522	7518	7509	7498	7484	7469	7453	7436	7418	7401	7383	7365	7347	7329
880	*529.3*	7498							7497	7494	7486	7475	7461	7446	7430	7413	7396	7379	7362	7344	7326	7308
900	*532.0*	7474							7474	7471	7463	7453	7439	7424	7408	7392	7375	7358	7341	7323	7305	7287
920	*534.6*	7450							7450	7448	7441	7431	7418	7403	7387	7371	7354	7337	7320	7303	7285	7267
940	*537.2*	7427							7427	7426	7419	7410	7397	7383	7367	7351	7334	7317	7300	7283	7266	7248
960	*539.7*	7405							7405	7404	7398	7389	7376	7362	7347	7331	7314	7298	7281	7264	7247	7229
980	*542.2*	7384								7383	7377	7368	7355	7341	7327	7311	7295	7279	7263	7246	7229	7211
1000	*544.6*	7364								7361	7355	7347	7334	7321	7307	7292	7276	7260	7244	7228	7211	7193
1025	*547.6*	7338								7336	7332	7324	7311	7298	7284	7269	7254	7239	7223	7207	7190	7172
1050	*550.6*	7314								7312	7309	7301	7288	7275	7261	7247	7232	7217	7201	7185	7168	7151
1075	*553.3*	7290								7288	7286	7279	7267	7254	7240	7226	7211	7196	7180	7164	7147	7130
1100	*556.3*	7267								7266	7264	7257	7245	7232	7219	7205	7190	7175	7159	7143	7126	7109
1125	*559.1*	7245								7245	7243	7236	7224	7212	7199	7185	7170	7155	7139	7123	7106	7089
1150	*561.9*	7223									7221	7215	7204	7192	7179	7165	7150	7135	7120	7104	7087	7070
1175	*564.5*	7202									7200	7194	7183	7171	7158	7145	7131	7117	7102	7086	7069	7052
1200	*567.2*	7181									7179	7174	7164	7152	7139	7126	7112	7098	7083	7067	7051	7034
1225	*569.8*	7161									7159	7154	7144	7133	7120	7107	7093	7079	7064	7048	7032	7016
1250	*572.4*	7141									7139	7134	7124	7113	7101	7088	7075	7061	7046	7031	7015	6999
1275	*574.9*	7122									7119	7115	7106	7096	7084	7071	7058	7044	7029	7014	6998	6982

TABLE 1 (*Cont.*) THEORETICAL REGENERATIVE-STEAM-CYCLE HEAT RATES, BTU PER KWHR (1 IN. HG ABS EXHAUST PRESSURE)

Throttle pressure, psia	Throttle temperature																				
	800F	820F	840F	860F	880F	900F	920F	940F	960F	980F	1000F	1020F	1040F	1060F	1080F	1100F	1120F	1140F	1160F	1180F	1200F
300	8418	8385	8352	8320	8288	8256	8224	8192	8160	8127	8094	8062	8029	7996	7963	7931	7898	7866	7834	7802	7771
310	8379	8347	8315	8283	8251	8219	8187	8156	8124	8092	8060	8028	7996	7963	7930	7898	7866	7834	7803	7772	7741
320	8342	8310	8278	8246	8215	8184	8153	8121	8089	8058	8027	7996	7964	7931	7899	7867	7836	7804	7773	7742	7712
330	8306	8275	8244	8213	8182	8151	8120	8088	8057	8026	7995	7964	7932	7900	7868	7837	7806	7775	7744	7714	7684
340	8272	8241	8210	8180	8149	8119	8088	8057	8026	7995	7964	7933	7901	7870	7839	7808	7777	7747	7717	7687	7657
350	8239	8208	8178	8148	8118	8088	8058	8027	7996	7965	7934	7903	7872	7841	7810	7780	7750	7720	7690	7661	7631
360	8207	8177	8147	8117	8087	8057	8027	7997	7966	7936	7905	7875	7844	7814	7784	7754	7724	7694	7664	7635	7606
370	8176	8147	8118	8088	8058	8028	7998	7968	7938	7908	7878	7848	7818	7788	7758	7729	7699	7669	7640	7611	7582
380	8145	8116	8087	8058	8029	8000	7971	7941	7911	7881	7852	7823	7793	7763	7734	7705	7676	7646	7617	7588	7559
390	8116	8088	8060	8031	8002	7973	7944	7914	7885	7856	7827	7798	7769	7739	7710	7681	7652	7623	7594	7565	7537
400	8088	8060	8032	8004	7976	7948	7919	7890	7861	7832	7803	7774	7745	7716	7687	7658	7629	7600	7572	7544	7516
410	8061	8033	8006	7979	7952	7924	7895	7866	7837	7808	7779	7750	7721	7692	7664	7636	7607	7579	7551	7523	7495
420	8035	8008	7982	7955	7928	7900	7872	7843	7814	7785	7756	7727	7698	7670	7642	7614	7586	7558	7530	7502	7475
430	8010	7983	7957	7931	7904	7877	7849	7820	7791	7763	7734	7706	7677	7649	7621	7593	7565	7537	7509	7482	7455
440	7986	7959	7933	7907	7881	7854	7826	7798	7769	7741	7713	7685	7657	7629	7601	7573	7545	7517	7490	7463	7436
450	7692	7936	7910	7884	7858	7831	7803	7775	7747	7719	7692	7664	7636	7608	7580	7553	7525	7498	7471	7444	7417
460	7939	7913	7887	7861	7835	7809	7782	7754	7726	7699	7672	7644	7617	7589	7561	7534	7507	7480	7453	7426	7399
470	7916	7891	7865	7839	7813	7787	7760	7733	7706	7679	7652	7624	7597	7569	7542	7515	7489	7462	7435	7408	7381
480	7894	7869	7844	7818	7792	7766	7740	7713	7686	7660	7633	7606	7579	7551	7524	7497	7471	7444	7417	7391	7364
490	7873	7848	7823	7798	7772	7746	7720	7693	7667	7641	7614	7587	7560	7533	7506	7479	7453	7426	7400	7374	7348
500	7852	7827	7802	7777	7752	7726	7700	7674	7648	7622	7595	7568	7542	7515	7489	7462	7436	7409	7383	7357	7332
520	7812	7788	7763	7738	7713	7688	7663	7637	7611	7585	7559	7532	7506	7480	7454	7428	7402	7376	7350	7324	7299
540	7775	7751	7727	7702	7677	7652	7627	7602	7576	7550	7524	7498	7472	7446	7421	7396	7370	7344	7319	7293	7268
560	7738	7714	7690	7666	7642	7617	7592	7567	7542	7517	7491	7466	7441	7415	7390	7365	7340	7315	7290	7265	7240
580	7702	7679	7655	7631	7607	7583	7559	7534	7509	7484	7459	7434	7409	7384	7359	7335	7311	7287	7262	7238	7213
600	7669	7646	7622	7598	7574	7550	7526	7502	7478	7453	7428	7403	7378	7354	7330	7306	7282	7258	7234	7210	7186
620	7637	7614	7591	7567	7543	7519	7495	7471	7447	7423	7398	7374	7350	7326	7302	7278	7254	7230	7207	7183	7160
640	7606	7583	7560	7537	7513	7489	7465	7441	7417	7393	7370	7346	7322	7298	7274	7251	7228	7205	7182	7159	7136
660	7576	7554	7531	7508	7484	7460	7436	7412	7389	7366	7343	7320	7296	7273	7249	7226	7203	7180	7157	7135	7112
680	7546	7526	7504	7480	7456	7433	7409	7386	7363	7340	7317	7294	7271	7248	7225	7202	7179	7156	7134	7111	7089
700	7521	7499	7476	7453	7429	7406	7383	7360	7337	7314	7291	7269	7246	7224	7202	7179	7157	7134	7112	7089	7067
720	7494	7472	7449	7426	7402	7379	7356	7333	7310	7288	7266	7244	7222	7200	7178	7156	7134	7112	7090	7068	7046
740	7468	7446	7423	7400	7376	7353	7330	7308	7286	7264	7242	7220	7198	7176	7154	7133	7112	7090	7068	7046	7025
760	7443	7422	7399	7376	7352	7329	7307	7285	7263	7241	7219	7197	7175	7153	7132	7111	7090	7069	7048	7026	7005
780	7419	7398	7376	7353	7329	7306	7284	7262	7240	7218	7197	7175	7154	7132	7111	7090	7069	7048	7027	7006	6986
800	7395	7374	7353	7330	7307	7284	7262	7240	7218	7197	7176	7154	7133	7112	7091	7070	7049	7028	7007	6987	6967
820	7372	7352	7331	7308	7286	7263	7241	7219	7197	7176	7155	7134	7113	7092	7071	7051	7030	7009	6988	6968	6948
840	7350	7330	7309	7287	7265	7242	7220	7198	7176	7155	7134	7113	7092	7072	7052	7032	7011	6990	6970	6950	6930
860	7329	7310	7289	7267	7245	7222	7200	7178	7157	7136	7115	7094	7074	7054	7034	7014	6993	6972	6952	6932	6912
880	7308	7289	7269	7247	7225	7202	7180	7159	7138	7117	7096	7076	7056	7036	7016	6996	6976	6955	6935	6915	6895
900	7287	7268	7248	7227	7205	7183	7161	7140	7119	7098	7077	7057	7037	7017	6997	6978	6958	6938	6918	6898	6878
920	7267	7248	7228	7207	7186	7164	7143	7122	7101	7080	7059	7039	7019	6999	6980	6961	6941	6921	6901	6881	6862
940	7248	7229	7209	7188	7167	7146	7125	7104	7083	7062	7041	7021	7001	6982	6963	6944	6924	6904	6884	6865	6846
960	7229	7210	7190	7170	7149	7129	7107	7086	7065	7045	7024	7004	6985	6966	6947	6928	6908	6888	6869	6850	6831
980	7211	7192	7173	7153	7132	7111	7090	7070	7049	7029	7008	6988	6969	6950	6931	6912	6892	6873	6854	6835	6816
1000	7193	7174	7155	7135	7115	7095	7075	7055	7034	7014	6993	6973	6953	6934	6915	6896	6877	6858	6840	6821	6802
1025	7172	7153	7134	7114	7094	7074	7054	7034	7014	6994	6973	6953	6934	6915	6896	6877	6858	6839	6820	6801	6782
1050	7151	7133	7114	7094	7074	7054	7034	7014	6994	6974	6954	6935	6916	6897	6878	6859	6839	6820	6801	6782	6763
1075	7130	7112	7093	7074	7054	7034	7015	6995	6976	6956	6936	6917	6898	6879	6860	6841	6821	6802	6783	6764	6745
1100	7109	7091	7073	7054	7035	7015	6996	6976	6957	6937	6918	6899	6880	6862	6843	6824	6804	6785	6766	6747	6728
1125	7089	7071	7053	7035	7016	6997	6978	6959	6940	6920	6901	6882	6863	6845	6826	6807	6788	6769	6750	6731	6712
1150	7070	7052	7034	7016	6998	6979	6960	6941	6922	6903	6884	6866	6847	6829	6810	6791	6772	6754	6735	6716	6697
1175	7052	7035	7017	6999	6981	6962	6944	6925	6906	6887	6868	6850	6831	6813	6794	6775	6756	6738	6719	6701	6682
1200	7034	7017	6999	6981	6963	6945	6927	6909	6890	6871	6852	6834	6815	6797	6778	6759	6741	6723	6704	6686	6667
1225	7016	6999	6982	6964	6946	6928	6910	6892	6874	6855	6836	6818	6800	6782	6763	6744	6726	6708	6689	6671	6652
1250	6999	6983	6966	6948	6930	6912	6894	6876	6858	6840	6821	6803	6785	6767	6748	6729	6711	6693	6674	6656	6637
1275	6982	6966	6949	6932	6914	6896	6878	6860	6842	6824	6806	6788	6770	6752	6734	6715	6697	6679	6660	6642	6623

TABLE 1 (*Cont.*) THEORETICAL REGENERATIVE-STEAM-CYCLE HEAT RATES, BTU PER KWHR (1 IN. HG ABS EXHAUST PRESSURE)

Throttle pressure, psia	*Saturation temperature, deg F*	Throttle temperature																
		Saturation	580F	600F	620F	640F	660F	680F	700F	720F	740F	760F	780F	800F	820F	840F	860F	880F	900F
1300	*577.5*	7104	7100	7096	7087	7077	7066	7054	7041	7027	7012	6997	6982	6966	6950	6933	6916	6898	6880
1325	*579.9*	7085	7082	7078	7069	7059	7048	7036	7024	7010	6996	6981	6966	6950	6934	6917	6900	6882	6864
1350	*582.4*	7067		7060	7051	7041	7030	7019	7007	6994	6980	6965	6950	6934	6918	6901	6884	6867	6849
1375	*584.8*	7049		7043	7035	7025	7014	7003	6991	6978	6964	6949	6934	6919	6903	6886	6869	6852	6834
1400	*587.1*	7033		7027	7019	7009	6999	6988	6976	6963	6949	6934	6919	6904	6888	6871	6854	6837	6820
1425	*589.4*	7016		7012	7004	6994	6984	6973	6961	6948	6934	6919	6904	6889	6873	6857	6840	6823	6806
1450	*591.7*	7000		6997	6989	6980	6970	6959	6947	6934	6920	6905	6890	6875	6859	6843	6826	6809	6792
1475	*594.0*	6984		6982	6975	6966	6956	6945	6932	6920	6906	6891	6876	6861	6845	6829	6812	6796	6779
1500	*596.2*	6969		6968	6961	6952	6942	6931	6919	6906	6892	6877	6862	6847	6831	6815	6798	6782	6766
1525	*598.4*	6954		6954	6947	6938	6928	6917	6905	6892	6878	6864	6849	6833	6817	6801	6785	6769	6753
1550	*600.6*	6940			6933	6924	6914	6903	6892	6878	6864	6850	6835	6820	6804	6788	6772	6756	6740
1575	*602.8*	6925			6920	6911	6901	6890	6879	6865	6851	6837	6822	6807	6791	6775	6759	6743	6727
1600	*604.9*	6911			6907	6898	6888	6877	6866	6852	6838	6824	6809	6794	6778	6762	6746	6730	6714
1625	*607.0*	6896			6894	6886	6876	6865	6854	6840	6826	6811	6796	6781	6766	6750	6734	6718	6702
1650	*609.0*	6882			6881	6873	6864	6853	6842	6828	6814	6799	6784	6769	6754	6738	6722	6706	6690
1675	*611.1*	6869			6868	6861	6852	6841	6829	6816	6802	6787	6772	6757	6742	6726	6710	6694	6678
1700	*613.1*	6857			6856	6850	6841	6830	6818	6805	6791	6776	6761	6745	6730	6714	6698	6682	6666
1750	*617.1*	6832			6831	6825	6817	6806	6794	6782	6769	6754	6739	6723	6708	6692	6676	6660	6644
1800	*621.0*	6807				6800	6792	6782	6771	6759	6746	6732	6717	6701	6686	6670	6654	6638	6622
1850	*624.8*	6783				6778	6770	6760	6749	6737	6724	6710	6695	6679	6664	6649	6633	6617	6601
1900	*628.6*	6760				6757	6749	6739	6728	6716	6703	6689	6674	6658	6643	6628	6613	6597	6581
1950	*632.3*	6738				6736	6728	6718	6707	6695	6682	6668	6652	6637	6622	6607	6592	6577	6561
2000	*635.8*	6717				6716	6708	6698	6687	6675	6662	6648	6633	6617	6602	6587	6572	6557	6542
2050	*639.2*	6696				6696	6689	6679	6668	6656	6643	6629	6614	6598	6584	6569	6554	6539	6524
2100	*642.8*	6676					6671	6661	6650	6638	6625	6611	6596	6580	6566	6551	6536	6511	6506
2150	*646.0*	6657					6653	6643	6632	6620	6607	6593	6578	6563	6549	6534	6519	6494	6489
2200	*649.5*	6638					6636	6626	6614	6602	6589	6575	6561	6546	6532	6517	6502	6487	6472
2250	*652.6*	6620					6619	6609	6597	6585	6572	6558	6544	6529	6515	6500	6486	6471	6456
2300	*655.9*	6602					6602	6592	6580	6568	6555	6541	6527	6512	6498	6483	6469	6455	6440
2350	*659.1*	6585					6585	6575	6563	6551	6538	6524	6510	6495	6482	6467	6453	6439	6424
2400	*662.1*	6568						6559	6547	6535	6522	6508	6494	6480	6466	6451	6437	6423	6408
2450	*665.1*	6551						6544	6532	6520	6507	6493	6479	6465	6451	6436	6422	6408	6393
2500	*668.1*	6535						6530	6518	6506	6493	6479	6465	6450	6436	6421	6407	6393	6378
2550	*671.0*	6520						6516	6504	6492	6479	6465	6451	6436	6422	6407	6393	6379	6364
2600	*673.9*	6505						6502	6491	6478	6465	6451	6437	6422	6408	6393	6379	6365	6350
2650	*676.7*	6490						6488	6478	6464	6451	6437	6423	6408	6394	6379	6365	6351	6336
2700	*679.5*	6476						6475	6465	6451	6437	6423	6409	6394	6380	6365	6351	6337	6322
2750	*682.2*	6462							6452	6437	6423	6409	6395	6380	6366	6351	6337	6323	6308
2800	*685.0*	6449							6439	6424	6410	6395	6381	6367	6353	6338	6324	6310	6295
2850	*687.7*	6435							6427	6411	6397	6382	6368	6354	6340	6325	6311	6297	6282
2900	*690.3*	6422							6416	6400	6385	6370	6355	6341	6327	6313	6299	6285	6270
2950	*692.8*	6410							6405	6388	6373	6358	6343	6329	6315	6301	6287	6273	6258
3000	*695.4*	6398							6394	6377	6361	6346	6331	6317	6303	6289	6275	6261	6246
3050	*697.9*	6386							6384	6367	6350	6334	6319	6305	6291	6277	6263	6249	6234
3100	*700.3*	6374								6357	6340	6324	6308	6293	6279	6265	6251	6237	6223
3150	*702.8*	6362								6346	6329	6312	6296	6281	6267	6253	6239	6226	6212
3200	*705.1*	6351								6336	6318	6301	6285	6270	6256	6242	6228	6215	6201

TABLE 1 (*Cont.*) THEORETICAL REGENERATIVE-STEAM-CYCLE HEAT RATES, BTU PER KWHR (1 IN. HG ABS EXHAUST PRESSURE)

Throttle pressure, psia	Throttle temperature 900F	920F	940F	960F	980F	1000F	1020F	1040F	1060F	1080F	1100F	1120F	1140F	1160F	1180F	1200F
1300	6880	6863	6845	6827	6809	6791	6773	6755	6737	6719	6701	6683	6664	6645	6627	6608
1325	6864	6847	6829	6812	6794	6776	6759	6741	6723	6705	6687	6669	6650	6631	6613	6594
1350	6849	6832	6815	6798	6780	6762	6745	6727	6709	6691	6673	6655	6636	6617	6599	6580
1375	6834	6817	6800	6783	6766	6748	6731	6713	6695	6677	6659	6641	6623	6604	6586	6567
1400	6820	6803	6786	6769	6752	6734	6717	6699	6682	6664	6646	6628	6610	6591	6573	6554
1425	6806	6789	6772	6755	6738	6720	6703	6686	6669	6651	6633	6615	6597	6578	6560	6541
1450	6792	6776	6759	6742	6725	6707	6690	6673	6656	6638	6620	6602	6584	6565	6547	6529
1475	6779	6763	6746	6729	6712	6694	6677	6660	6643	6625	6607	6589	6571	6553	6535	6517
1500	6766	6750	6733	6716	6699	6681	6664	6647	6630	6612	6595	6577	6559	6541	6523	6505
1525	6753	6737	6720	6703	6686	6668	6651	6634	6617	6600	6583	6565	6547	6529	6511	6493
1550	6740	6724	6707	6690	6673	6656	6639	6622	6605	6588	6571	6553	6535	6517	6499	6481
1575	6727	6711	6695	6678	6661	6644	6627	6610	6593	6576	6559	6541	6523	6505	6487	6470
1600	6714	6698	6682	6666	6649	6632	6615	6598	6581	6564	6547	6529	6511	6493	6476	6459
1625	6702	6686	6670	6654	6637	6620	6603	6586	6569	6552	6535	6517	6499	6482	6465	6448
1650	6690	6674	6658	6642	6625	6608	6591	6574	6557	6540	6523	6505	6488	6471	6454	6437
1675	6678	6662	6646	6630	6614	6597	6580	6563	6546	6529	6511	6494	6477	6460	6443	6426
1700	6666	6650	6634	6618	6602	6586	6569	6552	6535	6518	6501	6484	6467	6450	6433	6416
1750	6644	6628	6612	6597	6581	6565	6548	6531	6514	6497	6480	6463	6446	6429	6413	6397
1800	6622	6606	6590	6575	6559	6544	6528	6511	6494	6477	6460	6443	6426	6410	6394	6378
1850	6601	6585	6569	6554	6538	6523	6507	6491	6474	6457	6440	6423	6407	6391	6375	6359
1900	6581	6565	6549	6534	6518	6503	6487	6471	6455	6438	6421	6405	6389	6373	6357	6341
1950	6561	6545	6529	6514	6498	6483	6467	6451	6435	6419	6402	6387	6371	6355	6339	6324
2000	6542	6526	6510	6495	6479	6464	6448	6432	6416	6400	6384	6369	6353	6337	6322	6307
2050	6524	6508	6492	6477	6461	6446	6430	6414	6398	6382	6366	6351	6335	6319	6304	6290
2100	6506	6491	6475	6460	6444	6429	6413	6397	6381	6365	6349	6334	6318	6303	6288	6274
2150	6489	6474	6458	6443	6427	6412	6396	6380	6364	6348	6333	6319	6303	6288	6273	6259
2200	6472	6457	6441	6426	6410	6395	6379	6364	6348	6333	6318	6303	6288	6273	6258	6244
2250	6456	6441	6425	6410	6394	6379	6363	6348	6332	6317	6302	6287	6272	6257	6243	6229
2300	6440	6425	6410	6395	6379	6364	6348	6333	6317	6302	6287	6272	6257	6242	6228	6214
2350	6424	6409	6394	6379	6364	6349	6334	6319	6303	6288	6273	6258	6243	6228	6214	6200
2400	6408	6393	6378	6364	6349	6334	6319	6304	6289	6274	6259	6244	6229	6215	6201	6187
2450	6393	6378	6363	6349	6335	6320	6305	6290	6275	6260	6245	6230	6216	6202	6188	6174
2500	6378	6363	6348	6334	6320	6306	6291	6276	6261	6246	6232	6217	6203	6189	6175	6161
2550	6364	6349	6334	6320	6306	6292	6277	6262	6247	6233	6219	6204	6190	6176	6162	6148
2600	6350	6335	6320	6306	6292	6278	6263	6248	6234	6220	6206	6191	6177	6163	6149	6135
2650	6336	6322	6307	6293	6279	6265	6250	6235	6221	6207	6193	6178	6164	6150	6136	6122
2700	6322	6308	6294	6280	6266	6252	6237	6222	6208	6194	6180	6166	6152	6138	6124	6110
2750	6308	6294	6280	6267	6253	6239	6224	6209	6195	6181	6167	6153	6139	6126	6112	6098
2800	6295	6281	6267	6254	6240	6226	6212	6197	6183	6169	6155	6141	6127	6114	6100	6086
2850	6282	6268	6254	6241	6227	6213	6199	6184	6170	6156	6143	6129	6115	6102	6088	6074
2900	6270	6256	6242	6228	6214	6200	6186	6172	6158	6144	6131	6117	6103	6090	6076	6063
2950	6258	6244	6230	6216	6202	6188	6174	6160	6147	6133	6120	6106	6092	6079	6065	6052
3000	6246	6232	6218	6204	6190	6176	6162	6149	6136	6122	6109	6095	6082	6069	6055	6042
3050	6234	6220	6207	6193	6179	6165	6151	6138	6125	6111	6098	6084	6071	6058	6044	6031
3100	6223	6209	6196	6182	6168	6154	6140	6126	6113	6100	6087	6073	6060	6047	6033	6020
3150	6212	6198	6185	6171	6157	6143	6129	6116	6103	6089	6076	6062	6049	6036	6022	6009
3200	6201	6187	6174	6160	6147	6133	6119	6106	6093	6079	6065	6052	6039	6026	6012	5999

TABLE 2 APPLICATION OF DESIGN AND PERFORMANCE FACTORS TO THEORETICAL REGENERATIVE-CYCLE HEAT RATES FOR THREE POWER PLANTS

		A		B		C	
	Power plant	A		B		C	
	Throttle pressure, psia	396		628		841	
	Throttle temperature, F	693		815		899	
	Exhaust pressure, in. hg	0.89		0.99		0.98	
		Performance factor	Heat rate, Btu per kwhr	Performance factor	Heat rate, Btu per kwhr	Performance factor	Heat rate, Btu per kwhr
1	Theoretical heat rate		8237		7612		7223
2	Engine-efficiency and exhaust-loss factor	0.8130		0.8328		0.8352	
3	Cycle-design factor[a]	0.9523		0.9592		0.9625	
4	Operating efficiency[b]	0.9756		0.9752		0.9668	
5	Mechanical and electrical efficiency	0.9580		0.9698		0.9740	
6	Turbine heat rate, gross output, $\frac{\text{Item 1}}{\text{Item 2} \times \text{item 3} \times \text{item 4} \times \text{item 5}}$		11383		10075		9542
7	Auxiliary-power-usage factor	0.9270		0.9585		0.9707	
8	Evaporated make-up factor	0.9977		0.999		0.997	
9	Pipe-radiation-loss factor	0.9888		0.9929		0.993	
10	Soot-blowing and miscellaneous steam-usage factor	0.9918		0.9952		0.996	
11	Steam-generating efficiency	0.8869		0.885		0.881	
12	Plant heat rate, net output, $\frac{\text{Item 6}}{\text{Item 7} \times \text{item 8} \times \text{item 9} \times \text{item 10} \times \text{item 11}}$		14150		12030		11317
13	Plant design and performance factor	0.582		0.6328		0.6382	

[a] If and when the second stage of this study is developed, Item 3 along with many of the other items undoubtedly will be subdivided into several smaller factors for significant plant losses.
[b] Includes influence on turbine heat rate of (*a*) operation off ME point; (*b*) dirtiness of turbine blading; (*c*) departure from designed heater-terminal difference, etc.

TABLE 3 SHORT FORM FOR COMPUTING THEORETICAL REGENERATIVE-CYCLE HEAT RATES

Example: Throttle-Steam Pressure = 3200 psia; Throttle-Steam Temperature = 1200 F; Throttle-Steam Entropy = 1.5745

Column no.	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Heading	Throttle and stage pressures, psia P	Enthalpy of throttle and bled steam, Btu per lb H	Enthalpy of feedwater at saturation pressure and temperature, Btu per lb h	Boiler feed pump work, Btu per lb $F = Nv/5.4$	Enthalpy of feedwater at pump discharge, Btu per lb $h' = h + F$	Increase in feedwater enthalpy by bled-steam heating, Btu per lb $dh = h - h'$	Heat given up by bled steam in heating feedwater, Btu per lb $H - h$	Average of two adjacent readings of $(H - h)$	$\frac{dh}{\text{avg } (H - h)}$
Computation instructions	(a) Throttle pressure (b) Pressures corresponding to enthalpy shown in column (3) (c) Condenser pressure	(a) Throttle enthalpy (b) Enthalpy of steam at pressures given in column (1) and at throttle-steam entropy for isentropic expansion	(a) Enthalpy of saturated feedwater. Choose 25 to 35 steps so that the quantity dh/avg $(H - h)$ shown in column (9) will be constant[a]	N = throttle pressure minus first-stage pressure, also 1st–2nd, 2nd–3rd, etc. v = specific volume cu ft per lb at saturation temperature of lower of two pressures used to find N 5.4 is a constant $\left(\frac{778}{144}\right)$	(a) Add h for 1st to F for throttle minus 1st stage (b) Continue steps down column (c) Where F is less than 0.005 Btu, h' becomes h leaving next lower heater	Subtract data in column (5) from column (3) to determine heat added per lb of feedwater by bled steam	Column (2) minus column (3)	(a) Add first and second readings in column (7), divide by 2, and insert result in top position column (8) (b) Add second and third readings in column (7), divide by 2, and insert in second position column (8) (c) Continue down column	Divide column (6) by column (8)
Example	3200	1569.9	872.4	....	849.53	22.87	697.5	708.3	0.0322886
	3170	1568.4	849.3	0.23	825.39	23.91	719.1	729.5	0.0327759
	3099	1564.8	824.9	0.49	800.34	24.56	739.9	749.6	0.0327641
	2983	1558.9	799.6	0.74	774.37	25.23	759.3	767.9	0.0328558
	....	....	....	....	....	....	....	....	
	2.5	926.3	102.2	0.01	74.70	27.50	824.1	817.9	0.0336227
	1.16	886.4	74.7	....	47.05	27.65	811.7	805.42	0.0343299
	1 in. Hg abs	846.19[b]	47.05	....	....	....	799.14	....	....

[a] For a first approximation, it is satisfactory to divide $h_t - h_c$ into equal intervals. Having found dh/avg $(H - h)$ for the first approximation, h may be reoriented by proportion to result in obtaining approximately constant values of dh/avg $(H - h)$.
[b] Always compute end point to two decimal points using saturated-steam data in steam tables.

TABLE 4 LONG FORM FOR COMPUTING THEORETICAL REGENERATIVE-CYCLE HEAT RATES AND BOILER-FEED-PUMP WORK[a]

(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)
$\log_e (1 + w) = \int_{h_2}^{h_1} \frac{dh}{\text{avg } (H - h)}$	$\log_{10} (1 + w)$	$(1 + w)$ See footnote [b]	Average feedwater flow avg $(1 + w)$	Boiler-feed-pump work, Btu $F(1 + w)$	Work done by steam in prime mover, Btu per lb $H_1 - H$	$\frac{H_1 - H}{H - h}$	$\frac{H_1 - H}{H - h} (1 + w)$	Average of $\frac{H_1 - H}{H - h} (1 + w)$	$\frac{H_1 - H}{H - h} (1 + w) dh$	$\int_{h_2}^{h_1} \frac{H_1 - H}{H - h} (1 + w) dh = E$ Energy added to Rankine cycle
Add up column (9) step by step starting from bottom and inserting progressive total in column (10)	Change logarithm to base 10 by multiplying column (10) by 0.4342945	Read antilog of column (11) and insert in column (12)	Add adjacent readings in column (12) and divide by 2	Column (4) multiplied by column (13)	Subtract column (2) from enthalpy of throttle steam	Column (15) divided by column 7	Column (16) multiplied by column (12)	Add adjacent readings in column (17) and divide by 2	Column (18) multiplied by column (6)	Add up column (19)
0.9773690	0.4244518	2.6574	....	....	0	0	0	0.002683	0.0614	359.64
0.9450804	0.4104295	2.5729	2.6151	0.601	1.5	0.002086	0.005367	0.011266	0.2694	
0.9123045	0.3961956	2.4900	2.5315	1.240	5.1	0.006893	0.017164	0.026036	0.6394	
0.8795404	0.3819668	2.4097	2.4498	1.813	11.0	0.014487	0.034909	0.047485	1.1980	
....	....	....	....	....	....	....	....	....	....	
0.0679526	0.0295105	1.0703	1.0884	0.011	643.6	0.780973	0.835875	0.853662	23.4757	
0.0343299	0.0149088	1.0349	1.0526	0.0	683.5	0.842060	0.871448	0.888529	24.5678	
0.0	0.0	1.0000	1.0174	0.0	723.71	0.905611	0.905611	....	....	
				28.986 Total						

[a] Continuation of Short Form, Table 3.
[b] As calculated $(1 + w)$ is the theoretical flow of steam or feedwater past a chosen point in the infinite number of steps or stages of steam expansion and feedwater heating.

Discussion

Theodore Baumeister.[16] This paper is in many respects a companion piece to that offered by H. S. Horsman[9] before the British Institution in 1942, and cited by the authors. Both papers offer an ideal or theoretical standard of performance for regenerative steam-power-plant cycles.

The questions which the writer raised with reference to Mr. Horsman's paper can similarly be raised in discussion of the current paper. The criterion for judging the real worth of such a paper is in its practical utility. Will it facilitate or help in the solution of design and operating problems on actual steam power plants? If the answer is affirmative then the value of the current paper is demonstrated. If it does not give an affirmative answer then the paper is essentially an academic exercise. The authors imply, in Table 2 and its attendant paragraphs, that much additional work is necessary to accomplish this important affirmative step. Presumably it is the result of their experience that the theoretical heat rates of Table 1 can be used for the comparison of two proposed real alternate cycles. Or better, upon accumulation and correlation of further experience and data, Table 1 may then be used to answer the recurrent question of choice between two real alternate power-plant cycles.

Thus from detailed and accurate calculations the over-all heat rate may have been determined as 11,000 Btu per kwhr of net output, including all allowances for inefficiencies and auxiliaries. The appropriate theoretical heat rate of the idealized cycle might be read from Table 2, as 6945 Btu per kwhr. If the same arrangement of equipment and the same inefficiencies were applicable then some second cycle, which might have a threoretical heat rate of 6759 Btu per kwhr from Table 2, could then be estimated to give an actual over-all thermal performance of

$$11{,}000\,\frac{6759}{6845} = 10{,}860 \text{ Btu per kwhr net output}$$

This procedure, of course, would be a great convenience and reduce the tedium and errors of lengthy heat-balance calculations. Unfortunately, however, the mechanism or engine efficiencies will usually change along with the cyclic values, and the solution is not as convenient as may be anticipated.

In any event, high orders of accuracy are necessary because the power engineer is concerned with the evaluation of savings, i.e., differences in heat rates. These differences are small numbers and unless a high degree of accuracy is otained the answers, because of too few significant figures, cannot be used. The writer therefore would like to inquire if the authors believe that their paper can be so used to give reliable and convenient answers to the following typical questions which arise in the development of a plant cycle:

1 Proposed alternate use of drip pumps, drain coolers, or direct flashing for heater drains.

2 Terminal temperature differences to be used on closed feed heaters.

3 Proper disposition of turbine gland leakage.

4 Make-up evaporator location in the cycle.

5 Thermal advantages of alternate locations of feed pumps in the cycle.

6 Recovery or nonrecovery of heat supplied to generator coolers and lubricating-oil coolers in the feedwater.

7 Electrically driven, steam-driven or internal-combustion-engine-driven plant auxiliaries.

8 Pressure drop between turbine shell and heater.

If the authors believe that the solution of these and similar problems can be expedited by the methods outlined in their paper, then it is the writer's opinion that they should be encouraged to collect and correlate the necessary data for the preparation of the supplementary paper which they recommend.

[16] Professor of Mechanical Engineering, Columbia University, New York, N. Y. Mem. A.S.M.E.

M. W. Benjamin.[17] This contribution to steam-power engineering provides, in Table 1, data which are just as fundamental to the regenerative cycle as are the tables of theoretical steam rates to the Rankine cycle. The practical usefulness of the theoretical heat rates will depend to a large extent upon the successful working out of a second paper to provide the factors necessary for bridging the gap between the hypothetical and the practical. In years past the existence of such information could have saved unknown man-hours of calculation in the planning of new plants and rehabilitation projects.

As applied to the design of new plants, the value of this paper and its proposed companion may be limited henceforth to the investigation of projects outside the range of the standardized turbines now being considered by the industry. Performance figures for standard units will be well known, and calculations will be unnecessary except for minor changes in the cycle. It is doubtful whether the companion paper could provide factors for all such possible refinements. It would be a mistake to assume, however, that the present paper had come too late merely because turbine standardization is now being considered seriously. Until some radically new method of producing electricity is devised, it would seem reasonable to believe that engineers will continue to explore the possibilities of the regenerative cycle. The theoretical heat rates are now established with a high degree of precision, and from this paper one can determine quite easily the order of magnitude of improvement which still is possible. It appears from Fig. 5(a) and Table 2, that between good modern practice and the heat rate of an actual plant at the upper limit of this study (3200 psi at 1200 F), there is a margin of at least 1500 to 1700 Btu per kwhr (440 to 500 whr per kwhr).

Figure 5(a) also illustrates a point that has been made before, viz., that, as pressures and temperatures are increased, the rate of thermal economy improvement becomes smaller, thus making it more difficult to justify the improved conditions economically.

The authors have indicated, without elaboration, another use for these data in conjunction with the companion paper in establishing reasonable standards of plant operation by means of which it will be possible to estimate the degree of excellence with which a plant is operated. For many years it has been the practice in England to refer to some kind of standard for this purpose whereas in our country the emphasis has been more on designing the plant for economy and ease of operation and then expecting the operating staff to get the utmost in efficiency from the equipment that has been provided. One weakness of this arrangement is that it offers no incentive for careful operation of older plants in competition with newer ones. Perhaps Americans do not require a measuring stick for operating excellence, but for those who see merit in the idea the present paper offers data for determining a basis of comparison of regenerative-cycle plants.

The theoretical heat rates given in Table 1 are the result of a large amount of careful work for which the industry is grateful. The paper may develop into a tool of many uses.

A. A. Brooks.[18] The writer believes that the requirement $dh(H - h)$ be constant, could be eliminated by using a sufficiently large number of steps with possibly less labor involved than recalculation to get equal steps.

[17] Engineering Division, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

[18] Thermodynamic Engineer, Moore Steam Turbine Division, Worthington Pump and Machinery Corporation, Wellsville, N. Y.

No practical purpose is served for limiting the regeneration to saturated steam only. The use of superheated steam is simply a question of proportioning the heaters intelligently. It is doubtful if, in actual practice, it is feasible to get higher than the saturation temperature of the steam in the heater, and it would be interesting to see how figures on this basis compare with the author's figures. It is not clear whether the curves in Fig. 6, are given on this basis or on the basis of a perfect counterflow system for superheated steam.

That the cycle efficiency of better than 50 per cent for steam pressures has already been approached in actual practice is extremely interesting.

W. W. Dulmage.[19] In Table 2 of their paper Messrs. Selvey and Knowlton have applied design and performance factors for relating the operating results from three actual plants to the respective theoretical regenerative-cycle heat rates. Since further information of the same sort is needed for other plant cycles, the writer is pleased to furnish the following description of the cycle for River Rouge No. 1 power plant of the Ford Motor Co. with operating results for the year 1943.

In that power plant there are three steeple-compound condensing turbine-generators of 110,000-kw capacity each, and one 15,000-kw noncondensing turbine-generator. One of the large units operates on the reheating-regenerative cycle using throttle steam at 1200 psi 725 F and employing live-steam reheat to about 570 F at the crossover. The other two large units operate on the regenerative cycle without reheat, using throttle steam at 1200 psig 900 F. The noncondensing machine operates on throttle steam at 1200 psig 900 F and exhausts to large process evaporators at 250 psig. Since the paper deals only with the regenerative cycle the following remarks apply only to the two large units operating on that cycle, and are based on over-all plant data for the calendar year 1943 as adjusted to represent the performance of those two generating units. Several complications arise when attempting to allocate properly between the different cycles existing in the power plant such items, among others, as auxiliary power and soot-blowing steam. Therefore several of the factors of the authors' Table 2 have been grouped together in estimating the performance factors presented in Table 5 for the straight regenerative units. The cycle diagram for one of these units is given in Fig. 7 while the turbogenerator heat-rate curve obtained from tests[20] and corrected to standard conditions of throttle pressure and temperature and exhaust pressure is shown in Fig. 8.

[19] Power Department, Ford Motor Company, Dearborn Mich.

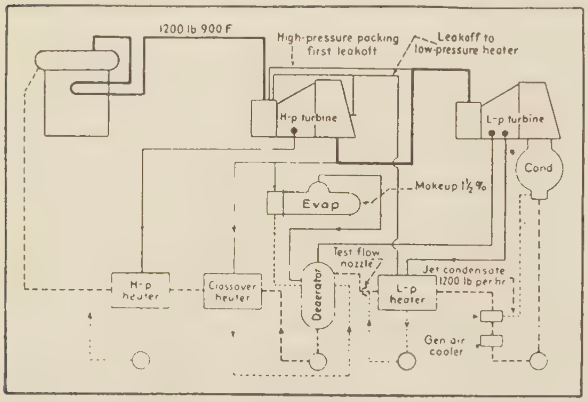

Fig. 7 Flow Diagram for Two 110,000 Kw Steeple-Compound Turbine-Generators Operating on 1200 Psig 900 F Regenerative Cycle in River Rouge No. 1 Power Plant of Ford Motor Co.

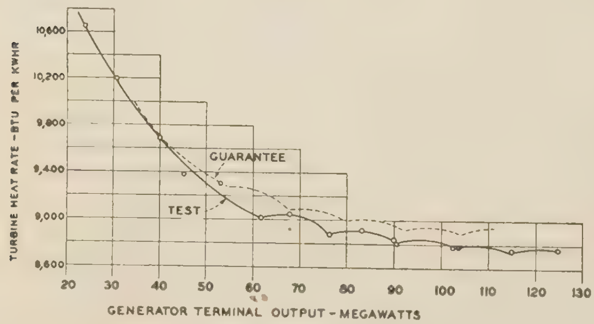

Fig. 8 Guarantee and Test Heat Rates for 110,000 Kw Steeple-Compound Turbine Generators Operating on 1200 Psig and 900 F Regenerative Cycle in River Rouge No. 1 Power Plant of Ford Motor Co.

Average values of the pertinent data for the year 1943 are as follows:

(1)	Over-all station heat rate, Btu per net kwhr	11,220
(2)	Over-all station coal rate, lb dry coal per kwhr	0.795
(3)	Heat value of coal, Btu per lb dry coal	14,107
(4)	Over-all boiler-room efficiency, per cent	87.1
(5)	Feedwater make-up, per cent	2.4
(6)	Auxiliary power, per cent of gross generation	4.5
(7)	Average load on regenerative units, kw each	52,600

TABLE 5 APPLICATION OF DESIGN AND PERFORMANCE FACTORS TO THEORETICAL REGENERATIVE-CYCLE HEAT RATES FOR RIVER ROUGE NO. 1 POWER PLANT OF FORD MOTOR CO.

		Factor	Btu per kwhr
(1)	Theoretical heat rate		6935
(2)	Engine-efficiency and exhaust-loss factor	0.7497	
(3)	Cycle design factor		
(4)	Operating efficiency[a]		
(5)	Mechanical and electrical efficiency		
(6)	Turbine heat rate, gross output (at average load)[a]		9250
(7)	Auxiliary-power usage factor	0.9465	
(8)	Evaporated make-up factor		
(9)	Pipe-radiation-loss factor		
(10)	Soot-blowing and miscellaneous-steam-usage factor		
(11)	Steam-generating efficiency	0.871	
(12)	Plant heat rate, net output		11,220
(13)	Plant-design and performance factor	0.6181	

[a] Average load was less than 50 per cent of rated capacity, which results in low value of item called Operating Efficiency in Table 2 of Selvey-Knowlton paper.

G. M. Dusinberre.[21] The authors ask for an explanation of the curvature of line *A-B* in Fig. 2(*b*). This line, by its deviation from the vertical, reflects the departure from reversibility in the feed-heating operation. This deviation is greatest at the top. Under the assumption of an infinite number of feed heaters, we are at this point using steam at 1200 F to heat water at 705 F, evidently with great thermal irreversibility. The situation is also apparent from the fourth line of Tables 3 and 4. Column 12 shows that we have extracted about 10 per cent of the throttle steam and column 2 shows that the through steam has done only 11 Btu of work per lb. This is not much different from heating the feed irreversibly in the boiler.

In their Appendix the authors state that a long form of computation would be necessary for investigation of the isothermal-compression regenerative cycle. This is true as regards pump and compressor work. But for net work or net heat rate, the calculations would be extremely simple. Such a cycle could ideally be made entirely reversible, internally, and the heat rejected per pound of throttle steam would be simply the tempera-

[20] See "Performance Test at Ford Motor Co.," by W. W. Dulmage and C. J. Walker, *Power*, vol. 85, July, 1941, p. 72.

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ture at rejection times the change of entropy during heat supply. Thus, the "short-cut" formula, shown above Fig. 3 of the paper, would be precisely correct for this cycle.

The writer was impressed with the suggestion by A. A. Markson of an analysis based on a divided cycle and has prepared a brief outline of the form which the proposed tables might take. It is true that the work done by the saturated cycle can be obtained by multiplying its Carnot efficiency by the heat rejected by the superheat cycle, but there are certain advantages in having additional tables instead (Tables 6 and 7 of this discussion).

TABLE 6 SUPERHEAT-REGENERATIVE DATA

p	t	800	1000	1200
3200	q	378.7	562.5	697.5
	q'	316.7	500.5	635.5
	w	45.1	145.1	242.8
	y	0.570	0.531	0.524
	p_s	1343	385.8	141.0
(385.8)	q		1102.8	
	q'		318.5	
	w		337.1	
	y		0.786	
	p_s		13.0	

TABLE 7 SATURATED-REGENERATIVE DATA

p	t_f	90	(79.03)	70	50
1400	w	273.0		283.9	294.9
	y	0.445		0.432	0.420
(1343)	w		282.0		
	y		0.446		
(385.8)	w		314.6		
	y		0.624		
(141.0)	w		293.0		
	y		0.720		
14	w	173.7		202.7	231.8
	y	0.881		0.861	0.842
(13.0)	w		185.2		
	y		0.875		

p—throttle (or reheat) pressure, psia
t—temperature at throttle (or after reheating), F
q—heat supplied to throttle steam, Btu/lb
q'—heat supplied to reheated steam, Btu/lb
w—work (net, turbine less feed pump) per lb steam supplied, Btu/lb
y—ratio of through steam to throttle steam
p_s—pressure at saturation, for isentropic expansion, psia
t_f—condenser (or cooling water) temperature, F

The method of calculation was similar to the authors' short form except that no effort was made to keep the feed-heating increments equal as long as they were sufficiently small. The use of the tables may be illustrated by an example:

Throttle steam 3200 psia, 1200 F, lowest temperature 79.03 F (this temperature is chosen to agree with the authors' 1 in. Hg).

	Steam	Heat	Work
From superheat-regenerative Table 5	1.000	697.5	242.8
To saturated-regenerative at 141 psia	0.524		
Interpolating for 141 and 79.03			
Work, 293.0 × 0.524			153.5
Steam to condenser 0.720 × 0.524	0.377		
Totals		697.5	396.3

Heat rate 3413 × 697.5/396.3 = 6000 Btu per kwhr (5999 by authors)
Throttle steam/through steam 1/0.377 = 2.65 (2.6574 by authors)

So far, we have little that could not have been obtained more readily from the authors' tables. But what about reheat? Is it not possible and desirable to set an ideal standard for regenerative plants which is also applicable when reheat is employed?

The advantage of reheat lies in the reduction of moisture in the low-pressure stages of the turbine, rather than in any striking gain in ideal-cycle efficiency. Since this is the case, it would be logical to base a "standard" cycle on reheat at the pressure where the throttle isentropic reaches the saturation line. This would mean that reheat actually began with a moderate amount of superheat, which is not out of line with current practice.

If such a standard should be acceptable, the analysis of the ideal cycle can be made very readily with Tables 6 and 7, for example: Throttle steam 3200 psia, 1000 F, reheat to 1000 F, lowest temperature 79.03 F.

	Steam	Heat	Work
From superheat regenerative Table 5	1.000	562.5	145.1
To reheat regenerative at 385.8 psia	0.531		
Heat (use q') 318.5 × 0.531		169.1	
Work 337.1 × 0.531			179.0
Steam 0.786 × 0.531	0.417		
To saturated regenerative at 13.0 psia			
Work 185.2 × .417			77.2
Steam to condenser 0.875 × 0.417	0.365		
Totals		731.6	401.3

Heat rate 3413 × 731.6/401.2 = 6220 Btu per kwhr

This division of the cycle should lend itself better to the application of performance factors. The efficiency of a turbine, operating only in the superheat region, and of another operating only saturated, can be set forth in a more representative way than the efficiency of a turbine operating in both regions.

The saturated-regenerative Table 7, based on temperatures, can be used for condensation temperature or, as suggested by P. H. Hardie, for cooling-water temperature. This latter seems more logical in a study of purely ideal performance.

M. D. Engle.[22] A standard of measurement is a fundamental requirement of engineering progress. Many business enterprises having to do with the sale of a service have been really unsuccessful until a suitable device was developed to measure the service supplied. The electrical industry was greatly hindered in development until a suitable kilowatthour meter was developed. The hot-water district-heating business has never fully developed and one of the principal reasons for this lack of success is the nonexistence of a suitable device for measuring the service supplied. On the other hand, the district steam-heat business, having suitable measuring devices, is growing quite rapidly.

So it is in the business of designing and constructing steam-electric generating stations. While it is true that big advances have been made in design, we have long needed a suitable standard of measurement of the success of our design.

The net Btu per kwhr performance of a station is a useful factor but it does not really show how closely we have approached the ideal performance for the steam conditions and cycle adopted.

The data in this paper will be very useful to powerplant design engineers in calculating just how good a job they have done.

Furthermore, the careful analysis of the losses, as shown in Table 2, will more clearly focus the attention of the designers upon the particular items to be investigated for possible improvement.

We have been so much interested in the problem of keeping the turbine blades clean that probably very few have given much thought to the loss in station economy due to the blade deposits. Maybe such an analysis will show that we are justified in spending much more money in reducing the amount of deposit.

Likewise such an analysis will focus our attention upon the loss due to excessive auxiliary power usage, evaporated make-up losses due to avoidable steam leaks and condenser leakage, and the losses due to so-called "soot-blowing." (In this day and age that term should really be changed to "ash and slag" blowing.)

The operating engineers should also become more fully acquainted with such a detailed analysis of the possibilities of, and losses in, the stations which they are operating. They have of course studied station losses for years but, when they have the theoretical possibilities of their plant in mind, a more careful analysis of the losses will quite often pay handsome dividends.

[22] Assistant Superintendent of Engineering, Boston Edison Co., Boston, Mass. Mem. A.S.M.E.

The writer feels that both design and operating engineers will benefit greatly by the work done by the authors and is sure that they will co-operate by supplying the data which are needed to complete the study.

P. H. Hardie.[23] The table of theoretical regenerative steam-cycle heat rates presented by the authors is a valuable contribution. The authors' suggested method of using this table as a basis for appraising plant performance should be adopted as an American standard, with possibly a slight modification which will be discussed later.

There are several points relating to the application of this table worth careful consideration. For example, the authors have proposed using the throttle steam conditions for the upper limit of the cycle instead of drum or superheater-outlet steam conditions. From a practical standpoint this would seem to be the best choice. However, for the lower boundary of the cycle, it would seem preferable to use the saturation pressure corresponding to the temperature of the condensing water, rather than the turbine exhaust pressure. This pressure is the theoretical minimum and will be more nearly constant for all units in a plant. Furthermore, it will not be affected by the differences in design, load, and cleanliness factor of the different condensers. These factors should properly appear in the plant-design-and-performance factor. If the authors agree to this change it would be helpful to have the data, presented in Fig. 5(*b*), replotted for condensing-water temperature.

No method has been proposed by the authors for computing the theoretical regeneration-steam-cycle heat rate of a plant having units operating at two or more pressures or temperatures. The theoretical heat rates of the different systems would have to be weight-averaged, but what is the proper basis of weighting? Boiler-steam output would not be satisfactory because of the differences in the enthalpy of the steam and/or the feedwater. Generator output would also be unsatisfactory because some of the steam may be throttled from the high-pressure system to the low-pressure system. Therefore fuel input would seem to be the only satisfactory basis for weight-averaging the theoretical regenerative-steam-cycle heat rates to determine the average for such plants.

The authors have shown good judgment in refraining from proposing the use of their table of the theoretical heat rates for appraising turbine performance. Barnard and Ellenwood's method of computing the ideal regenerative-cycle heat rates, based upon the actual number of heaters, is a more equitable standard for turbines. However when appraising the entire plant, the authors' tabular values for an infinite number of heaters should be used.

M. L. Ireland, Jr.[24] It would appear that the usefulness of the table of theoretical regenerative-steam-cycle heat rates presented in this paper can be increased by further comparison with the results obtained by "short-cut" formulas.

In the first place the comparison shown in Fig. 3 may give a somewhat exaggerated impression of the magnitude of error involved in the stated short-cut formula. When this error is expressed as a percentage correction and plotted against throttle temperature as shown in Fig. 9 of this discussion, it becomes apparent that up to about 2300 lb pressure, the correction is practically constant for a particular temperature, varying from about 0.2 per cent for 800 F to 1.2 per cent for 1200 F.

The necessity for a carefully prepared table such as is presented by the authors is particularly evident when considering steam temperatures above 1000 F and pressures over 2300 lb. However, the use of this table as a reference standard for actual power-plant cycles is subject to the basic disadvantage that actual cycles do not employ complete regeneration. A criterion based on actual or designed final feedwater temperature appears to be necessary. The short-cut formula, suitably corrected for additional feed-pump work done above the saturation pressure, corresponding to the final feed temperature, does supply such a criterion, but it is admittedly approximate. On the other hand, the computation of accurate results for individual cases by the authors' tabular integration method is quite laborious.

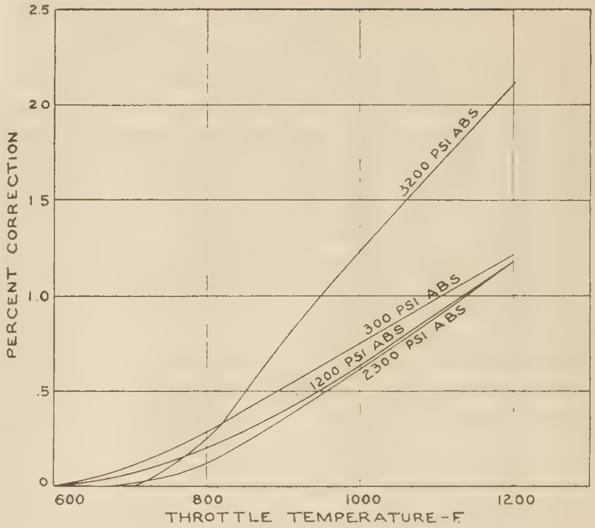

Fig. 9 Correction to Approximate Regenerative-Steam-Cycle Heat Rates Obtained by "Short-Cut" Formula

It is understood that the authors intend to prepare a companion paper in which many such factors will be considered; however, in the meantime it would be helpful if the authors can suggest a means to make further corrections to the short-cut formula for computing heat rates of partial-regeneration cycles. It has occurred to this writer that, since the errors of the short-cut method are associated with regeneration by superheated steam, it might be satisfactory to use the correction curves of Fig. 9, at the pressure and temperature on the turbine-expansion line which correspond to the final feed temperature and saturation pressure.

A. A. Markson.[25] One who estimates the value of this paper by its literal applicability to design problems may be disappointed, but the student of such matters will find in it exactly what he needs. The cycle chosen by the authors is of high conceptual usefulness. It is defined incompletely in the paper despite the careful assumptions given. The additional statement that nowhere in such a cycle is there a degradation of energy which is not inherent in the cycle itself would complete the definition, as it would include frictional pressure drops, supersaturation, etc.

The authors have wisely chosen to represent this cycle in tabular form, giving efficiencies correlated with different throttle conditions and a constant condensing pressure. The writer believes that a conceptually more logical method of constructing such tables might be of interest. Inasmuch as the saturated region has Carnot efficiency, there is at once an alternate method of

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[24] Newport News Shipbuilding and Dry Dock Company, Newport News, Va. Mem. A.S.M.E.

[25] Hagan Corporation, Pittsburgh, Pa. Mem. A.S.M.E.

tabulation which eliminates the necessity of assuming a constant condenser pressure.

Starting with 1 lb of throttle steam as the basis, using throttle pressure and temperature as the main tabular entries, the efficiencies could be tabulated for the superheated expansion zone alone. The pressure at which the isentropic expansion crosses the saturation line is also tabulated. This completes the skeleton of the tables, but for convenience, the weight ratio passing to the saturated zone might be given. To use these tables to any saturated back pressure whatever, the user simply calculates the Carnot efficiency of his saturated cycle and makes a simple weighted average of this and the superheated cycle. This would eliminate the necessity for the exhaust-pressure corrections used by the authors, thus gaining considerable simplicity of concept.

The writer closes by suggesting that performance figures, such as shown in Fig. 5(*a*), be referred to as the "thermodynamic merit" of the cycle to avoid possibility of misunderstanding. Thus an actual cycle having a certain heat rate could be assigned an efficiency analogous to Rankine efficiency on the basis of the "thermodynamic merit" figure.

C. A. Meyer.[26] It appears to the writer that a simpler and more accurate definition for the theoretical regenerative-steam-cycle heat rate can be obtained by utilizing the well-known concept of available energy.

In any cycle in which heat (dQ) is added at variable temperature T (deg abs) and rejected at constant temperature T_e (deg abs), the maximum work obtainable (W_M) is given by

$$W_M = \int_{T_1}^{T_2} dQ\left(\frac{T - T_e}{T}\right) = Q_{1-2} - T_e \int_{T_1}^{T_2} \frac{dQ}{T} = Q_{1-2} - T_e(S_2 - S_1) \quad [1]$$

At constant pressure, the heat added is equal to the enthalpy rise

$$Q_{1-2} = h_2 - h_1$$

so that

$$W_M = (h_2 - T_eS_2) - (h_1 - T_eS_1) = A_2 - A_1 \quad [2]$$

where $A = h - T_eS$ is the total available energy measured above an arbitrary state point. The available energy at any state point is obtained as the difference between the value of A at the given state point and the value of A at the state point which we regard as having zero availability (atmospheric temperature and pressure). The relation Equation [2] is quite general and could have been written directly.

The maximum thermal efficiency of the regenerative cycle is then

$$\eta_M = \frac{(h_t - T_eS_t) - (h_f - T_e S_f)}{h_t - h_f} = 1 - \frac{T_e(S_t - S_f)}{h_t - h_f} \quad [3]$$

The theoretical heat rate is then

$$THR = \frac{3412.75(h_t - h_f)}{h_t - h_f - T_e(S_t - S_f)} \quad [4]$$

where h_t, S_t = throttle enthalpy and entropy, respectively
h_f, S_f = feedwater enthalpy and entropy, respectively
T_e = absolute temperature of cooling water

Relation, Equation [4], has been labeled "inaccurate" in the paper for reasons which are not clearly stated. However, Equation [4] gives the correct heat rate of the ideal cycle, defined by the four specifications given in the paper (perfect processes allowing for required pump work).

It is of interest to show that the theoretical heat rate of any cycle can be obtained in a similar manner. As an example, the efficiency of the Rankine cycle is easily written as

$$\eta_R = \frac{A_t - A_f}{h_t - h_f} = \frac{(h_t - T_e S_t) - (h_f - T_e S_f)}{h_t - h_f} \quad [5]$$

Since

$$T_e(S_2 - S_c) = h_2 - h_c \qquad S_2 = S_t$$
$$S_c = S_f \quad \text{and} \quad P = h_f - h_c$$

where P = pump work
h_c, S_c = enthalpy and entropy of condensate
h_2, S_2 = enthalpy and entropy at turbine exhaust

Equation [5] becomes the well-known expression for Rankine-cycle efficiency

$$\eta_R = \frac{h_t - h_2 - P}{h_t - h_c - P} \quad [6]$$

Returning to the regenerative cycle, a further point of interest is that factors correcting for the various inefficiencies in actual cycles can easily be defined. For instance by setting down the loss of available energy in an extraction turbine, a suitable correction factor for the inefficiency of the turbine can be obtained. The loss of available energy in a single extraction turbine is written as

$$L = G_tA_t - G_1A_1 - G_eA_e - W$$

where L = loss of available energy in turbine
W = useful work output
$= G_t(h_t - h_1) + (G_t - G_1)(h_1 - h_e)$
G = flow
A = available energy $= h - T_eS$

Subscripts t, 1, e refer to throttle, bleed point, and exhaust, respectively.

The cycle efficiency (for losses in the turbine only) then becomes

$$\eta = \eta_M \times C_t = \left(\frac{A_t - A_f}{h_t - h_f}\right) \times \left(1 - \frac{L}{A_t - A_f}\right)$$

where $C_t = \left(1 - \dfrac{L}{A_t - A_f}\right)$ = turbine efficiency factor

Similar correction factors for terminal difference, mixing and pressure losses are easily obtainable from the losses in available energy due to these causes.

Ernest L. Robinson.[27] Just 20 years ago the writer presented a chart showing the efficiencies of the theoretical extraction cycle in comparison with efficiencies obtainable with the Rankine cycle.[28] This was a small-scale diagram outlining in a general way the possibilities of cycle improvement. This diagram was extended to a temperature of 700 F and a pressure of 1000 psia, limits which were then thought to represent the fringes of possibility.

The calculations on which these efficiencies were obtained were based upon the Goodenough chart of 1917. Although we did not then know it, the Goodenough steam table at that time was an

[26] Research Engineer, Westinghouse Electric & Manufacturing Co., South Philadelphia Works. Mem. A.S.M.E.

[27] Turbine Engineering Division, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[28] "The Margins of Possible Improvement in the Central Station Steam Plant," by Ernest L. Robinson, Trans. A.S.M.E., vol. 45, 1923, p. 644.

excellent steam table. All circumstances considered, the efficiencies then given corresponded remarkably well with the heat rates given in this paper on the basis of present-day steam tables up to the values then used.

Ever since the steam tables have gone through the process of refinement and improvement to which they have been subjected during the past 20 years, the writer has felt that theoretical heat rates for the extraction cycle should be determined with the same degree of precision as the theoretical water rates for the Rankine cycle. This has never been done until the authors undertook the task a couple of years ago with the result which is now presented.

The authors have suggested certain further steps toward the analysis of station performance. The writer would like to mention one or two other investigations into the nature of steam which might well be undertaken.

The question of reheat factors has never been satisfactorily cleared up. Three important contributions on the subject appeared several years ago, one in England and two in this country. The excellent British article by Dr. D. M. Smith[29] was based upon an unjustified use of the moisture correction, and another paper by Dr. R. B. Smith[30] was based upon certain analytical assumptions. The writer would prefer to see reheat factors based strictly upon the steam table without any analytic assumptions. Such an attempt was made by Prof. C. G. Thatcher.[31] However, the writer's attempt to apply Professor Thatcher's results differed by as much as 1 per cent in turbine performance from Professor Thatcher's application of his own results. Without being completely in the dark as to the reasons for this uncertainty, it is evident that there is still a job to be done in the matter of reheat factors.

The evaluation of reheating effects in connection with the extraction cycle requires determination of the weight flow at various stages in the turbine and heat rates corresponding to actual stage efficiencies. The writer previously charted these quantities for expansion lines at 70 and 80 per cent efficiency.[32] Professor Thatcher's reheat factors were based on 85 per cent stage efficiency from which suitable correction to neighboring efficiencies may be made. A table of heat rates for a theoretical extraction cycle with 85 per cent stage efficiency would be a welcome contribution.

In addition to the heat rates to 1 in. Hg abs back pressure, it is advantageous for a number of reasons to know the weight flow at any point in the course of the expansion through the turbine. A knowledge of the weight flow is necessary for the proportioning of the areas of cross section of the many successive passages throughout the turbine. This is particularly true of the extraction cycle where the steam entering the throttle is likely to be in the order of 50 to 100 per cent more than what enters the condenser. This is the quantity $(1 + w)$ which appears throughout the formulas and tables in the paper. Mollier charts for the weight flow per pound of steam at 1 in. Hg abs back pressure were given in the writer's article[31] for expansion lines of 70 and 80 per cent efficiency. Under modern conditions, a chart for 85 per cent efficiency would be more useful.

A further advantage of the weight-flow data lies in the ability to determine over-all performance between any two levels of pressure without resorting to superheat, pressure, or vacuum corrections. The vacuum corrections given in the paper are good as long as the expansion ends in the wet region. However, if one wishes to know the performance of the high-pressure element of a cross-compound machine, it may be determined directly when the weight flows entering and leaving are known.

Data of this sort automatically include the evaluation of the reheating effect and indicate the energy actually available from the steam when expanding at a definite stage efficiency. Of course, in any case there must be suitable corrections for the deviation from an infinite number of turbine stages and points of extraction.

However, when weight-flow data are available, one may readily determine the heat rate and corresponding thermal efficiency for any particular initial steam conditions and back pressure. It is then approximately correct to assume a linear relation between the thermal efficiency and the stage efficiency. Thus, knowing the thermal efficiency for 85 and 100 per cent stage efficiencies, it may readily be determined for any stage efficiency between 80 and 90 per cent. Of course, the linear assumption could be avoided by preparing data at both 80 and 90 per cent as well as for 100 per cent. With such a line drawn for the steam conditions in question, it is as easy to determine the internal efficiency corresponding to a test result as it is to determine the expected performance corresponding to the internal efficiency of a design.

The writer believes this procedure to be one of the best methods, if not indeed the best, of taking complete account of the influence of the steam conditions on the turbine performance. (It does not deal with stage design.) Beyond this point the analysis of station performance might well follow the lines outlined by the authors.

J. Kenneth Salisbury.[33] In most studies of the regenerative cycle, the ultimate objective is to simplify the evaluation of power-plant turbine cycles, and to facilitate the comparison, on an extraction basis, of different steam conditions used with a given heater arrangement. At least two methods of approach are possible.

One is exemplified by the material presented by the present authors. They use as a basis the best possible heat rate which can be obtained in a conventional steam power plant, namely, the heat rate of a turbine having 100 per cent efficiency (and an isentropic expansion line) and heating its own feedwater in an infinite number of heaters to the saturation enthalpy corresponding to throttle pressure. Such a heat rate, when divided by the turbine-generator efficiency, yields the turbine-cycle heat rate, if a correction is made to a finite number of heaters operating at the desired final feedwater temperature, and "cycle corrections," etc., are neglected.

A second method of approach is that used by the writer in a paper presented before the A.S.M.E. in 1941.[34] In this method the actual nonextraction heat rate (including the actual turbine-generator efficiency) is used as a starting point. To this nonextraction heat rate is applied again factor $(1 - PG)$, which depends upon such parameters as the number of heaters used, the steam conditions, and the feedwater temperature. A small additional correction is required to obtain the actual heat rate because of departures of the actual cycle from the assumed cycle (a series of equally spaced contact heaters, each having no pressure drop in the extraction line, and no terminal difference).

[29] "Stage Efficiency, Cumulative Heat, and Reheat Factor of Steam Turbines," by D. M. Smith, Proceedings of The Institution of Mechanical Engineers, vol. 140, 1938, pp. 399–452.

[30] "The Calculation of Steam-Turbine Reheat Factors," by R. B. Smith, *Journal of Applied Mechanics*, Trans. A.S.M.E., vol. 60, 1938, pp. A-156–A-160.

[31] "Reheat Factors for Expansions of Superheated and Wet Steam," by C. G. Thatcher, *Journal of Applied Mechanics*, Trans. A.S.M.E., vol. 60, 1938, p. A-161.

[32] "Notes on the Comparison of Steam-Turbine Efficiencies," by Ernest L. Robinson, *General Electric Review*, vol. 29, 1926, pp. 503–510.

[33] Turbine Engineering Division, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[34] "The Steam-Turbine Regenerative Cycle—An Analytical Approach," by J. Kenneth Salisbury, Trans. A.S.M.E., vol. 64, 1942, pp. 231–245; see also, "A Short Cut to Turbine Cycle Heat Rates," by J. Kenneth Salisbury, *Power*, Oct., 1943, pp. 78–81, 144, 146, and Nov., 1943, pp. 78–80.

Neither of the two general methods can, in a simple straightforward manner, take care of all of the combinations and permutations which can be conceived by power-plant designers. It is necessary therefore that certain "cycle-correction" factors be applied for either method to take care of departures from the standard cycles. Such correction factors should meet the following requirements:

1 There should be a minimum number of corrections, for ease in handling.

2 The cycle corrections should be clearly defined and arranged so as to be easily applied to heat rates determined by test of operating units, thus greatly reducing the uncertainties and drudgery now involved in correction to standard conditions of such tests.

3 The corrections should, when possible, be independent of each other so as to permit isolation of the various losses and independent appraisal of various features of the cycle.

4 The corrections should, of course, be accurate, relative to the over-all heat rate, but since they are small, considerable tolerance may be allowed in the corrections themselves.

The writer's original conception of the problem was that, as a first step, a method should be devised which would properly consider the major variables, such as the following:

1 Number of heaters
2 Feedwater temperature
3 Operating steam conditions
4 Turbine efficiency.

This was accomplished in the original paper.[34] It was considered that the second step should be calculation of a set of correction factors, the function of which would be to increase the accuracy of the calculations made using the first set of data but which, in the meantime, could be estimated from experience. The entire analysis to be covered in the second step would be devoted to the accurate determination of the 1 to $1^1/_2$ per cent (total) correction to the heat rate found using the original data.

The present authors have embarked on a similar task by tabulating the heat rates for theoretical cycles at the optimum feedwater temperature (that is, saturation) for an infinite number of heaters. These heat rates are listed for intervals which are closer than required for accurate interpolation, but which are desirable for convenience in use with the present steam tables. The next step in the analysis, that is, corrections to these tabulated heat rates for cycles having a finite number of heaters at various final feedwater temperatures other than the optimum for an infinite number of heaters, will no doubt eventuate to be closely related to those given in the references cited.[34]

The necessary work has been completed, and will be published, on a method of correcting heat rates from the original somewhat idealized basis to the actual cycle.[35] All of these correction factors should be adaptable, with little change, to the present authors' data, with the exception of the boiler-feed-pump correction. An additional correction will, however, be required to correct the authors' state line (an isentropic line) to the actual case, that is, a state line having the applicable efficiency.

The corrections which have been analyzed by the writer are for the following:

1 Heater terminal difference.
2 Pressure drop in the extraction piping.
3 Flashing of one or more heaters to lower heaters with or without drip coolers, of any terminal difference.
4 Distribution of heaters.
5 Evaporators in cycle.
6 Heat-recovery apparatus in cycle.
7 Boiler-feed-pump loss.

(Suggestions are solicited as to other correction factors which might be required in heat-balance calculation.)

The present authors have used an isentropic state line which is never encountered in actual practice. It has, however, the merit of being easily and precisely defined, and of having the absolute support of classicism. The gain due to feedwater heating (the heat rate is not relevant, at the moment—only the gain) depends upon an average value of $(H - h)$, which was integrated essentially graphically by the present authors and analytically by the writer.[34] Since $(H - h)$ at a given extraction pressure or value of h depends only upon the location of the state line, the gain which is obtained by feedwater heating is a function both of the turbine efficiency and the assumed curvature, for given steam conditions. In obtaining actual heat rates from the authors' theoretical heat rates, an additional correction will therefore be required for these items. The correction may be from 0.5 to 1 per cent. This difference in state line assumption, plus the difference in the assumption relative to the boiler-feed-pump work, prevents valid cross checks between the authors' work and the writer's work. Making estimates of the effect of the foregoing items, it is, however, possible to arrive at reasonably close checks between the two sets of data.

A correction will have to be provided by the present authors for the use of a single boiler feed pump (or two boiler feed pumps) instead of the infinite number of pumps which have been used in their study. It is believed that the authors' data, as presented, properly charge the boiler-feed-pump loss for an infinite number of pumps having 100 per cent hydraulic efficiency, in that the power required to run the pumps is deducted from the main output, and the increase in enthalpy realized from the pumps is credited as heat. The data presented in the paper do not, without amplification, readily permit taking into account the effect of using a finite number of pumps, nor are the effects of extra pumping pressure and pump efficiency readily evaluated.

The writer evaluated the expression

$$\left(\frac{\text{P.W.}_\infty}{\text{P.W.}_1}\right)_{1\text{lb}} = \frac{\int_0^{P_1} VdP}{V_0(P_1 - P_0)}$$

(which neglects the small compressibility effect) to obtain an approximation of the power when using an infinite number of pumps

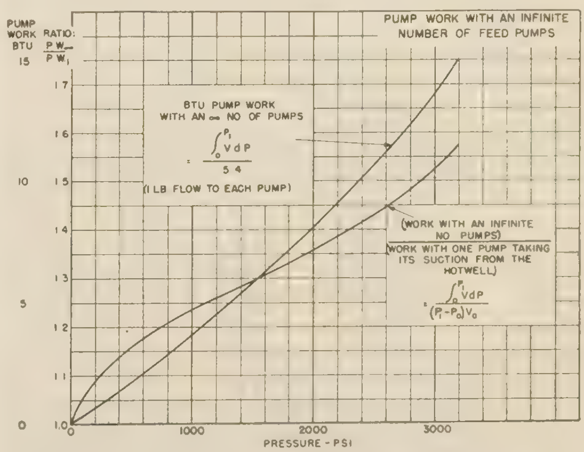

Fig. 10 Pump Work With an Infinite Number of Pumps

[35] A few of the results of this work appear in vol. 2 of "Marine Engineering," Chapter 1, on "Heat Engineering and Thermodynamics," to be published by The Society of Naval Architects and Marine Engineers, New York, N. Y., 1944.

as compared with one hypothetical pump taking water from the hot well. The results, which are approximate only, are shown in Fig. 8 of this discussion, for equal flow to all pumps.

To obtain the true pump work included in the authors' heat rates with an infinite number of pumps, the expression used would have to include a term describing the flow to each infinitesimal pump

$$\left(\frac{\text{P.W.}_\infty}{\text{P.W.}_1}\right)_{(1+w)\,\text{lb}} = \frac{\int_0^{P_1} (1 + w)\ VdP}{(1+w)_t V_0(P_1 - P_0)}$$

where $(1 + w) = f\,(P, \text{etc.})$
and $(1 + w)_t$ = throttle flow (= flow to pump when all drips are cascaded to hot well)

It may be found expedient in many cases to remove the loss due to boiler-feed-pump work so that, until corrections are available, the authors' theoretical heat rates will be of maximum usefulness. These data are apparently all available from the calculations made in obtaining the heat rates. Could the authors therefore undertake to present in their closure the boiler-feed-pump loss in some form such as the ratio just given, or in any other more suitable way?

A. H. Shapiro.[36] The exhaust-temperature correction for cycles supplied with wet steam can be derived directly and simply from the equations for the cycle efficiency, namely

$$\eta_0 = (T_1 - T_0)/T_1$$

and

$$\eta_T = (T_1 - T)/T_1$$

where η_0 and η_T are the thermal efficiencies for the throttle temperature T_1 and for exhaust temperatures T_0 and T, respectively. The ratio of heat rates at T_0 and T is given by the inverse ratio of efficiencies. Thus

$$F = HR_T/HR_0 = \eta_0/\eta_T$$

Elimination of η_T and T_1 from the system of three equations yields the expression

$$F = HR_T/HR_0 = \frac{\eta_0 T_0}{T_0 - T\,(1 - \eta_0)}$$

The authors point out that this correction loses validity when the throttle steam is superheated. Have calculations been carried out for cases involving large amounts of superheat to determine how large an error is incurred through using Fig. 5(*b*) and Table 1, rather than an independent calculation of the heat rate for an exhaust pressure other than 1 in. Hg abs?

The writer has compared the tabulated heat-rates of cycles using dry saturated steam at 3200, 1200, and 300 psia with the heat rates obtained from the expression for the Carnot efficiency, using for the temperature of absolute zero the two values —460 F and —459.69 F. The results are as follows:

Pressure, psia	Heat rates (Btu per kwhr): Selvey and Knowlton	Carnot (—460)	Carnot (—459.69)
3200	6351	6351.0	6349.3
1200	7181	7180.9	7178.7
300	8850	8850.4	8847.3

From the tabulation it appears that the authors have used the value —460 F when calculating the heat rates of cycles with saturated steam at the throttle. For the sake of thermodynamic consistency and of smoothness of the tables, the value —459.69 F would have been preferable, since this value is the one on which the "Steam Tables" of Keenan and Keyes is based and since most of the values in the heat-rate table are based in turn on the "Steam Tables."

The authors state that the intervals selected for the numerical integration of $\int dh/(H - h)$ must be so chosen that each of the intervals has the same value of $\Delta h/(H - h)$. From a mathematical standpoint there is no necessity for this restriction. The intervals Δh may be chosen at random, the sole requirement for application of the trapezoidal rule being that the quantity $1/(H - h)$ be linear with respect to h (to the degree of accuracy specified for the final results) throughout each interval Δh. There is an advantage in employing equal intervals of Δh, as either the trapezoidal rule or Simpson's rule may then conveniently be employed for calculating the total value of the integral between the condenser pressure and the throttle pressure.

In evaluating $\int dh/(H - h)$ the authors take the summation $\sum \frac{\Delta h}{(H - h)_{\text{avg}}}$. There should be employed instead the summation $\sum \left(\frac{1}{H - h}\right)_{\text{avg}} \Delta h.$

The difference between $\frac{1}{(H - h)_{\text{avg}}}$ and $\left(\frac{1}{H - h}\right)_{\text{avg}}$ is small when the two neighboring values of $(H - h)$ are nearly alike, but can appreciably affect the results when the two neighboring values of $(H - h)$ differ by more than a few per cent. In the example given by the authors, the error is of the order of two parts in 10,000 and will probably affect the heat rates by the same order of magnitude. The tabulated heat rates are therefore too small by about 1 or 2 Btu per kwhr.

There appears to be a fundamental inconsistency in the method of the authors for calculating the throttle flow. To begin with, the integral which is the starting point for the calculations, i.e.

$$\int_{h_c}^{h_t} dh/(H - h)$$

is not, as used by the authors, a true integral in the mathematical sense, since the summation of the infinitesimal values of dh is not equal to $h_t - h_c$. In other words, the integral is taken only over those portions of the $(h_t - h_c)$ interval which correspond to feedwater heating, the remaining portion of $(h_t - h_c)$ being accounted for by feed-pump work. The means employed by the authors for overcoming this difficulty seems (to the writer, at least) to involve the confusion of two fundamentally different methods of attack for calculating the throttle flow, namely:

Method 1. Calculations are made for a cycle with a finite number of heaters, the bleed rate to each heater being determined by a heat balance on the individual heater. In principle, the heat rate of the cycle with an infinite number of heaters may be calculated to any desired accuracy by selecting a large finite number of heaters.

Method 2. The differential equation for a single infinitesimal heater is developed and is then integrated to give the total throttle flow. For purposes of numerical integration, values of the integrand may be chosen at finitely separated intervals, but this in no way alters the distinction between the two methods, i.e., that in one case the heat rate for a finite number of heaters is calculated, while for the other case the heat rate for an infinite number of heaters is calculated.

The authors have used Method 1 for calculating the enthalpy change of the feedwater stream across each finite heater but have

[36] Assistant Professor, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. Jun.A.S.M.E.

used an equation of Method 2 which applies to infinitesimal heaters for calculating the amount of steam bled to that heater. This combination of portions of two different procedures is inconsistent and leads to errors of unknown magnitude.

There follow the analyses for the two methods:

Method 1. Fig. 11 of this discussion shows a finite heater. A heat balance yields

$$\Delta w = \Delta(1 + w) = \frac{\Delta h}{H - h}(1 + w)$$

where h is the enthalpy of the feedwater stream entering the heater, Δh is the change in enthalpy of the feedwater stream across the heater, and $(H - h)$ pertains to that heater alone (in contrast to the procedure of the authors, in which $(H - h)$ is the average value for two neighboring heaters). Having tabulated

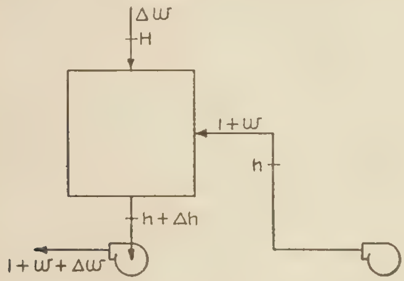

FIG. 11

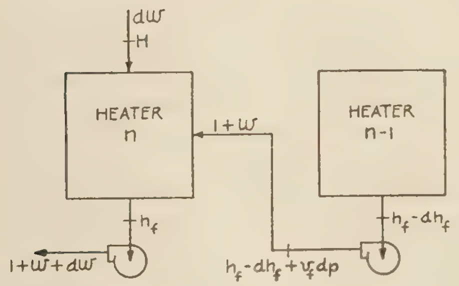

FIG. 12

the values of Δh and of $(H - h)$ for heaters at a number of different pressures, the bleed rate to each heater may be calculated in turn, starting from the low-pressure end. The throttle flow is then the sum of the bleeder flows and the condenser flow.

Method 2. Fig. 12 of this discussion shows two infinitesimal heaters. The symbol h_f refers to the enthalpy of saturated liquid leaving heater n. The enthalpy of saturated liquid leaving heater $n - 1$ is $h_f - dh_f$, and the enthalpy of the feedwater stream entering heater n is $h_f - dh_f + v_f dp$ (note [37]). Equating the enthalpy influx and outflux for heater n, as follows

$$(1 + w + dw)h_f = H\,dw + (h_f - dh_f + v_f dp)(1 + w) \quad\ldots\ldots[7]$$

there is obtained after rearrangement the expression

$$\frac{d(1 + w)}{1 + w} = \frac{dh_f}{H - h_f} - \frac{v_f}{H - h_f}dp \quad\ldots\ldots\ldots\ldots[8]$$

Integration of Equation [8] between condenser and throttle conditions yields for $(1 + w)$ at the throttle

$$\log_e (1 + w)_t = \int_{(h_f)_c}^{(h_f)_t} \frac{dh_f}{H - h_f} - \int_{p_c}^{p_t} \frac{v_f}{H - h_f}dp \quad\ldots[9]$$

The total feed-pump work is obtained from the integral

$$F = \int_{p_c}^{p_t} (1 + w)v_f dp \quad\ldots\ldots\ldots\ldots\ldots[10]$$

in which the value of $(1 + w)$ corresponding to any pressure p is found from an integral like that of Equation [9], except that the limits of Equation [9] are modified to correspond to condenser conditions and the conditions at the pressure p.

Analysis of an infinitesimal stage of the turbine yields for the turbine work the expression

$$dW = (1 + w + dw)(H + dH) - (1 + w)H \quad\ldots\ldots[11]$$

After rearrangement and combination of Equation [11] with Equation [8], the total turbine work may be written

$$W = \int_{\text{condenser}}^{\text{throttle}} (1 + w)\left[dH + \frac{H}{H - h_f}dh_f - \frac{v_f H}{H - h_f}dp\right] \quad\ldots[12]$$

The writer has used Equation [9] for calculating the throttle flow and the heat rate for the illustrative example given by the authors, i.e., throttle conditions of 3200 psia and 1200 F and condenser pressure of 1 in. Hg abs. The value of

$$\int_{(h_f)_c}^{(h_f)_t} \frac{(dh)_f}{H - h_f}$$

was determined numerically with Simpson's rule, using 16, 8, 4, and 2 intervals in $(h_t - h_c)$ to determine the necessary number of intervals. To avoid a separate series of steam-table calculations for the value of $\int_{p_c}^{p_t} \frac{v_f}{H - h_f}$ the expression $v_f/H - h_f$ was plotted against p for the same values of h_f which were previously used for the first integral of Equation [9]. From this plot there were read values of $v_f/H - h_f$ at equal intervals of pressure and the second integral of Equation [9] was evaluated with the aid of Simpson's rule. This semigraphical procedure was found to be amply precise for the second integral, because the value of the latter is very small as compared to the value of the first integral. The following table summarizes the results of the calculations:

$$\int_{p_c}^{p_t} \frac{v_f}{H - h_f}dp = 0.018089$$

Number of intervals	$\int_{(h_f)_c}^{(h_f)_t} \frac{dh_f}{H - h_f}$	$1 + w$	HR Btu per kwhr
16	0.995033	2.65632	6001.2
8	0.995032	2.65632	6001.2
4	0.995833	2.65844	5997.5
2	1.00524	2.68359	5955.3

The authors give for the heat rate 5999 Btu per kwhr. It appears from the tabulation that not more than 8 intervals are required in this case, as compared with 25 or 35 intervals used by the authors in conjunction with the trapezoidal rule.[38] For certain throttle conditions more than 8 intervals may be required, but the number of intervals will always be vastly less than that required with the trapezoidal rule.

The writer has pointed out what appear to him to be several defects in the construction of the table of heat rates. This

[37] From the thermodynamic expression $Tds = dh - vdp$ it follows that the change of enthalpy across an infinitesimal and reversible adiabatic pump is given exactly by the expression vdp, irrespective of volume changes.

[38] Simpson's rule requires that the relation between $1/H - h_f$ and h_f be of the third order or less throughout the chosen intervals of h_f. The trapezoidal rule requires that $1/H - h_f$ be linear with respect to h_f throughout the chosen intervals of h_f. Far fewer intervals may therefore be used with Simpson's rule than with the trapezoidal rule.

scarcely alters the worth of the table, since these defects generally cause errors only in the final digit of the tabulated heat rates. The introduction of these errors is unfortunate in the sense that the table of heat rates does not reflect the accuracy and precision of the thermal data on which the "Steam Tables" of Keenan and Keyes is based.

G. B. Warren.[39] The work that led to this paper was started several years ago, following the quite successful application of a system of quickly determining turbine extracting performance which had been in use by the writer's company for some time.

This first system consisted of curves giving the heat consumption of turbines of 100 per cent efficiency based upon available energy and operating with two, three, four, or five extraction points, also with no pressure or temperature drop, and with the extracted steam mixed with the feedwater being heated. If to the data obtainable from such curves, internal turbine-efficiency figures similar to those given in the Knowlton and Warren paper[40] of 1941 are applied, corrections made for leaving loss and for the usually small departure of the actual from the ideal extraction cycle, a close approximation to the real turbine performance can be arrived at. If the feedwater temperature is known, and it can be chosen over quite a range without affecting greatly the foregoing results, the throttle and condenser flows can be easily obtained.

During the early study of this problem, it apparently appeared as though the original form of presenting these data should be modified, and it was put upon the basis of an infinite number of heaters heating the feedwater to the saturation temperature of the boiler, going back to a conception introduced in 1923, by Ernest L. Robinson.[41]

It is to be hoped that the present authors will have an opportunity to prepare the remaining portion of this work which will permit making full practical use of the data.

In the meantime, while the work from which this present paper was prepared was progressing rather slowly, J. K. Salisbury worked up his very excellent system of determining turbine-extraction performance,[34] based upon actual nonextraction heat rates. To some extent this followed an earlier method worked out by and used by the General Electric Turbine Division. This system also facilitated determination of over-all turbine-extraction performance from the nonextraction-turbine efficiencies obtainable from the Knowlton and Warren paper,[40] or from any other convenient source.

Power-plant engineers have at their command therefore published information from which complete and accurate studies of the relative heat consumption of any steam-turbine plant can be calculated. Knowledge of boiler performance and miscellaneous operating losses permits a close approximation of the over-all fuel consumption.

The authors are to be congratulated for carrying out a very long and laborious undertaking and can have the satisfaction of having made a lasting and fundamental contribution that should stand as long as the present steam tables maintain their acceptance and the many conditions of operation are not exceeded.

Author's Closure

The authors are gratified that so much worth-while discussion of their paper was forthcoming. The interest manifest in theoretical regenerative-steam-cycle heat rates has confirmed their opinion concerning the importance of this subject to steam-power engineers, and the need for basic data in a form convenient for applying to the solution of such problems. The discussers have brought to attention several points which can well be explained or amplified with a view to making the data more understandable and useful.

Professor Baumeister rightly states that the value of the paper rests in its utility to engineers engaged in designing and operating power plants and wishes to know whether it is proposed that the answers to several questions of detailed apparatus arrangements will be forthcoming. These answers are obviously not contained in the present paper, so that the question really is as to what is proposed for the future second part of the work. This second part has not been crystallized, but the general philosophy now existing regarding what should be done is about as follows:

The effects on performance of losses in any of the apparatus required in steam plants would be pointed out in some detail with the view to showing from past experience what should be reasonably attainable.

The fact that engine efficiencies and other factors change with steam conditions and with advances in the art of manufacturing turbines, as pointed out by Professor Baumeister, would seem to justify the development of the heat-rate tables on the basis of theoretical conditions as was done in the present paper. Thus the tabular values need not be changed (unless the steam tables are changed), and it is believed that correction factors can be worked out for different engine efficiencies so that the plant designer may use the engine efficiency which is applicable at the time of making his calculations.

It is not to be expected that such factors as can be evolved later will be sufficiently all inclusive to furnish an answer to every question; but in general it is hoped to give the plant designer enough information to permit him to assign an over-all efficiency to his plant, based upon actual or calculated heat consumption, and to show why this over-all efficiency is what it is assumed to be. When the major elements of a proposed cycle have been decided, it may then be necessary to make detailed calculations to evaluate the relative merits of minor refinements to the cycle such as changing the location in the cycle of an evaporator or of a heater-drain pump.

The data already presented also should be useful in making comparisons between different cycles, as Professor Baumeister suggests. These comparisons in practical cases require consideration of the effect on performance of many factors of an involved nature, although the net differences in the final answers may be small. A simple comparison of the theoretical heat rates will show the theoretical differences, which can be compared with the practical ones, whereby a better understanding of the reasons for discrepancies may be had.

Mr. Brooks raises a question as to the assumption made regarding the temperature to which feedwater is heated by superheated steam. This is stated in paragraph (2) of the paper under "Description." The curves in Fig. 6 are drawn from the figures in Table 1, simply to show the trend of decreasing heat rate with increasing temperature. With reference to the use of superheated steam in heaters, Mr. Brooks doubts that "it is feasible to get higher than the saturation temperature of the steam in the heater. . ." Actually several heater manufacturers are guaranteeing performance on the basis of heating the feedwater to a temperature several degrees above the saturated-steam temperature in the heater if the heater is supplied with superheated steam.

Commander Dusinberre, Mr. Markson, and Dr. Robinson apparently have a common idea, viz., that information concerning weight flows, as well as "parts" of complete cycles, should be made available. Commander Dusinberre has proceeded to make some sample figures and tabulations illustrating how this idea

[39] Turbine Engineering Division, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[40] "Relative 'Engine Efficiencies' Realizable From Large Modern Steam-Turbine-Generator Units," by G. B. Warren and P. H. Knowlton, Trans. A.S.M.E., vol. 63, 1941, pp. 125–135.

[41] "The Margins of Possible Improvement in the Central Station Plant," by Ernest L. Robinson, Trans. A.S.M.E., vol. 45, 1923, p. 644.

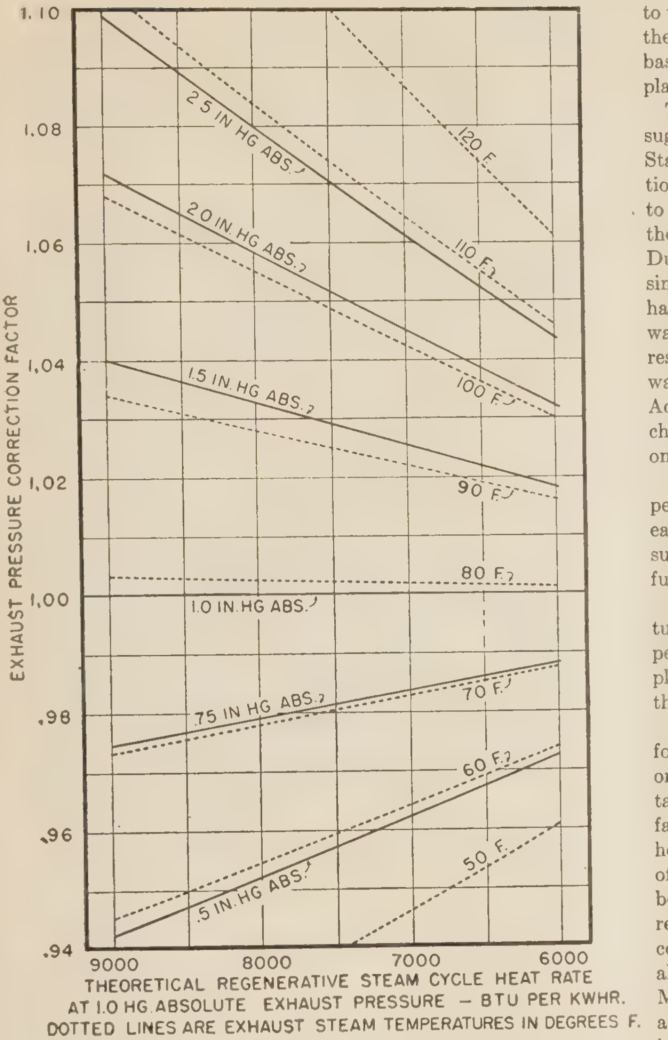

Fig. 5(*b*) Revised Chart of Exhaust-Pressure Correction Factor

(In the theoretical cycle, exhaust steam temperature is the same as condensing water supply temperature.)

might be carried out. The authors believe that such information, if completely worked out, would be useful. In fact, it may be the only good way of handling the reheat cycle but, at present, they are unable to do more than commend this suggestion to anyone having time to work on it.

In his statement concerning power-plant performance of River Rouge No. 1 Power Plant of the Ford Motor Company, Mr. Dulmage has made a welcome contribution to the store of information which will be needed in preparing the second paper. The authors appreciate this early co-operation. For further information about this plant, reference may be made to an article by W. W. Dulmage and C. J. Walker.[42]

Messrs. Engle and Benjamin both point out the possible value of the paper and its proposed sequel as a means of relating the actual performance of a plant to what can reasonably be expected of its cycle. Such uses are precisely along the lines of the authors' own ideas. There also may be considerable merit to the idea of rating the actual performance of a given plant upon the basis of what fairly can be expected of it, rather than upon the basis of comparison with a more modern or more complex plant.

The authors feel that Mr. Hardie has complimented them in suggesting that the heat-rate tables be adopted as an American Standard basis for appraising plant performance. His suggestion that the condensing-water temperature would be preferable to the turbine exhaust pressure as the lower boundary of the theoretical cycle is supported by the discussion of Commander Dusinberre. This suggestion appeals to the authors as well, since it serves to emphasize that in the theoretical cycle the exhaust steam would condense at the temperature of the cooling-water supply whereas in an actual cycle thermodynamic losses result from the necessity for a temperature rise in the cooling water, and a terminal temperature difference in the condenser. Accordingly, constant-temperature lines have been added to the chart in Fig. 5(*b*), which originally had exhaust pressure lines only. The revised Fig. 5(*b*) appears with this closure.

The authors had not considered the problem of appraising the performance of a mixed-pressure plant. It would seem that each section of such a plant could be appraised separately and a suitable basis of averaging arrived at. The suggested basis of fuel input may well be the most satisfactory for this purpose.

Mr. Hardie mentions the pressure loss from boiler drum to turbine throttle, which of course is not present in a theoretically perfect cycle. This is a practical loss, and presumably the actual plant should be charged with this loss, part of which occurs in the boiler and part in the piping.

Mr. Ireland has requested a basis for using a short-cut formula for computing heat rates of partial regeneration cycles on the assumption that, with the full regeneration heat-rate tables available, it would be possible to derive suitable correction factors for using a short-cut formula. In actual practice, the heater spacing is not ideal in all cases, only a small finite number of heaters are employed, and the same final feed temperature may be achieved using any number of feedwater heaters. For these reasons it does not appear that any simple set of corrections could be developed which would yield satisfactory results under all conditions when applied to the short-cut formula. If Mr. Ireland is thinking in terms of an actual turbine, the authors feel that the best way to accomplish what they believe he wants, the method developed by J. Kenneth Salisbury, referred to by other discussers, should be used, or the older methods of the large-turbine manufacturers.

On the other hand, if Mr. Ireland is thinking in terms of theoretical performance, with only partial regeneration, it is suggested that the results to be obtained from the short-cut formula with corrections could be derived rather easily by superimposing a Rankine cycle on a theoretical regenerative cycle having its initial pressure equal to that of the highest pressure bleed point. The bleed-point pressure would be the saturation pressure corresponding to the final feedwater temperature, and the exhaust pressure of the Rankine unit would have that same value. This suggestion is an application of the thoughts expressed by Commander Dusinberre and Mr. Markson. The heat rate of the regenerative portion of the cycle can be taken from the present paper while the heat rate of the Rankine portion of the cycle may be determined from published tables of theoretical steam rates.[43] The effect of partial regeneration would be covered in the correction factors of the proposed companion paper.

Mr. Meyer suggests that the calculations could be made by

[42] "Performance Test at Ford Motor Company," by W. W. Dulmage and C. J. Walker, *Power*, vol. 85, July, 1941, pp. 72–75.

[43] "Theoretical Steam Rate Tables," by J. H. Keenan and F. G. Keyes, published by The American Society of Mechanical Engineers, New York, N. Y., 1938.

formulas derived from the concept: "In any cycle in which heat (dQ) is added at variable temperature T (deg abs) and rejected at constant temperature T_e (deg abs) the maximum work obtainable (W_m) is given by" etc., from which he proceeds to derive the formula. This formula is the same as that given in the paper and labeled "inaccurate." The error of this formula is shown in Fig. 3 and comes about because the formula is only correct for a "reversible" cycle, whereas the cycle used by the authors contains an irreversible process whenever the throttle steam is superheated, this irreversible process being the transfer of heat from superheated steam to feedwater, which transfer involves a definite degradation of heat.

Professor Shapiro derives an expression for exhaust-pressure correction which is the equivalent of the one given in the paper. These expressions are exact, regardless of the initial condition of the steam, as long as the exhaust is in the wet region. Therefore there is no error in using Fig. 5(b) and Table 1. The authors believe Professor Shapiro must have misread the text since they did not intend to cast any doubt on the usefulness of the exhaust-pressure correction factors. Professor Shapiro's full discussion of the methods of calculation of the theoretical heat rates constitutes a valuable addition to the paper. The authors agree that his "Method 2" is an accurate means of performing the necessary integration, and that, even for the highest pressure cycles, a relatively small number of intervals is satisfactory. A similar method, although with not quite as precise a determination of the feed-pump work, was used in some of the calculations made in producing the heat-rate table. Unfortunately, a few computation errors were made, so that the true value of this method was obscured. Also, small errors in determining feed-pump work made the computations slightly inaccurate at the high pressures, which difficulty seems to have been overcome by Professor Shapiro. Regarding his statement as to the accuracy of the heat-rate table, we believe the figures to be correct within about ± 3 Btu per kwhr.

Dr. Robinson's remarks concerning the desirability of having weight-flow data available apparently reflect the same viewpoint as is expressed by Commander Dusinberre. The authors do not question the potential value of such information, if it could be prepared. Quite possibly it might be the most advantageous way of handling the reheat cycle for which Commander Dusinberre wants data. If such data were prepared it also would be desirable, as Dr. Robinson points out, to have figures based upon various practical turbine efficiencies.

Mr. Salisbury's remarks, comparing the present paper with his own excellent presentation, point out the differences as well as the similarities in the two approaches to this question of theoretical and practical steam-plant performance. The authors have hopes, with Mr. Salisbury's permission, of using some of his results, in preparing the proposed sequel to the present paper. It is thought that practically all of Mr. Salisbury's work is directly applicable except that, in general, his results must be cast into a somewhat different form. This is because he has dealt with "percentage gains," built up upon the nonextraction cycle, whereas the authors intended to deal with "percentage losses" from the theoretical regenerative cycle.

The authors do not agree with Mr. Salisbury that "it may be found expedient to remove the loss due to boiler-feed-pump work." In their opinion the feed-pump work is not a loss at all in the theoretical cycle, since the pump is a necessary part of the cycle. There are a number of practical ways in which the feed pump has been arranged in practical cycles, varying from a single pump to a separate pump for each of a finite number of feed heaters. It will eventually be necessary to work out the differences between the theoretical cycle infinite-number-of-steps pump and the practical one-or-more-step pump.

The authors appreciate the comments made by Mr. Warren, which present a concise picture of the way in which it is felt these theoretical heat rates will be of use. They also appreciate his constructive interest and consistent support accorded to this project.

Mechanical Features of the Glenville Impulse Turbine

By ARNOLD PFAU,[1] MILWAUKEE, WIS.

At the Glenville Plant of the Nantahala Power and Light Company near Franklin, N. C., is the only large impulse-wheel turbine unit east of the Rockies. Its performance has established a water-wheel efficiency record for the United States of practically 89 per cent. This paper is devoted to descriptive details of the unit, its operations, and performance.

The Glenville Plant of the Nantahala Power and Light Company is located on the West Fork of the Tuckasegee River about 13 miles east of Franklin, N. C. It has the distinction of being the only large-capacity impulse turbine east of the Rocky Mountains. With a gross head of 1207 ft, a net head at maximum discharge of 1188 ft, and a rated capacity of 30,000 hp at 1150 ft, 27,000 kva, 21,600 kw, its capacity exceeds by far that of the few small impulse turbines in the East.

The Tuckasegee River at Glenville Dam has a drainage area of 39.4 sq miles and an average annual computed flow of 135 cfs. The dam with a height of 150 ft has a net storage volume of 68,000 acre-ft. Since the turbine uses about 275 cfs at full load, this unit will operate at approximately 50 per cent annual load factor, drawing out the available storage and generating power to suit the requirements of the downriver plants most advantageously.

General Operating Details

The Glenville unit is a horizontal-shaft single-jet double-overhung impulse wheel with shaft supported in two bearings and the generator rotor located between them. The turbine is rated 30,000 hp, 1150 ft net head, 257 rpm. One shaft end is extended for coupling to exciter, pilot exciter, and to the independent generator supplying current to the motors driving the flyballs of the governors.

Being of the double-overhung type, the two water wheels, one on each shaft end, constitute two independent hydraulic prime movers, each being provided with its own governor and gate valve, so that the water on one side can be shut off entirely without interfering with the operation of the other. When small outputs of less than 50 per cent are necessary or desirable, a better wheel efficiency can be maintained. Also, the individual control of each water wheel by its own governor assures a double safety, maintaining a control of one half of the output of the unit in case the governor of the other half should refuse to act properly.

This double control, two governors for one single generator, was successfully introduced by the company with which the author is connected and has since become a generally accepted design here and even abroad, not only for double-overhung impulse wheels but also for double-overhung Francis-turbine units, the first of this latter type having been furnished by the author's company in 1913, for the 18,000-hp unit at the Halsey Plant of the Pacific Gas & Electric Company of California.

The Glenville unit is designed for a static head of 1215 ft and rated two times 15,000 bhp under 1150 ft net head at 257 rpm. The generator is rated 27,000 kva maximum, 6600 v, 60 cycles, three phase. It is designed for the full runaway speed which the rotating parts may attain, with both jets wide open under 1150 ft net head and no load on the generator. Speed-and-pressure control of the unit is maintained by the individual governing mechanism of each water-wheel side.

To avoid dangerous pressure rises and surges in the pressure tunnel, which is several miles long, with no surge tank, the change of velocity of water under fluctuating load must be very gradual. To obtain quick response to load changes, the output of the unit must be changed quickly by a corresponding change of water quantity impinging upon the buckets. The most perfect method of quick control would be simply to control the impingement of the jets and to hold the flow-controlling needles stationary, i.e., water-wasting regulation. This is prohibitive when water is to be used economically. Here provision must be made to adjust the flow according to the load demand.

Since the Glenville plant does not permit of quick changes of flow in the tunnel and pressure pipe, it is realized that load increases can be taken care of only if so gradual that no serious pressure drops will be experienced. The two needles controlling the flow are therefore set to stop the flow at an average rate of 120 sec, slowing down near closed position to a rate of about 200 sec, and to accelerate to full flow in about 100 sec. Such rate of closing movement would involve a full runaway of the unit on full sudden load rejection such as is the case when the unit becomes detached from the power system. To hold the momentary speed rise within tolerable limits, the quantity of water directed upon the buckets must be changed more rapidly.

In the older art this was accomplished by moving the needle rapidly and by opening a relief outlet as synchronously as possible to discharge the amount shut off by the needle, and thereafter to close the relief outlet at a rate to prevent undesirable secondary pressure rises in the pipe line; in other words, a "switch-over" of flow out of the needle nozzle to relief outlet. This is ideal only if the two discharges are strictly inversed at all times so that the sum of the two discharges equals the former total discharge, i.e., that the velocity of water in the pipe line is not changed.

A practically perfect method, first introduced by the author's company in connection with the 60,000-hp impulse-wheel unit at the Big Creek 2-A Plant of the Southern California Edison Company, Los Angeles, Calif., is to provide means for diverting quickly the jet from the buckets and to let the flow-controlling needle follow the movement at a rate "independent" of the rate of deflection, but in strict accordance as dictated by the pipe-line characteristics.

The various processes are illustrated in Fig. 1 (*a*) for both relief outlet and jet deflection.

Relief Outlet. In Fig. 1 (*a*) at ordinate A-$A'T$ (governor time) $= 0$, the flow is 100 per cent (full load of unit). The load is suddenly dropped from A' to $A = 0$. The speed rises, but the flow-controlling needle does not start to move until time B is

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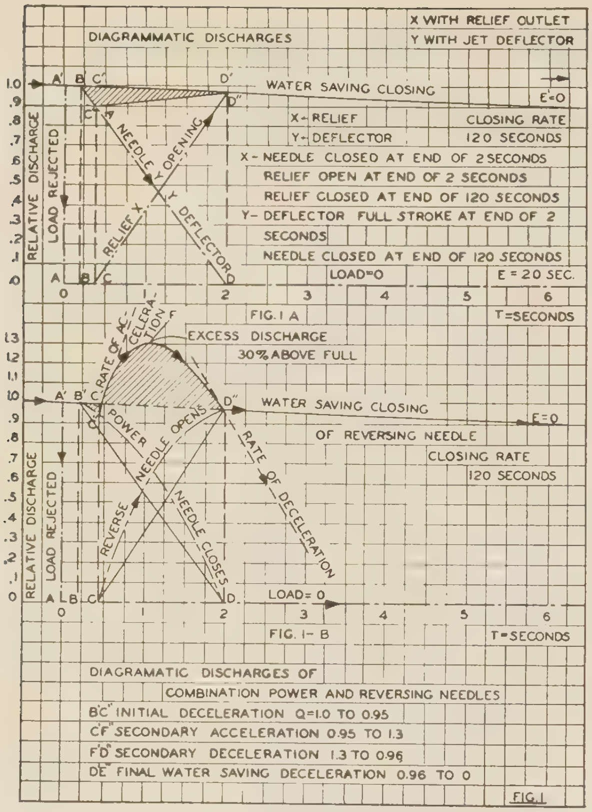

Fig. 1 Diagrammatic Discharges of Combination Power and Reversing Needles

$B'C''$—Initial deceleration: $Q = 1.0$ to 0.95
$C''F$—Secondary acceleration: $Q = 0.95$ to 1.3
FD''—Secondary deceleration: $Q = 1.3$ to 0.96
$D''E'$—Final water-saving decelerations: $Q = 0.96$ to 0

reached. The flow through the needle is reduced along line B'-D and the flow upon the buckets is stopped in 2 sec from the beginning of load rejection. The movement of the needle acts upon the relief outlet which begins to open at time C, and its water-saving mechanism immediately starts its relative closing movement at a rate of 120 sec (from C' to E'). The delay BC, during which the relief outlet remains closed, causes a reduction of flow from B' to C'' at the full rate of closure of the needle, thereby producing a material initial pressure wave in the pipe line.

If the relief outlet has a capacity of 100 per cent of that of the needle, it opens the relief outlet wide at the time D, when the needle is closed. However, the relative water-saving movement reduces the flow to D'' instead of D'. If the discharge rates of the needle and relief outlet were linear, the resultant total discharge (being the sum of the ordinates of needle- and of relief-outlet discharges, respectively) would be as indicated by curve C''-D'', and the area $B'D'D''C''B'$ represents the change of velocity in the pipe line. It is obvious that any deviation from the straight lines B'-D and C-D'' causes even more serious changes of water velocity, as illustrated in Fig. 1 (*b*).

Jet Deflector. In Fig. 1 (*b*), the deflector is directly actuated by the governor. Therefore no additional initial delay BC is experienced. The output flow is reduced along line B'-D, and the needle controlling the flow in the pipe line is independently reduced to zero from B' to E'. The undesirable feature of change of resultant flow represented by former area $B'C''$ D'' is reduced to $B'D''$. The diagram at once discloses that here the deflector movement from B' to D (taking the place of the former needle movement) can be made as quickly as is mechanically desirable.

Fig. 1 (*b*) represents the case of two reversing needles of approximately parabolic discharge characteristics, resulting in an initial deceleration $B'C''$ until the reversing needle begins to open, and changing suddenly into an acceleration $C''F$, diminishing to zero at F, where the total flow is a maximum, and changing into a deceleration from F to D'', reaching its maximum at D'', when the power needle is closed and the reversing needle wide open, and followed by the gradual deceleration $D''E'$, caused by the water-saving movement of the reversing needle to closed position at the end E' of 120 sec. Serious accidents to pipe lines and loss of life are on record caused by a faulty correlation in movements and discharge characteristics of such reversing needles.

Mechanical Features of Glenville Unit

Main Shaft. The main shaft is a hollow-bored steel forging with integrally forged flanges on each end for bolting the water-wheel runners to it. It is bossed up in the center so that the generator rotor can be brought into position over one of the flanges. It rests in two bearings of 22 in. diam, 18 ft apart. One end is arranged for flexible coupling to the direct-connected exciter.

Bearings. The total rotating weight (water wheels and generator rotor) of about 135 tons is carried in two ring-oiling pedestal-type bearings, each of 1450 sq in. bearing surface, in split, babbitt-lined shells with ball-and-socket seat, which permits of adjustment of the bearing surface in accordance with the deflection of the overhung portion of the shaft carrying the weight of wheel runner and the resultant impact force produced by the jet, which varies as the output of that side of unit varies. The two bearing shells are water-cooled, and lubrication is assured by five oiling rings which pick up the oil from the basin formed in the lower bearing housing and discharge it over the top of the shaft. Thus the normal operation is not in need of outside lubrication under pressure, but is self-contained. However, to facilitate starting the unit, when the contacting bearing surfaces may have become dry, provision is made to use oil under pressure from the oil-pressure system of the governors.

Temperature relays are provided which cause the governors to shut the unit down when the temperature becomes excessive. Excellent results have been obtained with this design of bearing, the largest being those at the Big Creek 2-A Plant with a shaft diameter of 32 in. operating at 250 and 300 rpm, delivering 50- and 60-cycle current at times, depending upon the network served.

Wheel Disks and Buckets. The forged-steel wheel disk is machined all over and is held to the forged-shaft flange by central-recess and ream-fit steel bolts with secured nuts. Twenty-one buckets of cast steel are held to the disk each by three taper ream-fit bolts with secured nuts on either end. From a point of view of mechanical strength, the design is liberal, involving only moderately high stresses of material even under maximum centrifugal force due to runaway speed, as also under conditions of rapid changes of impact force. It is not generally realized fully to what forces such a bucket is exposed, both as to amount and as to duration. At 257 rpm, a bucket passes through the jet 4.3 times per sec. When the full jet impinges, such as is the case here where the pitch is larger than the total length of intersection of the jet on the periphery of impingement, the impact force is 63,000 lb, beginning with zero value when the bucket lip enters the jet, reaching above maximum as long as the full jet impinges on the same bucket and reaching zero value again when the last particle of jet discharges from the bucket. This process takes

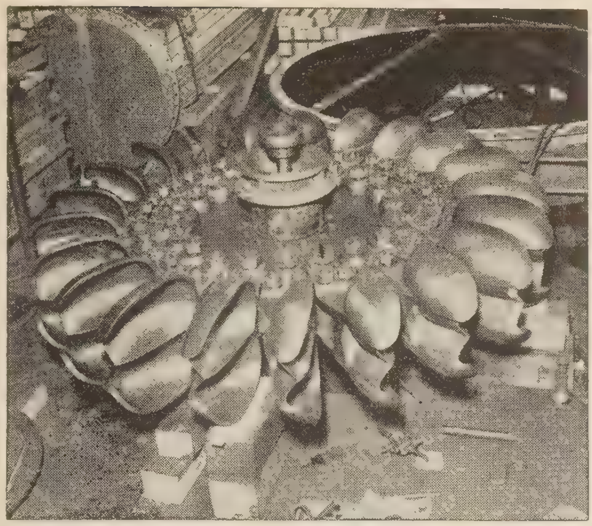

Fig. 2 Shop-Assembled Disk and Buckets for 10-In-Diam Jets

place over about $^1/_4$ of the periphery of the wheel, therefore, in

$$\frac{1}{4 \times 4.3} \text{ or about } \frac{1}{17} \text{ sec.}$$

This intense hammer blow (some practical engineers call it "a swift kick in the pants") must be resisted by the bolts holding the bucket to the disk, in addition to the steady centrifugal force of about 110,000 lb at normal speed, or about 370,000 lb at full runaway speed.

This explains the cause of serious accidents experienced early this century when buckets exploded, due to crystallization of bolts and breakage of bucket castings, and due to weakening on account of pitting because of faulty inlet and discharge angles of the bucket bowl.

Fig. 2 shows the shop-assembled bucket wheel with all bolt holes properly reamed and bolts and buckets fitted and assembled on the wheel disk. The weight of each rotating element is about 37,500 lb.

Nozzle Pipes. In the nozzle pipe, the energy of the flowing water (partly pressure and partly velocity), due to the total available net head of 1188 ft at the entrance to the nozzle pipe is transformed into kinetic energy which is contained in the jet after leaving the orifice. This process of transformation can be

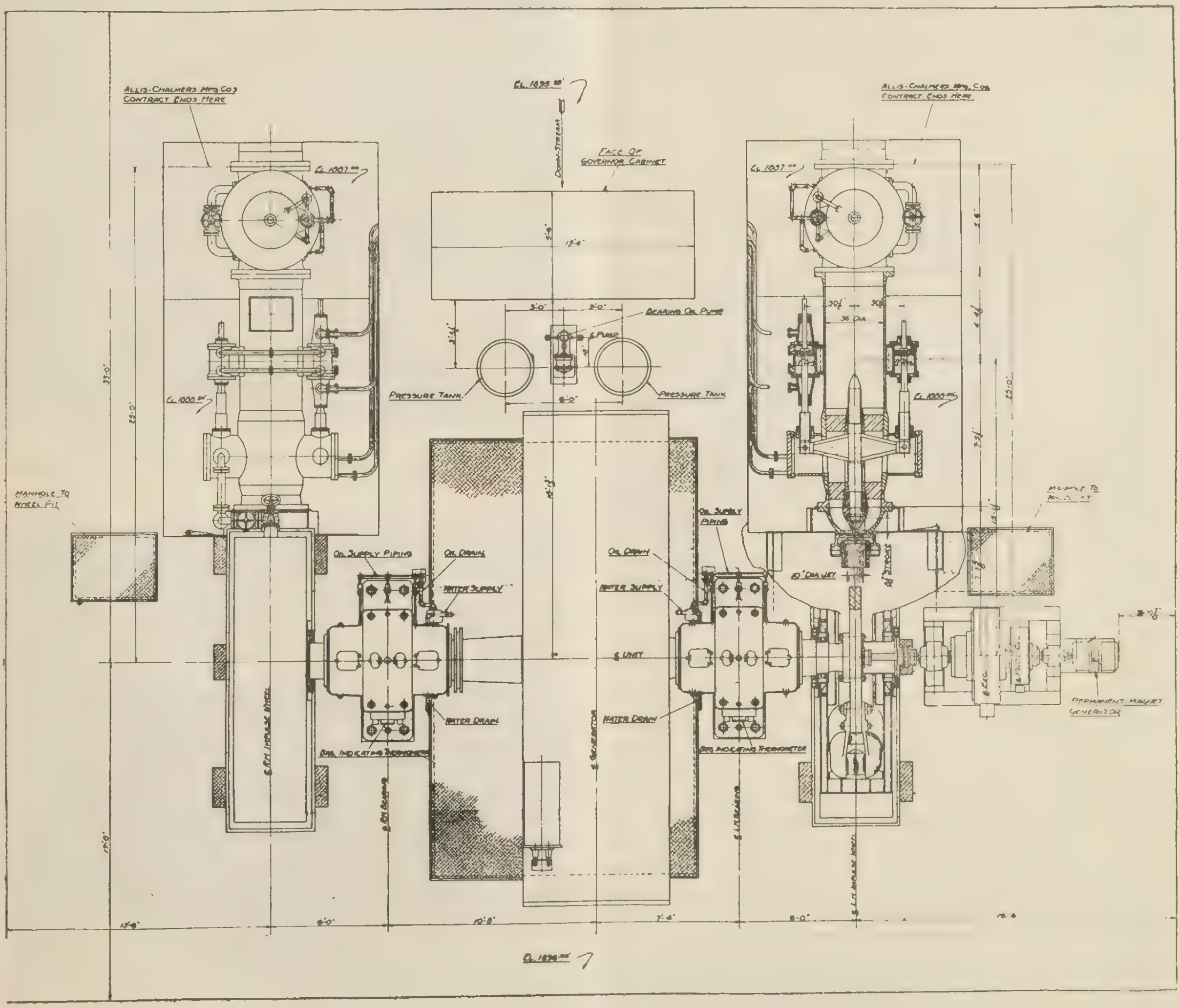

Fig. 3 Plan of Hydraulic Impulse-Turbine Unit at Glenville Plant

accomplished more or less efficiently, depending upon the design and care of workmanship.

Let C_n be the velocity due to the net head H_n at the entrance to the nozzle pipe, and V the velocity of the jet. Thus the jet efficiency e is

$$e = 1 - \frac{M \times C_n^2}{2} - \frac{M \times V^2}{2} : \frac{M \times C_n^2}{2} \text{ or } \frac{(V)^2}{(C_n)}$$

If the jet velocity were equal to the available net velocity C_n, the efficiency of the jet would be 100 per cent. Naturally, the bucket, transforming the kinetic energy $\frac{M \times V^2}{2}$ in the jet into torque force, producing output, cannot do more than what it receives. This justifies the designer to endeavor to produce the best possible nozzle design. It at once cautions against using multiple-bend piping, such as is involved with a multiple-jet arrangement or crowded settings of the water wheel. This applies equally well to the Francis and propeller types.

The Glenville nozzles are of the patented design first used in 1927, in connection with the 40,000-hp water wheel at the San Francisquito No. 1 Plant of the Bureau of Power and Light, Los Angeles, Calif., and from which to date still issue the largest diameter of jets (14 in.) in the world.

Fig. 3 shows a plan view of the complete unit.

The nozzle pipe is made of welded steel with an inlet diameter of 36 in. for flanged connection to the gate valve. It is straight and slightly tapered, containing a four-arm spider forming the support of the needle stem carrying the steel needle tip. This needle tip is made of two removable parts to facilitate and economize repairs. The stem is provided with a crossarm, the two ends of which project into pockets on either side of the nozzle pipe where they are pivoted to plungers projecting through packing boxes to the outside where they are coupled to the piston rods of the servomotor jointly actuating the needle. The diameters of the plungers are so dimensioned that they partly offset the force tending to hold the needle tip against the orifice from which the jet issues. The downstream end of the nozzle pipe carries the cast-steel nozzle tip with renewable steel throat ring which, when the needle tip seats against it, closes the orifice tightly.

Fig. 4 shows the shop-assembled nozzles with needles, and mounted on them are:

The Jet Deflectors. These are disposed between the nozzle orifice and the rotating buckets, with a minimum space required to maintain compactness of the full jet impinging upon the buckets. The jet deflector surrounds the jet without interference when in neutral position to assure highest output efficiency. The portion surrounding the jet is lined with a renewable steel sleeve. The deflector is hinged and extended at the lower end for pivoted connections into the nozzle-pipe pit where it is coupled to its oil-pressure servomotor.

The orifice part of the nozzle pipe, consisting of the throat ring and needle tip, to a great extent affects the condition (or efficiency) of the jet, and since it is exposed to the high velocity of water issuing, it is essential that it be kept in good condition by frequent examination. For this purpose the supports for the hinges of the stream deflector can be readily moved out of the way for quick access to the orifice.

Upper and Lower Wheel Housing. Fig. 5 shows one of the housings shop-assembled. It is made of welded plate steel, suitably reinforced to minimize breathing. The lower part is completely grouted into the concrete foundation and is machined on top to form the base to which the upper wheel housing is bolted. The rear face is also machined for supporting and surrounding the nozzle pipe. A door is provided for access from the nozzle-

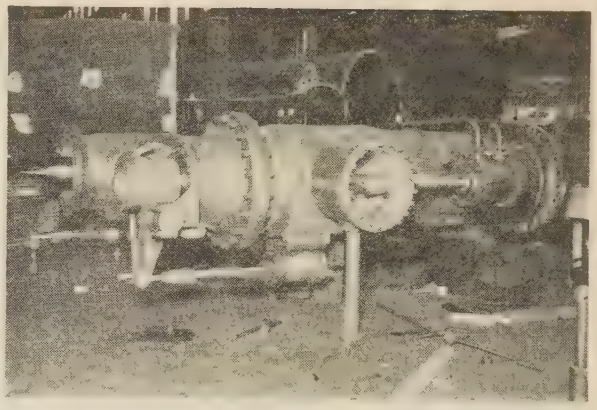

Fig. 4 Nozzle With Jet Deflector and Needle Operating Cylinders Set Up in Shop

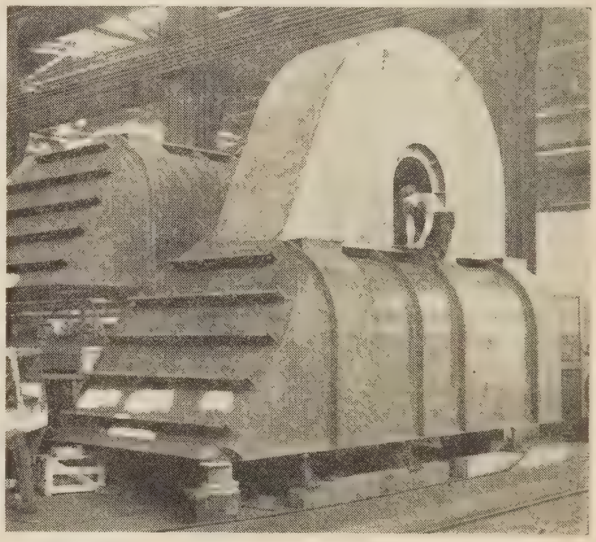

Fig. 5 Upper and Lower Welded-Steel-Plate Impulse-Wheel Housings

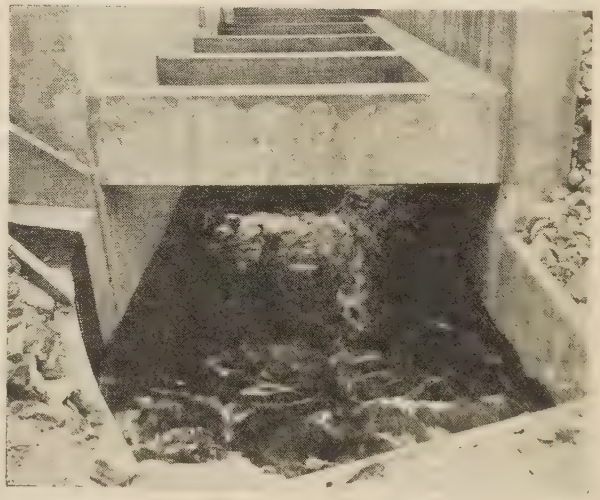

Fig. 6 View of Stilling Pool

pipe pit to the front of the wheel pit. The downsteam face is inclined and heavily reinforced to be capable of receiving the full impact force of the jet when, due to runaway, the major portion of the jet passes forward without impinging upon the buckets. Proper baffles are provided in order to deflect the discharging water, rather than to throw it back against the buckets, or the orifice.

The upper housing is double-walled for stiffness and for the purpose of ventilation.

Where the main shaft projects through the housing, proper baffles are provided to prevent splash water from escaping along the shaft.

Wheel Pits. These are lined with steel on both sides, the bottom and the ceiling of the outlet.

Inspection shafts for access to the wheel pits are provided on each outer upstream end of the unit. They are covered by floor plates securely bolted to frames to avoid rattling caused by the heavy air-pressure fluctuations in the wheel pit when the jet becomes deflected on sudden load rejections.

Energy Translator. When the jet is fully deflected from the buckets by the sleeve in the jet deflector, it presses against the upper-cylinder portion of the sleeve and thus begins to spread, whereby it loses part of its impact force. This would still be so high as to cause destruction to the tailrace. Fig. 6 is a view of the stilling pool showing the water discharging when the jets are fully deflected. It was designed and built by the engineers of the Aluminum Company of America and is giving very satisfactory service. A combination of splitters is built into the end of the apron and these are assisted by the pool of water in the stilling pool itself. The splitters are wedge-shaped with the point of the wedge facing the unit, three to each water wheel, the center splitter being in line with the center line of the water wheel. This splits the deflected jet into seven parts. That part of the jet impinging against the wedgelike splitter is elevated, while the jet between the splitters and on either side of the end splitters goes out horizontally. This scheme has proved very satisfactory on full-load rejections, no water ever being spilled beyond the confining walls of the stilling pool when the full-load quantity of 270 cfs is discharged under 1188 ft net head with a velocity of about 264 fps.

Hydraulic Brake. The liberal flywheel effect of rotating parts of the two water-wheel runners and the generator rotor would keep the unit revolving for about 40 min after the water is shut off. In case of mechanical accident, this delay in coming to a stop may cause more damage. Provision is made to reduce this time materially by applying a water jet onto the buckets opposite to the direction of rotation. The brake jet is controllable by a hand-operated needle and a gate valve connected to the upstream side of the main gate valve, so that the brake can be applied even when the main gate valve has been closed.

Gate Valves. These are illustrated in Fig. 7, under actual water-pressure test in the shops with a pressure of 550 psi applied to the upstream side of valve plug. The valve, built of cast steel throughout, is of the single-seat parallel-face follower-ring type, as successfully used since 1913, for the eight 24-in. valves at Big Creek No. 1 and No. 2 Plants under heads up to 2350 ft. These are designed to be closed against the full flow of water under the maximum head. The filler ring is an extension, integral with and below the valve plug, which, when the valve is in the wide-open position, fits into the gap otherwise left in the valve housing. It thus protects the valve seat against scouring by sand, etc., passing through the valve during operation, and it serves as a guide for the plug to prevent chattering of the plug during closing or opening movement.

The valve plug is raised and lowered by a hydraulic piston and cylinder, with control valve admitting penstock pressure to the top and bottom, respectively, of the piston for closing and opening the valve.

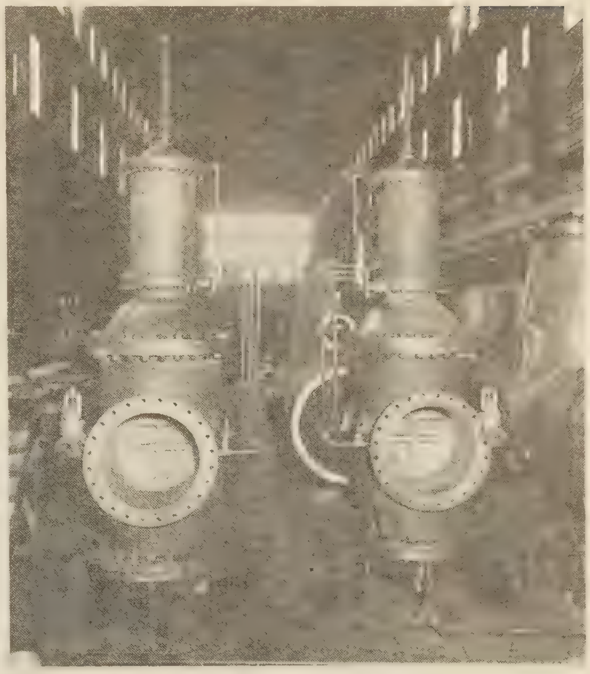

Fig. 7 Two 36-In. Follower-Type Gate Valves for Glenville Unit Undergoing Shop Tests

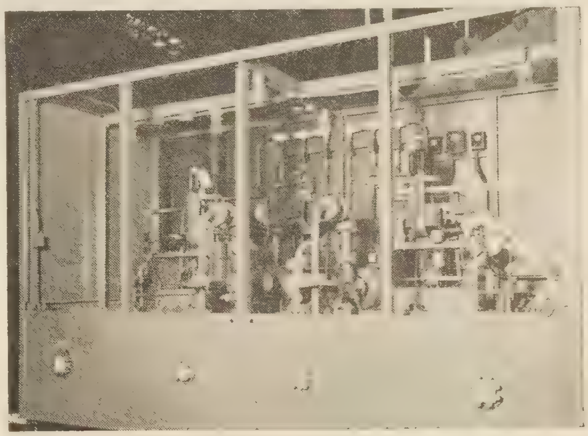

Fig. 8 Rear View of Cabinet Governor for One Double-Overhung Impulse Wheel

Governors. As previously explained and diagrammatically illustrated in Fig. 1, speed and pressure regulation is accomplished by a combination of quick-acting water-wasting jet deflectors. Subsequent gradual movement of the flow-controlling needle is at a rate to assure protection of the pressure pipe line against undesirable pressure variations. Each water-wheel side has its own governor, with two servomotors attached to its nozzle pipe for movement of the needle, and one servomotor located below the nozzle pipe in the nozzle pit for the purpose of actuating the jet deflector.

The actuators of the governors are housed together in a cabinet near the upstream wall of the powerhouse, where the

two oil pump sets, pressure and receiver tanks are also located. Fig. 8 shows the rear of the cabinet opened up for a full view of the equipment and wiring circuits behind the front wall of the cabinet.

The principal elements of a modern governor are:

The speed-responsive part, or flyballs.

The control valve actuating the servomotor controlling the output producing flow of water.

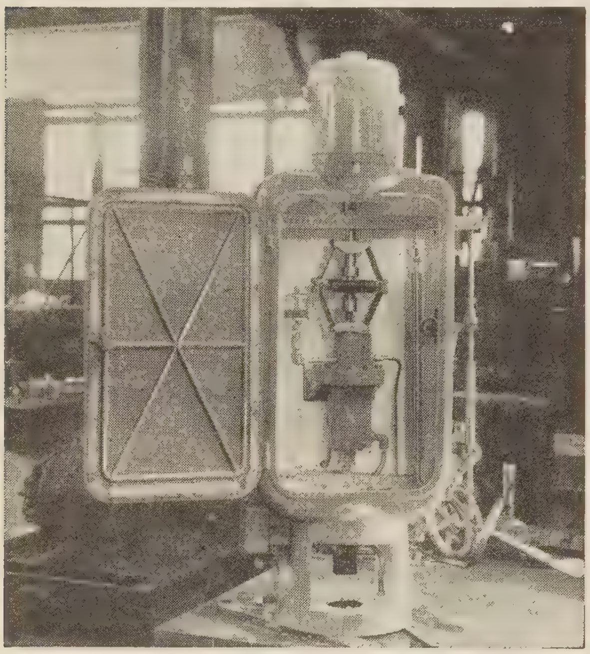

Fig. 9 Supersensitive Flyballs for a Reconstructed Governor

The synchronizer for paralleling the unit, or shifting the load.

The speed droop device, for adjusting the difference between no-load and full-load speed.

The load-limiting device.

The solenoid shutdown.

The restoring mechanism which holds the control at rest at constant speed (or load).

The oil-pressure hand control.

The oil-pressure supply system.

Flyballs. Fig. 9 shows the flyballs of a modernized governor of older design, mounted in a cabinet and directly coupled to the driving a-c motor operating in synchronism with the speed of the turbine. It is of the improved (patented) supersensitive, spring-balanced high-speed design in which the centrifugal forces are directly counteracted by adjustable springs, so that only the resultant governor force is transmitted to the flyball collar, no pivoted joints being employed. The flyball collar is flexibly coupled (however, without lost movement) to a stem which rotates with the flyballs and serves as a pilot for the servomotor actuating the levers leading to the control valve. Thus the pilot describes a high-speed rotary movement which, combined with the axial movement on speed changes, requires a minimum of energy for its operation, but produces a fixed force in the differential oil-pressure servomotor actuating the leverage leading to the control valve. The speed-responsive element is thus relieved of any mechanical duty, but acts merely as a speed-responsive device, which, by means of the servomotor, at once produces a liberal force available at the slightest speed change.

These flyballs respond safely to 0.01 per cent above and below speed, so that prompt control of the output is assured, resulting in sensitive speed control with practically entire absence of tendency to hunt. The servomotor is coupled to the center of a single-arm lever, one end being adjustable by a handwheel or by an electric motor for switchboard control, starting, paralleling, or stopping the unit, and for changing the output when paralleled.

The opposite lever end is pivoted to the main floating lever, one end of which actuates the control valve, the opposite end receiving the movement from the servomotor actuating the output control of the unit. An adjustable relay-compensating oil-dashpot device is inserted into the relay connection, which permits of small droop compensation when the movement of the servomotor is slow, and a temporary larger droop on quick movement with gradual return to the smaller droop, and which is adjustable from 6 per cent to $1^1/_2$, or even 0 per cent when the unit does not operate in parallel with other alternators.

An adjustable load-limit device, either hand- or remote-

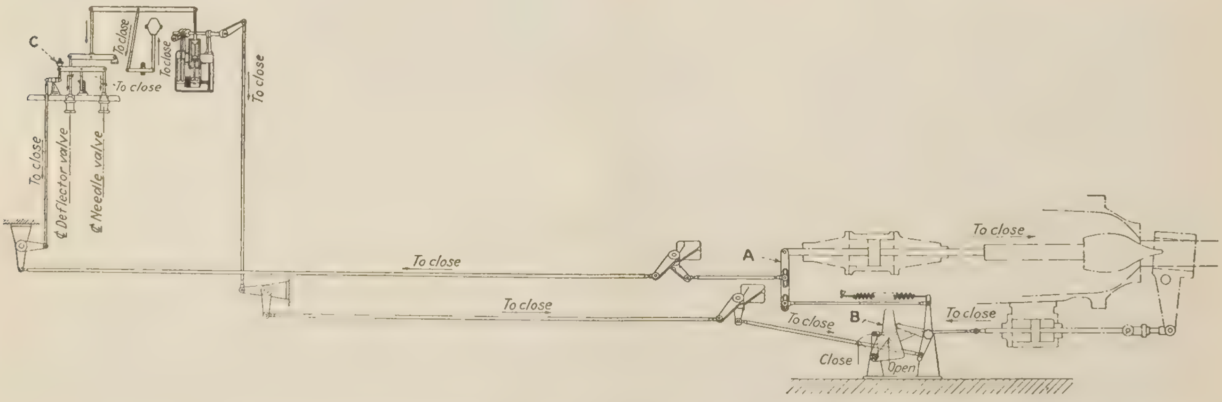

Fig. 10 Operation of Restoring Mechanism for Governors at Glenville Plant; 30,000 Hp, 257 Rpm Double Overhung Impulse Wheel

[(1) *The deflector:* The deflector regulating valve is operated by the governor flyballs through the floating lever linkage; this valve is restored to midposition through the action of the restoring mechanism operated by the deflector piston. (2) *The needle:* The position of the needle relative to the deflector lever *A*—that is, when the flyballs move to deflect the jet, floating lever *A* is shifted clockwise by the cam *B*. The needle-regulating valve then admits oil to the closing side of the needle-regulating cylinder. Floating lever *A* is again rotated clockwise until the regulating valve is restored to midposition and the needle motion stopped. Handwheel *C* is the device for shifting the relative position of the needle to the deflector.]

electric-operated, permits of holding the output within desired upper limits without preventing complete no-load action in case of total load rejection.

The solenoid shutdown device acts upon the load-limit device for the purpose of complete shutdown of the unit in case of failure of current to the flyball motor, or for any other protective features, such as excessive bearing temperature, overload, or overvoltage.

Where no independent mechanical hand control is combined with the main servomotor, a hand control by means of oil pressure is provided in order that the automatic parts of the governor can be examined or repaired without causing a shutdown of the unit.

The governors of the Nantahala unit are performing a combination of deflector and needle movement, involving two control valves and a combination relay mechanism. This is illustrated diagrammatically in Fig. 10.

The floating lever actuated by the flyball servomotor is pivoted not directly to a control valve but to the center of a single-arm lever with a fixed point on one end and connection on the opposite end to the control valve actuating the servomotor which moves the jet deflector, the movement of which is transmitted to a relay shaft carrying a lever B with a cam against which a roller is held by a tension spring. The roller is part of a lever keyed to a relay shaft carrying two levers, one with relay connection to the compensating oil dashpot actuating the end of the floating lever for the purpose of restoring neutral or dead-beat condition.

There is another lever keyed also to the relay shaft which transmits the movement of the roller to a floating lever, the outer end A of which receives the movement of the servomotor actuating the needle or, if the latter is stationary, serves as fulcrum and thus causes a second relay movement transmitted over a double-arm lever to the control valve actuating the needle. By proper cam movement the final dead beat or neutral position of the control is such that the jet deflector touches the jet so that water is not wasted, nor is there a dead or lost active movement of jet deflector, all of which assures prompt speed control.

Since the deflector can act quickly while the needle must move slowly only by reason of the rate-limit devices controlling the flow of oil directly in the needle servomotor, as required by the long pressure pipe line, the stroke at point C may cause an excessive stroke of needle-control-valve piston. To prevent this the fulcrum of this lever is made flexible; however, always tending to restore to normal position.

By means of a handwheel at C, the relative position can be changed between jet deflector and power needle.

Likewise, the shape of the cam surface of lever B can be so formed that at small outputs the jet deflector does not clear the jet until it has to assume a certain size in accordance with the output demand.

This particular feature has proved especially valuable, as will be set forth in the discussion of the special requirements of speed control of the local power system which the Glenville unit is to take care of at times.

Flyball Drive. Current to the motors driving the flyballs of both governors is supplied by an independent generator directly coupled to the exciter of the generator. It is of the permanent-magnet principle, 2-phase with common circuit for connection to the two flyball motors.

The oil-pressure system serves each governor independently, or can be interconnected. The 15-hp a-c-motor-driven oil pumps pick up the oil from the two-compartment receiver tank directly below, and deliver it to the respective pressure tank, a check valve preventing backflow when the pump does not deliver oil under pressure. Combined with the pump is an unloader valve which opens when an oil pressure of 300 psi is reached and closes at a lower limit of about 270 psi in the pressure tank. With the unloader is combined a relay switch which shuts the motor down when unloading and starts it when oil must again be delivered to the pressure tank.

The oil returning from the governor discharges over a strainer into the receiver tank. All individual elements of this pump system are arranged to be readily accessible.

Setting of Unit Due to Flood Conditions. Fig. 11 shows the complete hydroelectric unit. It will be noted that no windows are provided at elevations below the top of the generator stator.

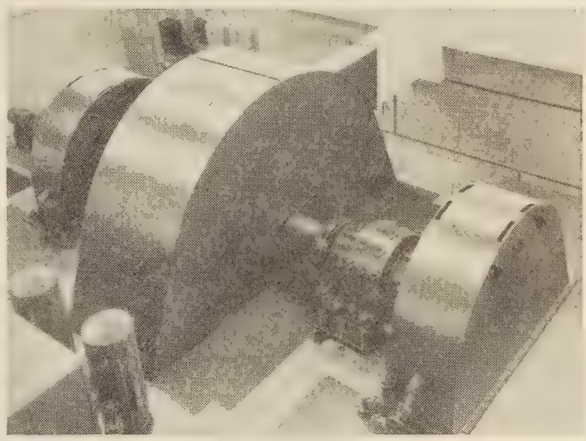

Fig. 11 Interior of Station Showing Turbine and Generator

The powerhouse is located in a gorge, and abnormal floods in the river might raise the water level so far above the floor that even the top of the generator would be submerged. This would drown the two water wheels, thereby materially reducing the output, if not rendering operation impossible. Since such floods are exceptional and of short duration only, it would be a waste of available head if the unit were placed safely above exceptional flood-water elevations. It was calculated that the kilowatthours permanently lost in normal times by this higher setting (and loss of head) would exceed the loss of kilowatthours output during the floods, with plant shut down completely. It was therefore economically feasible to locate the unit for the normal head condition, and merely shut it down during flood periods by providing tight-shutting gates at each wheel-pit outlet to keep the water from entering the wheel housings and putting them under pressure.

A system has been worked out which would permit setting the unit as high above lowest tail-water level as possible, putting the wheel housing under slight vacuum at the high head and under slight back pressure at extreme flood levels, however, without drowning the buckets. With this system the water level is held at safe distance by supplying compressed air, similar to what is now accomplished with Francis and propeller units, operated idly as synchronous condensers, so that they revolve in air instead of absorbing considerable power when revolving idly and submerged at normal speed. If the flood periods at Glenville were of longer duration it would have been economically feasible to provide for such an arrangement including auxiliary compressor equipment, which can be made automatic in case floods arrive unexpectedly.

Pressure and Speed Control. With a pressure conduit exceeding 16,000 ft in length, the critical time $\frac{2L}{a}$ at which a maximum

pressure rise of $\frac{aV}{g}$ will result is over 8 sec. Therefore, it necessitates a very gradual deceleration and acceleration of water on sudden load rejections and load increases in order to avoid pressure variations of a destructive value. Pressure control is assured by the rate of movement of the flow-controlling needles, which is fixed by admitting the oil into and out of the needle servomotors at a rate individual for each position of needle, entirely independent of the rate of governor action.

The system of the Nantahala Power and Light Company is connected through a rather long transmission line over mountainous country with the Aluminum Company's transmission line at Santeetlah, N. C. From Santeetlah the power is transmitted over the lines of the Aluminum Company of America for a considerable distance to Alcoa, Tenn., for use by the Aluminum Company. At Alcoa, the Aluminum Company's system is interconnected with the T.V.A. system.

Under these conditions, the regulation of the system will not be seriously affected when one of its large units becomes suddenly detached.

However, there are times when the Nantahala system is disconnected from the Aluminum Company's system at Santeetlah and is therefore isolated from both the Aluminum Company and T.V.A. systems. When this occurs, the system load of the Nantahala Power and Light Company amounts to a maximum of approximately 5000 kw. This remaining load has relatively wide fluctuations due to the operation of a paper mill at Sylva, N. C. Except for the Nantahala and Glenville plants, the remaining generator plants of the Nantahala Power and Light Company are too small to carry the load fluctuations satisfactorily.

The Glenville Plant was put into operation several months before the Nantahala Plant was completed.

The power company does not plan to use the Nantahala Plant for regulation of its local load due to the fact that it is a Francis-type turbine with a long tunnel and pipe line, operating under an exceptionally high head which requires slower governor-operating time, and which subjects the turbine parts to extreme cavitation and wear when operated at light loads.

It is therefore the duty of the Glenville unit to take care of the speed control when the load is restricted to the local system alone.

Under such conditions, prompter action is required than can be secured with the slow rate of movement of the needle. In other words, the water-wasting stream deflector must actively participate in the control of the momentary load changes of the isolated, small system.

When the unit operates in parallel with the Aluminum Company system, and in turn when that system is operating in parallel with T.V.A., the flywheel effect of all the rotating masses operating in synchronism is so large that even large, sudden load changes will cause only gradual changes in speed; in fact, so gradual that the slow-moving needle may be adequate to take care of it without causing the stream deflector to cut into the jet on sudden load rejections, i.e., if such load rejections do not increase the frequency more than 62 cycles.

At loads above 4000 kw per jet, the relative position of the deflector and needle is such that the deflector does not disturb the water stream issuing from the needle valve and therefore in no way influences the efficient operation of the unit.

At loads below 4000 kw per jet, the relation is such that the stream deflector remains partly in the jet, thereby deflecting some water from the buckets so as to be ready to back out of the jet promptly on load increases which the needle could not take care of by reasons of its slow rate of movement. Thus a materially better speed control is obtained for the small isolated system, although it inevitably involves a slight wastage of water at loads below 4000 kw.

Two requests of the power company had to be met:

1 That the frequency of the machine following full load rejection should be held to a minimum possible value, and that the unit should be promptly brought back to synchronous speed so as to be ready for prompt paralleling in a minimum length of time.

2 That penstock water-pressure swings, following full-load rejections and during light-load operation, should be held to a minimum.

To meet the first requirements, an overspeed switch was adjusted to close contacts at approximately 62 cycles. This over-

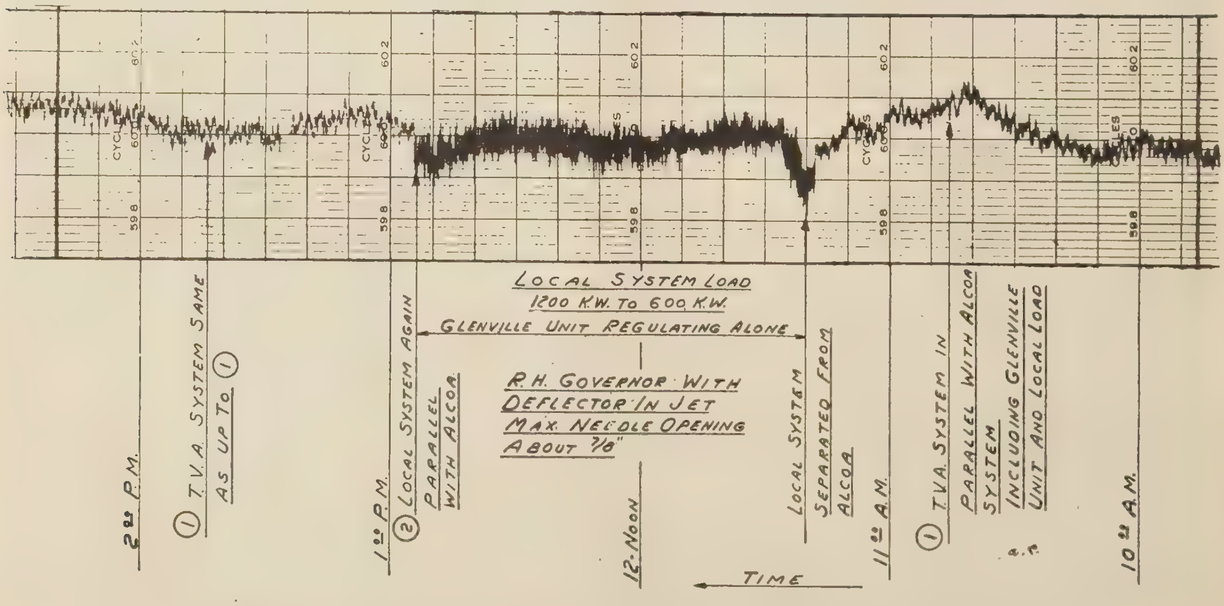

Fig. 12 Record of Cycles on March 31, 1942, Under Different System Operation of Glenville Unit
(Voltage-limiting device is subsequently adjusted to release 15 sec earlier to allow voltage to return to normal in 40 sec after trip of breaker.)

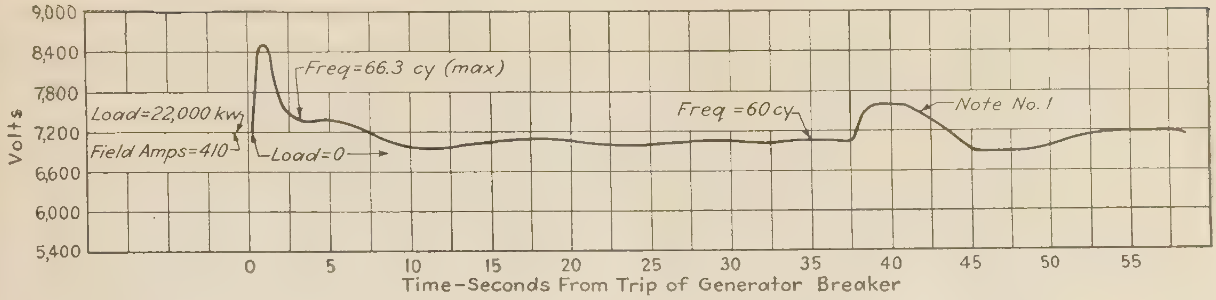

Fig. 13 Curve Showing How Voltage at Generator Terminals Returns to Normal After a Load Rejection of 22,000 Kw

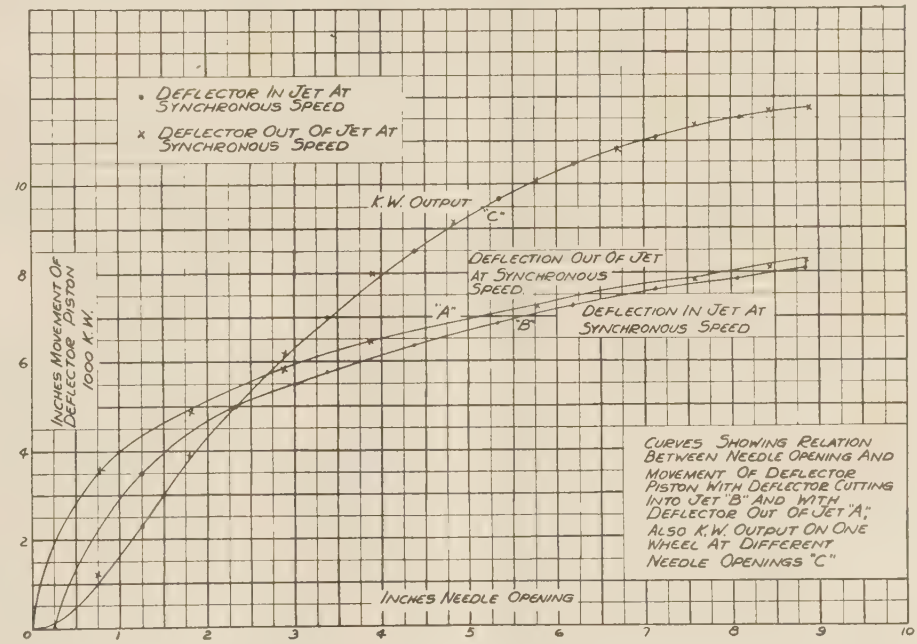

Fig. 14 Curves Showing Relation Between Needle Opening and Movement of Deflector Piston With Deflector Cutting Into Jet B, and With Deflector Out of Jet A; Also Kw Output on One Wheel at Different Needle Openings, C

speed switch energizes a solenoid-operated valve to admit oil pressure to the deflector and also energizes the circuit to the synchronizing motor to operate it in the lowering direction. The synchronizing motor continues to operate until it returns the mechanism to the speed no-load position where the motor is stopped by means of a limit switch. The solenoid remains energized until the frequency returns to approximately 61.5 cycles. By means of these two devices, the latter of which was suggested by H. H. Gnuse, Jr., electrical engineer of the power company, the speed rise on rejection of 22,000 kw has been held to 66.3 cycles, as measured by a standard frequency meter, and the machine speed and voltage have returned to normal values within 35 sec, so that the unit was ready to be reparalleled with the system. These tests were made with a speed-droop setting of approximately 5 per cent.

During this load rejection, the penstock surges were very minor due primarily to the fact that, with the present adjustments, the needle never completely closes until the gate-limit adjustment has been operated to the zero position. The slight opening which remains in the needle valve very effectively prevents the customary penstock pressure surges which are obtained upon complete closure of such a valve.

Fig. 12 shows a record of the cycles on March 31, 1942. Up to point marked (1) it shows the cycles of the systems, Nantahala, Glenville, Alcoa, and T.V.A. interconnected, the Glenville unit operating with one water-wheel side only. At point (1) Glenville was detached from T.V.A. and Alcoa, to take care of the Nantahala system alone, with units at Franklin Plant and Bryson City operating on blocked governors to limit all frequency control to one Glenville governor. The load was around 600 kw with a total maximum of 1200 kw.

It will be noted that even one governor of the Glenville unit held the cycles within very satisfactory limits.

Fig. 13 is a record taken August 11, 1943, showing that voltage at generator terminals returns to normal after a sudden load rejection of 22,000 kw. The curve also indicates that the speed returned to normal 35 sec after the load change.

Fig. 14 shows the kilowatt output curve C and the inch movements of the servomotor piston actuating the deflector, curve A when the deflector does not cut into the jet under stationary needle positions, and B when it is set to cut into the jet. The points marked X on curve C apply to curve A, and those marked (·) apply to curve B. It will be noted that the outputs for the various needle openings do not vary appreciably whether the deflector cuts into the jet or not at normal speed. The adjustment, it will be noted, is so made that, at small jets (or needle

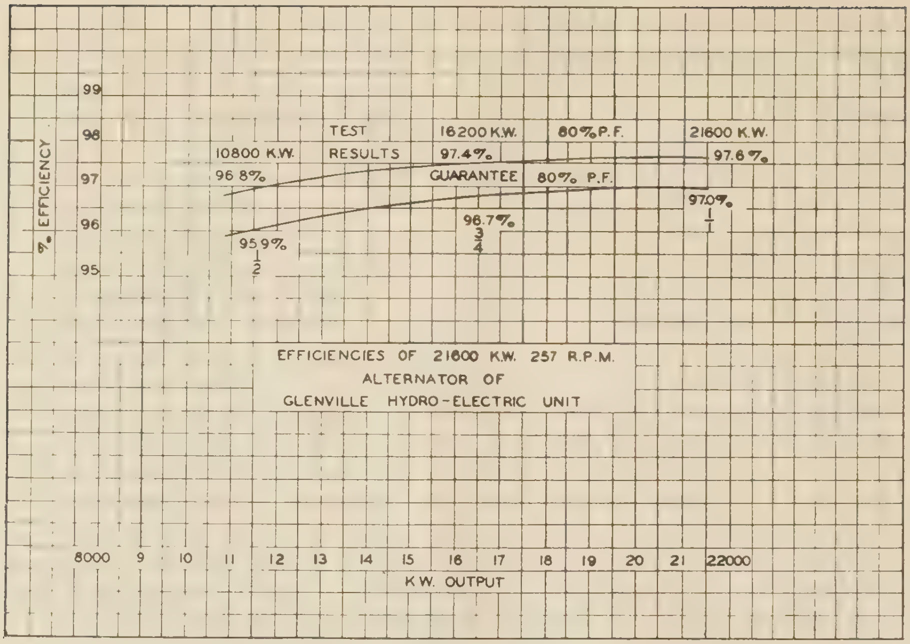

Fig. 15 Efficiencies of 21,500-Kw 257-Rpm Alternator of Glenville Unit

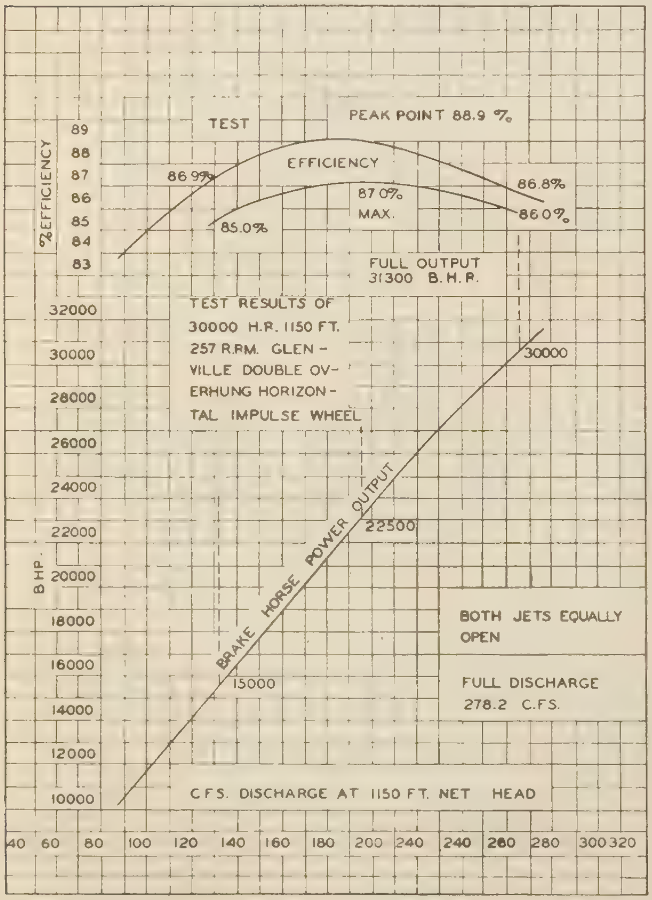

Fig. 16 Test Results of 30,000-Hp, 1150-Ft, 257-Rpm Glenville Double Overhung Horizontal Impulse Wheel

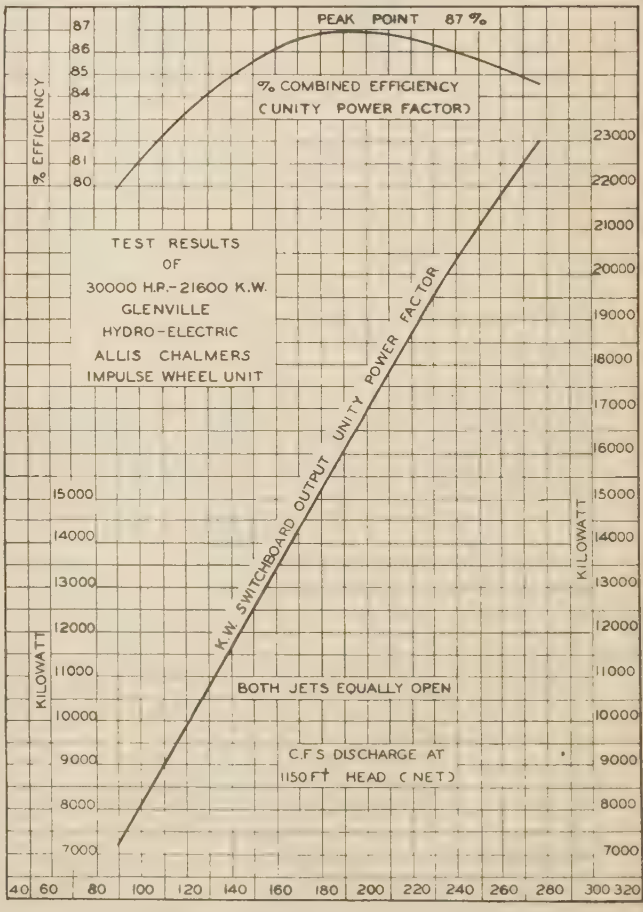

Fig. 17 Over-All Efficiencies of Glenville Unit

strokes), the deflector cuts into the jet and is clear of the jet at larger needle strokes or outputs. This is desirable to insure equally close regulation on very small loads of the isolated Nantahala system.

Generator. The generator associated with this unit is rated 27,000 kva, 80 per cent power factor, 6600 v, three phase, 60 cycles, 257 rpm. The rotor is designed to provide the required 4,400,000 lb-ft^2 moment of inertia without the use of a separate flywheel. The poles are dovetailed into a laminated "floating" rim which is keyed to the fabricated spider. A damper winding is provided in the pole faces. The damper winding has no connections between poles but has the bars silver-soldered into the pole end plates. The stator core is made up of low-loss high-permeability silicon-steel laminations supported in a box-type welded plate steel yoke. The stator winding is of the double-layer lap-wound type, designed for minimum eddy-current loss.

A number of standard 10-ohm temperature-detector coils are embedded between coil sides. The generator is completely enclosed and is cooled by air circulated by axial-flow fans mounted on the generator rotor. Surface coolers are mounted below the generator to maintain the temperature of the in-going air to the generator at not more than 40 C. A main exciter and pilot exciter are direct-connected to the shaft. A rocking-contact-type regulator in the main exciter field controls the generator voltage.

The performance of this generator, as indicated by the tests made in August, 1942, and by the operating records is fully satisfactory. Conventional efficiency is more than 0.5 per cent above guaranteed values. Operating temperatures are well within those recommended by the standards.

Efficiency Tests

To determine the efficiencies of the entire hydroelectric unit, tests were made at the power plant, first to establish the generator efficiency to arrive at the actual brake-horsepower-output curve at the generator shaft.

The generator tests were made by purchasers' engineers and according to the A.I.E.E. Test Code.

Fig. 15 shows the guaranteed and the generator efficiencies actually obtained, indicating a substantial margin above the guarantees.

The water-wheel efficiency of the unit was determined by Dr. Norman R. Gibson's method, and in accordance with the rules of the A.S.M.E. Test Code in force.

Fig. 16 shows these guaranteed and the actually obtained efficiencies, indicating also a substantial margin above guarantees made.

Since the entire hydroelectric unit was designed and built "under one roof," so to speak, the individual test efficiencies of water wheel and of generator can be combined into one over-all efficiency from cfs discharge at normal contract net head to kilowatt output measured. These values are plotted in Fig. 17, as is also the corresponding over-all efficiencies as computed from the individual guarantees of water-wheel and generator efficiencies.

For a hydroelectric unit designed, built, and guaranteed by one single manufacturer, the "undivided responsibility" would permit of a simplification of the tests, in that a guarantee made from "water to switchboard" eliminates the necessity of establishing the individual efficiencies of the generator and turbine.

The Glenville unit is the only large impulse-wheel unit east of the Rockies, and its performance establishes a record in this country as regards water-wheel efficiency of practically 89 per cent.

Acknowledgments

The writer wishes to express his appreciation and thanks to Mr. J. E. S. Thorpe, president of the Nantahala Power and Light Company, and to his staff for the valuable assistance and co-operation rendered by contributing plans, photographs, and operating data.

Discussion

Robert Lowy.[2] Having had 25 years of experience in the construction of European Pelton wheels, the writer would like to point out certain differences between the trends of construction abroad and in this country, as follows:

1 The safety of the construction indicated in the paper is highly insured. The author himself mentions that the design, from a viewpoint of mechanical strength, is liberal. It may even be that the construction does not give a true picture of the current utilization of materials in the United States.

There was a great difference in Europe between constructions before and after the first world war; economy in the use of materials was far greater after the war.

This condition does not appear to have existed to anything like the same extent in the United States, owing to the fact that losses through the war were not as great, and there was no impoverishment. Perhaps this may change after the present war. However, we must not forget that, at the same time, impoverishment in Europe will be far greater than in America, and utilization of materials and labor in Europe will be forcibly magnified.

The construction of the Glenville plant as described is surely far safer than European installations. The principle of streamlining is also more highly developed than is the usual European practice.

2 A pertinent example is the construction of a turbine and generator for runaway speed. The price difference between a wheel which is built for a 1.8 runaway speed and for the runaway speed of only 1.2 is not very great; but, for the generator, the price difference is a multiple of the generator price. The runaway speed will in all likelihood never be reached. The European point of view is now that such constructions for complete runaway have in some way an exaggerated safety and indicate use of superfluous materials and labor. This amounts to a loss of national wealth. It would be the same as though a dam built for a 200-ft head were used for a 150-ft head. In Europe all large Pelton turbines have a special safety device which prevents racing of the turbine. This control is independent of the oil governor and also of the system of speed regulation. The safety device works hydraulically with the water pressure, sometimes with dead weight. Such constructions are indicated in a previous paper[3] by the writer.

A fundamental rule in using such safety devices is the instruction that these contrivances have to be tested either weekly or monthly. In this way operating personnel will become accustomed to having the unit running at higher speeds, limited by the safety device, and will not become alarmed in the event of racing.

Nevertheless, the generators for the Pelton plants in Europe are built up to date for the runaway speed, but owing to the safety device employed, the safety factor of the generator construction is tacitly lowered. Even with such construction, the safety of such units is far greater than that of steam turbines.

It seems there now exists an understanding among the producers in Europe to make the customer acquainted with the safety device to show that such a safety arrangement is more

[2] Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

[3] Yearbook of Waterpower, 1930–1931.

practical, more economical, and really provides a sufficient factor of safety. In a few years, maybe directly after the war, the construction of such turbine units to take care of runaway speed will be eliminated, and the arrangement of an efficient safety device will be all that is necessary, as in the case of steam turbines. As the price of the safety device is small in comparison with the construction for runaway speed, a real saving results. These remarks intend only to show the trend in Europe without recommending the use of them under existing economic conditions.

In studying the data given in the description of the Glenville Plant, the writer fails to find the closing time of the deflector servomotor. It would be interesting to know these relations because the speed rise by rejections of full power seems to be extremely high. The writer's experience indicates that to attain a smaller speed rise by full-power rejections is not so difficult. It is most difficult to obtain relatively small speed rise with small load rejection.

D. J. McCormack.[4] It is to be regretted that other American manufacturers have not accomplished any real development work on impulse wheels for many years past. European turbine builders have had engineers specializing on impulse turbines, and exhaustive studies and tests have been made continually to improve the results. Consequently, even 15 years ago, they were obtaining 89 to 91 per cent efficiency on field tests of impulse turbines. At that time American manufacturers thought they were achieving great results if they got 84.5 to 85.5 per cent efficiency. As a result, some of the large American installations were obsolete when they were installed. This turbine under discussion, showing 89 per cent efficiency on field test, is not in any way exceptional. If it showed 91 per cent efficiency or over, that would be notable. Seven years ago, the writer's company had field tests of 88 per cent on small units of only 2700 and 7000 hp.

Upon questioning, the author has stated that he would not take a chance with the American steel foundries, and cast the buckets solid with the disk or in a continuous ring. To date the writer's company has supplied 75 such runners up to 90 in. impulse diameter, which have stood up in service and are continuing to give fine results. Excellent castings are being produced which show up well under most rigid inspection. The construction is stronger, and a greater number of buckets can be used to obtain higher efficiencies. All buckets wear alike, so there is no advantage in replacing just one bucket. A whole new runner can be supplied for less than the cost of one set of buckets.

The author evidently does not favor the use of elbow-type nozzles. Of course, the answer to this is that higher efficiencies are being obtained with the elbow-type nozzle than with the straight-line nozzle originally developed by The Pelton Water Wheel Company, and the one under discussion which is only a slight modification of the Pelton design. However, the elbow-type nozzle must be designed properly.

E. B. Strowger.[5] The author has indicated that the pressure conduit of the Glenville unit is something over 16,000 ft long, and that no surge tank is installed. Further, the critical time of the pipe line is 8 sec. As a consequence, a long time of closure was chosen for deceleration of the flow. While the jet deflector acts to unload the wheel in about 2 sec, the needle stroke is timed to about 100 sec for full load.

The plant was tested by the Gibson method and, in view of the foregoing conditions, it may be of interest to describe briefly the application of this method and the results obtained. On

[4] Sales Manager, S. Morgan Smith Company, York, Pa. Mem. A.S.M.E.

[5] Hydraulic Engineer, The Niagara Falls Power Company, Niagara Falls, N. Y. Mem. A.S.M.E.

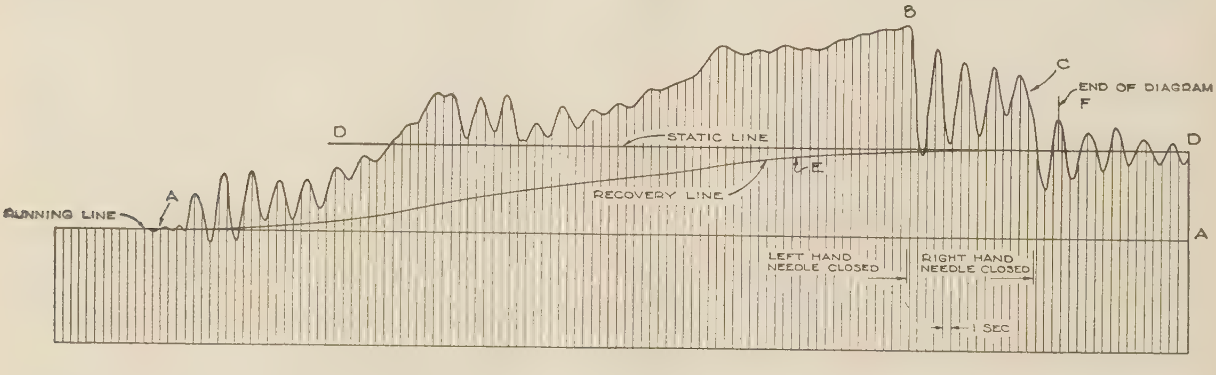

Fig. 18 Gibson Diagram 100 Per Cent Nozzle Opening
(Measured cubic feet per ser second, 280.1.)

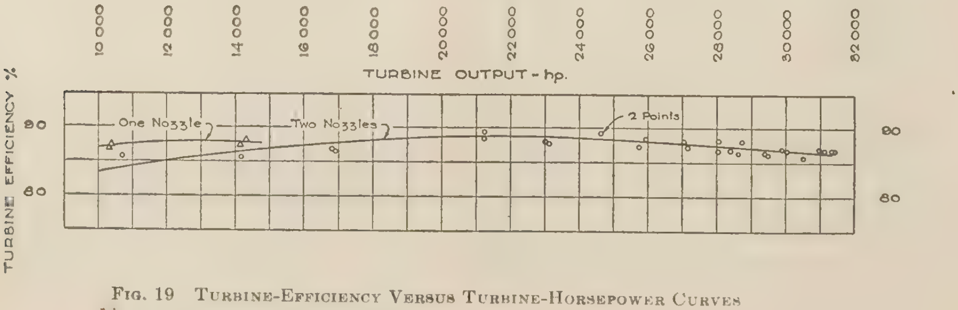

Fig. 19 Turbine-Efficiency Versus Turbine-Horsepower Curves

account of the long closing time, it was necessary to utilize a relatively long length of conduit in order to secure a pressure-time diagram of satisfactory size. The Gibson piezometers were therefore placed 378 ft apart. Using this length of pipe, the differential pressure-time diagram shown in Fig. 18 of this discussion, was obtained for the rejection of full load. The velocity in the pipe line was reduced from about 10 fps to zero during the gate closure.

The delineation of this diagram shows an area of approximately $13^1/_2$ sq in., which corresponds to a discharge of 280 cfs flowing in the pipe just before closure started. During each test run, the two nozzle openings were adjusted to the same value, and it was assumed that each wheel used 50 per cent of the total discharge. The timing of the two needles was not adjusted prior to the test to exactly the same value. For the diagram shown, the left-hand needle had a closing time of 88 sec, and the right-hand needle had a timing of 103 sec by stop-watch measurement. This can be checked from the diagram by counting the second marks from *A* to *B* for the left-hand needle and from *A* to *C* for the right-hand needle. *A-A* is the running line of the diagram, *D-D* the static line, and *A-E* the recovery line. The end of the diagram is at *F*.

Since the author showed the efficiency curve of the wheel plotted with cfs as abscissas, Fig. 19 is presented to show the variation of efficiency with turbine horsepower. The test points are shown on the curve which is very flat from 10,000 to 31,000 hp.

Author's Closure

Dr. Lowy's remarks that the safety factors of American water-wheel designs are materially higher than those of Europe are undisputed. It is a question as to whether the element of safety is to be invested in the unit itself, or into a safety device as an auxiliary. Today many power companies, well experienced by long years of operation of units, still prefer to witness a unit operating safely at its highest runaway speed without a safety device, because such a safety device does not guarantee absolute absence from failure even when its action is being witnessed during one test case.

Mr. McCormack's statement that field tests made seven years ago with units of his company's make showed efficiencies of 88 per cent should have been backed by a proof of authenticity equal to that of the Glenville test. It may be stated also that the author's company built impulse wheels with straight-flow needle-controlled nozzles as early as 1908, although not yet of the capacities of the world record as to physical size of the wheel at San Francisquito No. 1 plant of the Bureau of Power and Light, Los Angeles, Calif.

Mr. Strowger has answered Dr. Lowy's question by stating that the stream deflectors are moved in two seconds. This does not preclude the possibility of even a quicker rate if it should be desired.

Efficiency Analysis of Pelton Wheels

By ROBERT LOWY,[1] PHILADELPHIA, PA.

This paper is devoted to a discussion of the various flow losses in a Pelton bucket. Special attention is given to whether or not such losses are avoidable. Three phases of flow are investigated in detail, i.e., the entrance, the main flow, and the outlet. The author describes the application of a new method to study such movements. The investigation is carried out on a stationary bucket with a jet having the same entrance conditions as the jet on a rotating bucket. The influence of the lip on the entrance conditions is shown by example, and the conditions in the deflected part of the jet are indicated. The loss through the deflected jet is shown in kinetographic representation. A notable loss due to crossing of streamlines on a bucket is discussed. Finally, a list of all possible losses is tabulated.

THE efficiency of Pelton wheels heretofore built has been relatively high, and it is probable that the securing of such good efficiencies is the cause of the lack of much close study and analysis as applied to this type of unit. On the other hand, more study has been given to the reaction type of hydraulic turbines, both of the Francis and propeller type, due to the past necessity for improving the efficiencies of such units.

A stage has now been reached, however, where efficiencies of the reaction type often surpass those usually obtained from the impulse type; and it seems desirable to develop methods whereby the impulse-wheel efficiencies may be raised as the result of analyzing the various losses with a view to reducing them.

Thus it became obvious that the current simplified theory is not sufficient, and that a more exact and comprehensive examination of the flow would be necessary. The investigation of the flow of a jet on a revolving bucket is very complicated, and a complete theoretical examination is not feasible. It is possible to find certain hydraulic potential motions which have some resemblance to the flow mentioned; but these examples are relatively far from the true condition. Nevertheless, they are of great help in arriving at a better understanding and a clearer explanation of the actual phenomena.

Pelton-Wheel Nozzle and Bucket Details

The essential parts of a Pelton wheel involved in this analysis are the nozzle and the bucket. The nozzle consists of the orifice ring and the needle. Both parts form the control valve for the power-supply line. The bucket consists of two ellipsoidal half-shells connected with a common middle edge, the splitter. The outer edge of the bucket is provided with a lip which varies greatly in form for various designs. The lip results from the need to form a suitable entrance for the jet coming from the nozzle.

One of the principal differences in the general appearance of buckets lies in the depth. There are flat and relatively deep buckets. Nevertheless a practical investigation reveals that the relative depth has no determining influence on the efficiency, regardless of the specific speed.

The efficiency of a complete wheel, obviously, can be divided into efficiency of the nozzle and efficiency of the wheel itself, these parts being absolutely independent of each other.

The efficiency of a nozzle can be found by direct measurements; and it has been established that, in general, a loss of 2 to 3 per cent can be attributed to this part of the unit. A precise test of a special nozzle and needle at full opening gave the highest efficiency of 99.8 per cent. At 15.4 per cent opening the efficiency was 98.1 per cent. Material improvement of these values is not possible.

The efficiency of the wheel itself results from a great number of different losses. All these losses are relatively small, but in total they are considerable, so that the complete efficiency of a Pelton wheel reaches a maximum of 87 to 91 per cent.

Any increase in the efficiency can be obtained only by reducing, one by one, the separate losses. Therefore it is particularly essential to determine all the losses and to know how to deal with them.

This study will consider all losses in a bucket, and special attention will be given as to whether or not the loss is inevitable.

Losses in Pelton-Wheel Bucket

It has been found in many practical examples that low efficiency occurs mostly due to losses which are not known generally, and which are in some degree avoidable.

In considering the analysis of the losses, first we have those due to extraneous forces which are not dependent upon the flow. These are mechanical losses in friction in the bearings, and the air resistance of the rotating wheel. Both are inevitable and have to be minimized by proper adjustment.

Concerning the losses dependent upon the flow, we must distinguish principally three phases, as follows:

1 The entrance of the jet into the bucket, partially on the splitter, partially on the lip.
2 The flow in the bucket itself.
3 The discharge or the outflow from the bucket.

Each phase shows a loss which is necessary and which can be reduced only to a certain degree. A complete elimination of these losses is impossible. They are connected with the function of the flow itself and are in a certain sense necessary in the turbine system.

The principal loss is the discharge loss. It is necessary for the water to flow away from the bucket; and therefore some velocity head must be sacrificed to do this work. Then we have the loss due to friction on the surface of the bucket. Even the smoothest surface has some resistance, but with proper surface finish this loss can be reduced greatly. Finally we have the entrance loss, a singularly involved loss, which is dependent upon the specific speed.

These three losses are inevitable and cannot be avoided. Only arrangements with highly decreased losses are possible. The efficiency of a Pelton wheel is therefore determined by at least all these losses. It is necessary to take care of all the relevant circumstances to be able to reduce these losses, by proper adjustment, and to reach a high standard of performance.

In addition to the foregoing necessary losses, there may be other losses which appear not infrequently. Such losses are connected with the flow, but they are avoidable. Therefore an arrangement should be chosen so that such occurrences will be completely avoided, or the losses should at least be reduced as far as possible.

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[1] Engineer, I. P. Morris Department, Baldwin-Southwark Division, Baldwin Locomotive Works. Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

Study of Flow in Bucket

To evaluate these losses, we have to investigate the flow in the bucket. It should be considered principally that the flow of the jet on the bucket changes because the nozzle through which the jet flows is fixed, whereas the bucket is moving. Of course, there is always an intermittent movement at the beginning of the impact of the jet. The impact of the jet on the bucket takes place both on the lip and on the splitter. Where the impact occurs on the lip, the jet is divided into two parts. One part remains flowing on the bucket and forms the impinging jet branch. The second part, the deviated jet branch, flows past the bucket and reaches the next bucket.

If the impact occurs on the splitter alone, two symmetrical jet branches result. These two branches of the jet on the bucket will then be turned back. The impact on the splitter takes place at different points, and, furthermore, every single water filament will have a different impact angle on encountering the splitter.

The construction of the impact angle, i.e., the entrance triangle, is well known and is represented in Fig. 1. In this figure, for one water filament a, the absolute velocity V, the peripheral velocity u, and the relative velocity v are indicated. Subsequently, the complete relative paths a', b', c', for three water filaments a, b, and c, are traced, which is a well-known construction. Furthermore six positions of a bucket are represented for the following cases:

1 First contact of a single water filament
2 First contact of the half jet
3 First contact of the complete jet
4 Last contact of the complete jet
5 Last contact of the half jet
6 Last contact of a single water filament.

It is necessary, of course, to choose the number of the buckets on a wheel so that the last drop encounters a bucket and no outflow between buckets is possible.

These procedures are generally known and used in practice. They contribute to a good efficiency of the bucket, but this alone does not give absolute assurance that a high efficiency will be reached.

Nevertheless, there have been cases in which buckets designed in conformity with these considerations, and with very smooth bucket curvatures, have given mediocre results. Investigations of the running wheel and the model, with and without stroboscopic instruments, have brought forth no explanations of such discrepancies. To study such cases and to find and eliminate disturbances, the author used an original method which gave excellent results.

Use of Fixed Bucket for Experiments

This method consists of a thorough investigation of the flow on a fixed bucket, but with a jet striking the bucket at the correct relative angle, under the same entrance conditions which exist on the running bucket.

The primary objection which can be made against this method is the fact that the flow on a stationary bucket is not identical with the flow on a rotating bucket. This is, in general, correct; however, the difference is not so far-reaching that this method cannot be applied. This can be proved easily by an examination of the flow.

We can distinguish three different kinds of flow motion:

1 The relative motion of the jet and the rotating bucket.
2 The absolute motion of the jet.
3 The flow of a jet on a fixed bucket with practically the same entrance conditions as for case 1. We designate this briefly as "absolute" motion but we repeat that this "absolute" motion has nothing in common with the motion in case 2.

This particular method involves an investigation of the motion in case 3 with the reservation that case 1 should be used.

We may now examine the difference between case 1, the relative motion, and case 3, the particular "absolute" motion. Regarding the course of the flow, the difference between the motions on the various parts of the bucket is not the same.

The greatest difference shows the part of a streamline which lies in a plane of rotation. This is the case which shows the relative paths in Fig. 1 (a', b', and c'). The absolute motion on the fixed bucket would be a straight line, the tangent to the relative path, therefore identical with the direction of the relative velocity v. A streamline on a bucket will run only to a small amount in a plane perpendicular to the axis of rotation. Therefore only a small difference between tangent and curved line can result.

No difference between "absolute" and relative motion will be found for a streamline which is formed by an enforced guidance in the bucket with a cylindrical surface, the cylinder being parallel to the axle of rotation. Here the relative path and the absolute path are always identical.

All other streamlines show differences which lie between these two extreme cases. If we desire to know the specific difference in the course of the flow on a bucket, we can trace a sketch which shows in which sense and to what extent such differences occur. Fig. 2 shows, on a bucket, different "absolute" and relative paths:

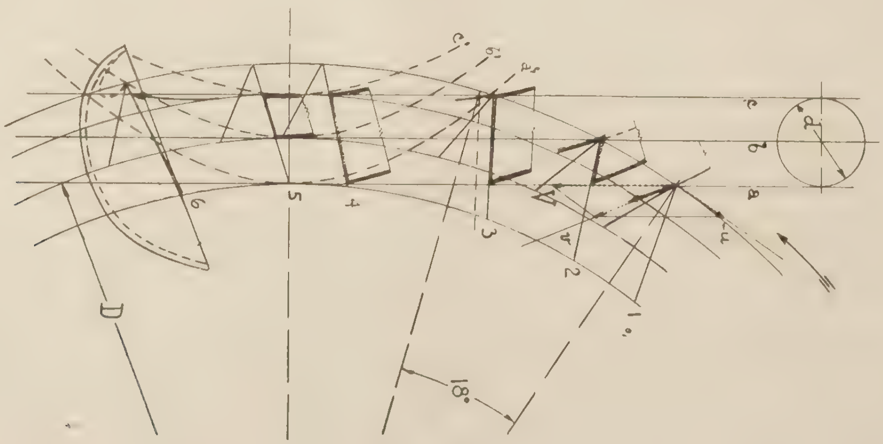

Fig. 1

The "absolute" paths are given in full lines 1, 5, the relative paths are given in dashed lines 1, 5. We conceive that the paths 1 and 5 in Fig. 2 are practically the same in both motions. The paths 2, 3, and 4 show deviation in a rotating system wherein path 3 shows the greatest deviation.

It is now essential to note that, even if relative and absolute courses of the flow are identical, the motions on the fixed bucket and the rotating bucket are not the same. This difference is, of course, due to the bucket rotation and is expressed by the value of the velocity.

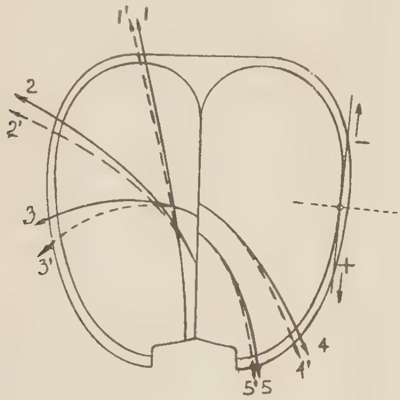

Fig. 2

In a stationary system, the velocity remains constant, if we do not take the losses into account. In a moving system with free flow, that is, with no essential pressure in the flowing liquid, the relative velocity is no longer constant but maintains the following relation according to the fundamental equation: For two points, 1 and 2, with the corresponding velocities, we have

$$\frac{u_2^2 - u_1^2}{2g} + \frac{v_1^2 - v_2^2}{2g} = 0 \qquad [1]$$

where u_1 and u_2 are the peripheral velocities, and v_1 and v_2 are the relative velocities.

This is an equation which could be derived also from mechanical principles alone inasmuch as in the free flow there is no difference in pressure. The sense of this equation can more easily be understood by a simplified example. If we suppose that $u_2 = v_2$ (and this will be nearly the case with high efficiency), then also $u_1 = v_1$. Now it can be established in general that "the relative velocity in the rotating system changes in the same degree as the peripheral velocity."

We now see clearly that the motion on a bucket in a fixed system, as compared with a rotating system, has a different velocity, even if the path should be the same. If, for instance, the particle comes to a greater radius, the relative velocity increases and vice-versa. But, as we know, the difference in the peripheral velocity of a bucket is relatively small, due to the fact that the bucket for low and moderate specific speeds is small in comparison with the diameter of the wheel. Therefore, there can arise only a small degree of difference in the relative velocity.

Equation [1] provides an explanation of the mixing loss. We conceive that the relative velocity at one point of the water filament will be different from the relative velocity at another point of the water filament c. But both relative velocities are not connected by a relation similar to Equation [1]. This will be clear by the way in which the relative velocity was found, see Fig. 1. If the two water filaments, a, c, in the relative system, should represent only mechanically separate particles, and if the relative flow of these particles were in a manner free and independent, then the existence of different relative velocities would be possible.

But both filaments, a, c, are, in fact, not independent; they represent streamlines of a hydraulic continuity, i.e., confined flowing liquid, in which, between filaments, pressure transformations sometimes arise. The relation for this transformation can be given in a form similar to the Bernoulli equation for absolute motion. We have in an absolute system, the Bernoulli relation with

$$\frac{V^2}{2g} + \frac{p}{\rho} = \text{const} \qquad [2]$$

In a relative system there exists the relation

$$\frac{v^2}{2g} + \frac{p}{\rho} = \text{const} + \frac{u^2}{2g} \qquad [3]$$

It is important to know that such a connection exists between the velocities of a closed stream in a relative system. The constant can be regarded as similar to an energy capacity and remains, for a closed system, invariable. It is essential to understand that, in the present case, the single filaments do not have the same energy capacity, due to the fact that the relative velocity is not even at the same radius.

To explain this, we can make a comparison with a reaction turbine using water at different pressures. We conceive that in one runner this cannot be done without losses. If we try to use water at two different pressures in one runner, we receive a mixing of both pressures with a loss in energy. It is, in fact, necessary that an exchange in the energy capacity take place between the various water filaments on the bucket.

To show these relations in a practical example, consider Fig. 1, as the diagram for a wheel with the following data:

Head		1000 ft
Jet velocity	$V = \sqrt{(2gH)} =$	252 fps
Water quantity		24.2 cfs
Jet diameter	$d =$	4.2 in.
Wheel diameter	$D =$	59.6 in. = 4.97 ft
Power		2380 hp
Wheel speed	$n =$	450 rpm
Ratio	$D/d =$	14.2
Specific speed	$n_s =$	3.87

Now examine Fig. 1, position 3, "first contact of the complete jet," and we have the following velocities for the border filaments:

Peripheral velocities	113.5 fps	129.4 fps
Relative velocities	148.5 fps	133.0 fps
The corresponding heads are:		
For peripheral velocities	201 ft	261 ft
For relative velocities	345 ft	276 ft

The energy capacity measured in feet, as difference, is therefore 114 ft and 15 ft.

The jet on the surface of the bucket should present, in relative motion, a hydraulic continuity, with uniform energy capacity in every filament. But this is not the case. Therefore an energy exchange between the different filaments has to take place, owing to the fact that all filaments on the running wheel do not start with the same energy capacity.

The loss consists in the appearance of vortex motions between the various water filaments with an equalization of the different energy capacities, i.e., a mixing loss.

However, there is no need for a complete equalization of the energy to take place. If two filaments should be separated, then the closed system would be disturbed, and every filament could

remain separated with its own capacity, and no further mixing loss would occur.

This difference in energy capacity increases essentially with greater specific speed, that is, if the diameter of the jet is relatively larger in comparison with the diameter of the wheel. In this case, the influence of these appearances on the efficiency of the wheel is very unfavorable.

In case of an investigation of a stationary bucket, there is no mixing loss, the water impinging on the bucket with the same speed throughout. This fact consequently results in a difference between relative and "absolute" motion.

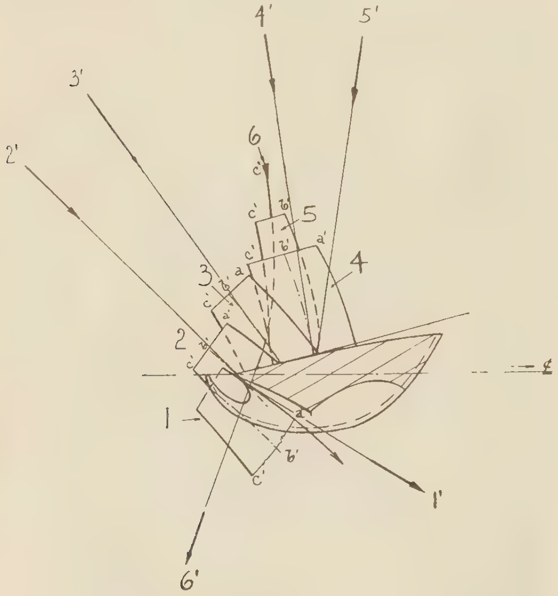

Fig. 3

Consideration and reflection show now that all these divergences and velocity differences are not so far-reaching that the general character of the flow would be changed. In fact, appearances in a stationary system are practically the same as in a rotating system.

By such means, it is possible to make an investigation of the flow on a stationary bucket, and this contributes deeply to the knowledge of the flow, as has been found by experience with this method. If desired, and as sometimes may be necessary for scientific investigations, it is also possible to rectify exactly the motion on a stationary bucket to obtain a precise reproduction of the flow in a rotating system. For practical purposes, however, an exact rectification is not necessary. Besides, we know that the flow of a jet on a surface takes place with a spreading effect, with an increase of the width. Then the outer water filaments of the surface are no longer parallel. The calculation could be made only for the filament which represents the center of gravity. But the side deviation of the border streamlines cannot be computed. The calculation is therefore from the beginning not very helpful.

We have defined the "absolute" motion as a flow on a bucket which shows the same entrance conditions as the flow in relative motion. In Fig. 3 the different positions of the impact of a jet on a stationary bucket are designated. This construction is developed from Fig. 1 which shows the same bucket in its six different positions. In the same way, in Fig. 3, we have six different positions for the jet on the stationary bucket.

It now becomes possible to form a simple arrangement to bring a nozzle into such position that the jet will have the same entrance conditions as in the relative motion. Such a testing arrangement was used for trial performance, and it was not difficult to make all necessary measurements.

The investigation of the flow on a stationary bucket seems to call for an arrangement where these various directions of the impinging jet will be secured; but it has been established that this sharp connection between jet and bucket is not necessary.

We know principally that the position of the impinging jet on the bucket is dependent upon the peripheral velocity, i.e., number of revolutions. With a change of the peripheral velocity, for instance, during speed regulations, or as a result of differences in pressure, a change of jet-impact conditions must take place. The flow should be regular without disturbances for a range of angles on the impinging jet; and therefore it is not advisable to make the investigation only for the computed main positions.

It is indeed practical to mark the main directions of the impinging jet on the bucket by painting. Nevertheless, the investigation will also embrace directions which are not far from these computed directions. In this way the investigation will be enlarged and will be made wider in scope than if merely fixed by the normal entrance conditions.

Cavitation and Bucket Efficiency

Now what can be proved by such an investigation on a "fixed" bucket? There are two principal factors which can be studied: The problem of cavitation and the problem of efficiency. Concerning the efficiency, we distinguish, as before, three phases of the flow, and consider first the discharge.

The best efficiency of a wheel will generally be reached if the value of the relative exit velocity is nearly identical with the value of peripheral velocity, or if the absolute exit velocity is perpendicular to the peripheral velocity. It is obvious that the absolute velocity has to be as small as possible, because the velocity head of the absolute exit velocity is a real inevitable loss, as previously mentioned.

To explain these relations, Fig. 4 shows all the relations at the discharge for a filament. We see in this figure that the absolute exit velocity consists of two components. The component perpendicular to the brim of the bucket is useful, inasmuch as this velocity carries the water away from the bucket. The second component which is directed parallel to the brim of the bucket and gives practically no help in discharging water is practically useless and should be eliminated completely, if possible.

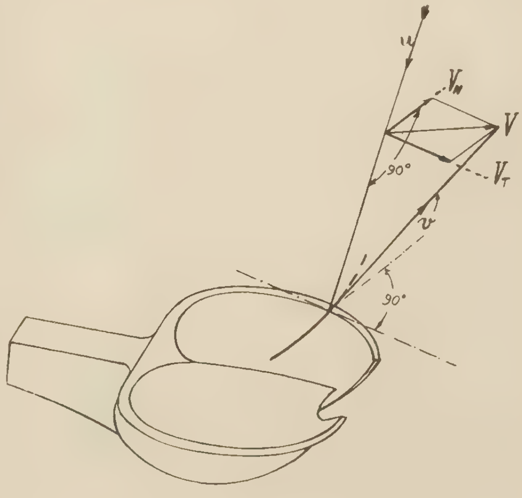

Fig. 4

We know that there is always a spreading of the jet on the bucket. Then the filaments on the border will always contain useless tangential components. Therefore a spreading with very inclined border lines should be avoided as far as possible. On the other hand, it should be borne in mind that the relative motion on the bucket is in a certain way different from the "absolute" motion on the fixed bucket. It is just this tangential component at the discharge which forms a principal difference between the two motions (Fig. 2).

Let us consider now the "absolute" flow on a fixed bucket, and imagine that we have found and tabulated the effective relations for each jet position. The tangential component of the exit velocity can be positive in some points, that is, directed outward, perhaps in some points negative (see Fig. 2). To arrive at a relative motion on the bucket, it is necessary to make the corrections with an additive positive tangential value; this means that points which show, in the absolute motion, an extreme negative tangential exit component will have an amelioration of the discharge on the rotating bucket. A flow on a fixed bucket, which shows a great positive tangential exit velocity, will suffer a deterioration in the case of the rotating bucket. In this case, a change of the bucket will be advisable, to reduce the tangential exit loss.

The examination of the discharge on the bucket is therefore reduced in finding the exact direction of the discharge velocity, especially with reference to the brim of the bucket. It is obvious that all positions of the jet under the corresponding impact angle should be investigated.

With these trials, it is necessary to determine that the jet coming from a bucket does not interfere with the preceding bucket.

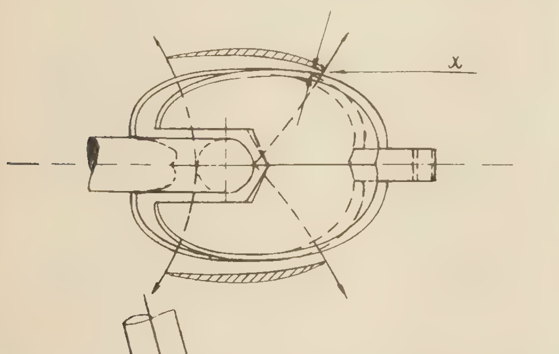

Fig. 5

There is a simple way to make all investigations in one by using a dummy bucket, Fig. 5. This dummy bucket has a far greater cutout than the normal bucket, so that the jet can be brought into the correct position with reference to the test bucket. In Fig. 5, the jet coming through the dummy bucket impinges on the test bucket, with nearly the same entrance conditions as on the rotating bucket. The discharge of the jet is, by this arrangement, clearly visible and easy to register.

In this way, it is possible to make exact measurements concerning the discharge of the jet. These measurements can be essentially simplified if two similar graduations are used on the test, and on the dummy bucket. It is also possible thereby to find the smallest clearance between jet and the preceding bucket, in this case the dummy bucket (see Fig. 5).

It is necessary to remark that a slight touching of the preceding bucket by the flow can be tolerated without sacrificing the efficiency, but such a condition can easily cause cavitation at high heads.

The inclination of the bucket relative to a radial line on the wheel should likewise be investigated. It is well known that there exists for the bucket a certain position with the highest efficiency. By changing this inclination slightly, there is almost no influence on the efficiency. On the stationary bucket, it will be found that by changing the inclination, a change in the discharge at the inner and outer part of the bucket brim will result. The discharge on the side of the brim remains practically unaltered.

Trials on a stationary bucket have proved, further, that the discharge from the bucket is not alone dependent upon the outlet form of the bucket, i.e., the brim itself, but also on the manner of the entrance relations. In this way, the splitter and the lips form a deciding factor for the discharge.

The investigation of the lip is now necessary in a twofold manner. We are interested in a correct form with reference to the discharge relations, that is, for the impinging branch of the jet. We are also interested in knowing what will happen to the deviated jet branch.

On Pelton buckets, there are now different forms of lips. The simplest form is that of an edge perpendicular to the splitter. Other forms have evolved by forming a bucket as a double ellipsoidal shell and by a cutout in the front of the bucket, the cutout being a surface of rotation with the center in the rotation axis. Another cutout is made with a cylindrical surface with its axis in the average path of the jet, and various other forms are also possible.

Thus quite diversified forms of lips may result, each form having certain advantages and disadvantages. The investigation of such lips gave such wide variety of results that it was practically impossible to classify them.

It was decided therefore to clarify the problem, and to study, first of all, the impact of the jet alone under simplified conditions. A series of tests was made to investigate the impact of a jet on specific plates under conditions similar to those arising in the bucket. This is practically equivalent to the arrangement of a lip with an edge perpendicular to the splitter.

Equipment and Test Procedure

The testing equipment for this purpose is shown in Fig. 6. Testing plates are brought into a circular jet at different positions, Figs. 7 to 12, inclusive.

A description of the procedure is given, first, with an attempt to classify the measurements. The results in connection with the bucket will be discussed later. The results which were obtained with two test plates are shown as A and B, Fig. 7; Figs. 8 and 9 show the jet impinging on the plate A. Plate A was selected from a number of others as the most favorable arrangement. The thickness of plate A is very small, being only 6 per cent of the jet diameter. Experience has shown the importance of a clear distinction between the front part δ_f and rear part δ_r of the edge angle δ. The values of these angles are designated in Figs. 7 and 8.

The deformation of the deviated jet shows surprising forms. In Fig. 8 we see the form of the jet in two cross sections by using an

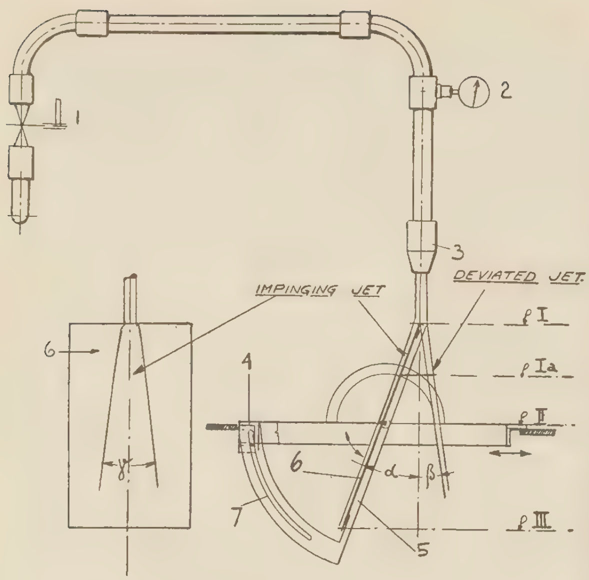

Fig. 6

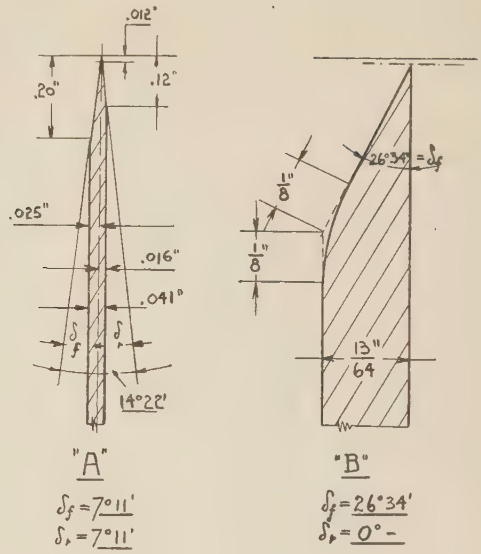

Fig. 7

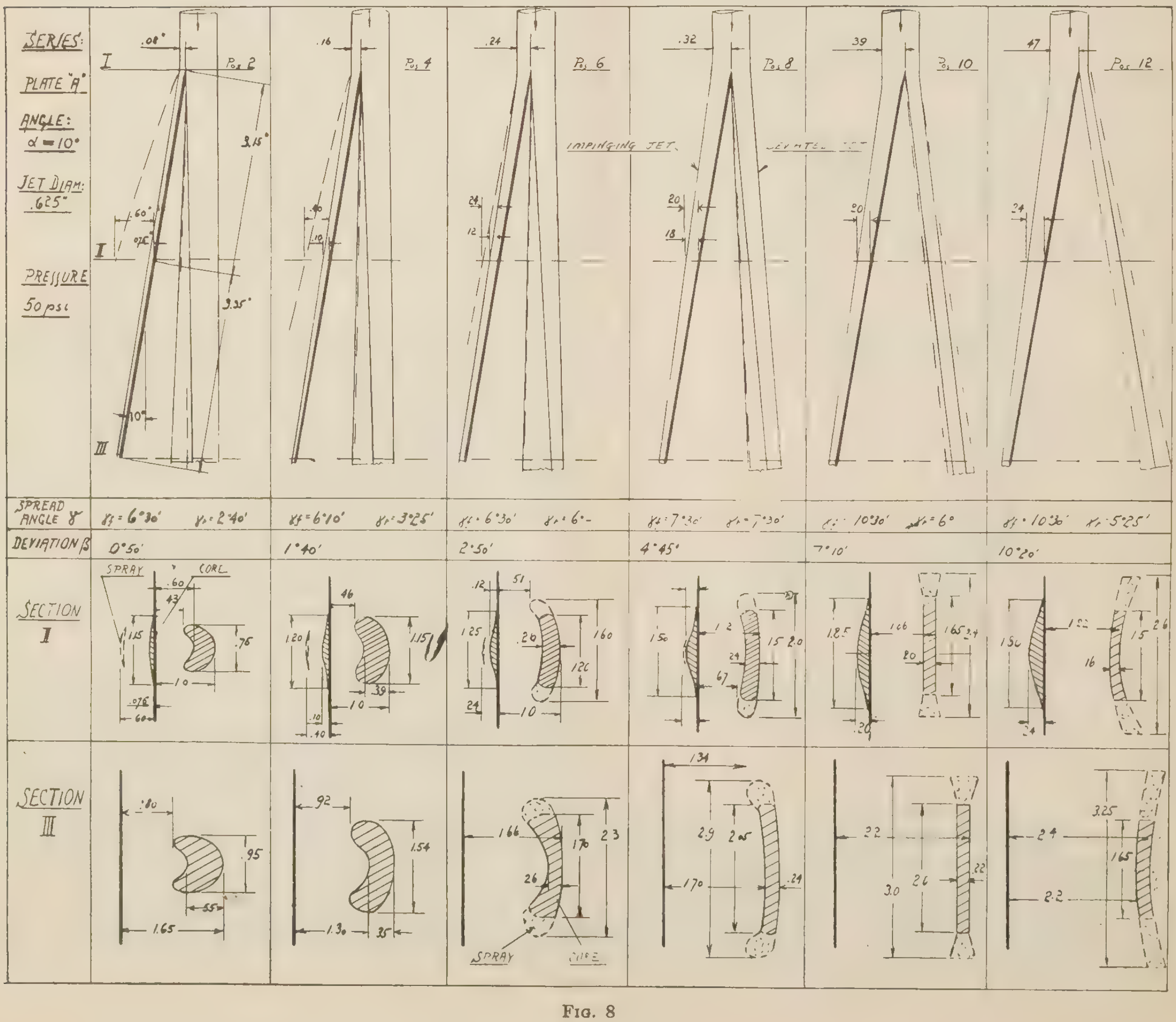

Fig. 8

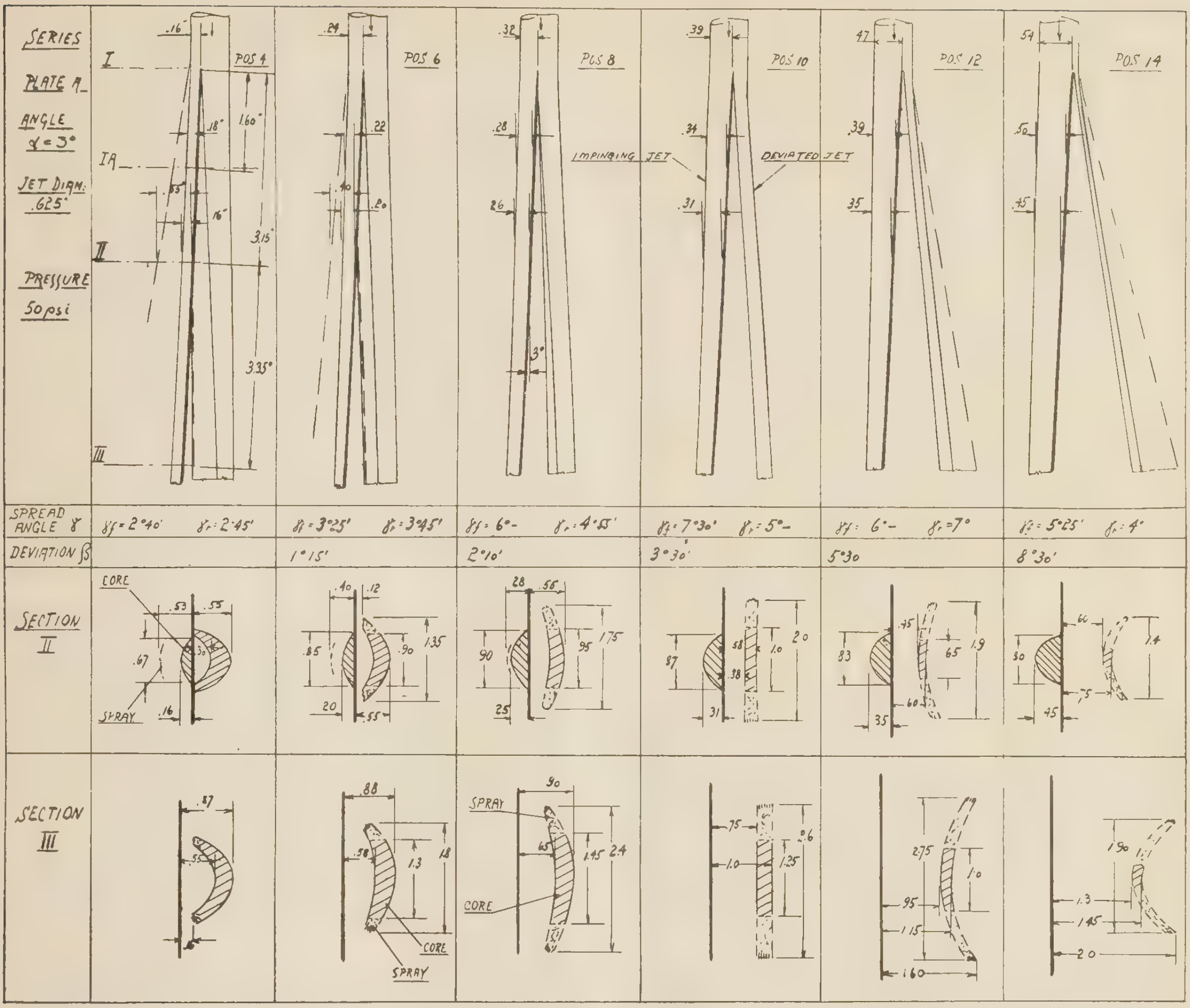

Fig. 9

impact angle of $\alpha = 10$ deg. The plate placed under this constant angle of 10 deg was shifted sidewise. The positions 2–12 in Fig. 8 correspond to the partition of the jet in its two branches, i.e., the impinging and the deviated jet. If the deviated jet is relatively large, (positions 2–6) from which it follows that the impinging jet branch is small, the cross section of the deviated jet is crescent-shaped, concave against the plate. With larger impinging jets, the crescent-shaped form changes to a flat rectangular cross section. With extremely small deviated jets, their cross sections will be curved in the direction opposite to the foregoing, i.e., convex against the plate. The deflection of the deviated jet will be greater with smaller jet. (Angle β is the angle of deviation.)

It should be mentioned that the cross section consists of two different parts; the core, which is practically filled with liquid, and the spray, which consists of numerous single separated water threads. The spray is developed extensively at a greater distance from the impact point, and, at certain positions of the deviated jet, the spray is at the point of impingement.

In addition to the deformation of the jet in the direction of deviation, there is also a spreading of the jet perpendicular to deviation. Both the impinging and the deviated jet spread on the plate, particularly with smaller thickness. (γ is the angle of

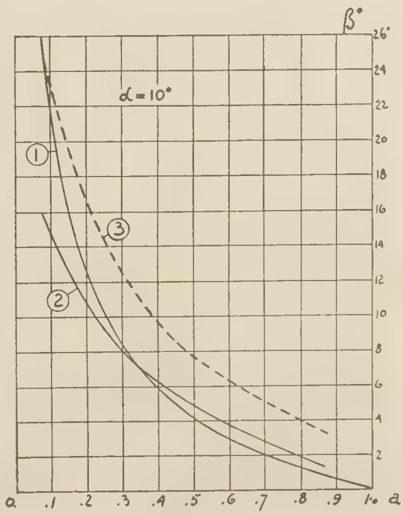

Fig. 10

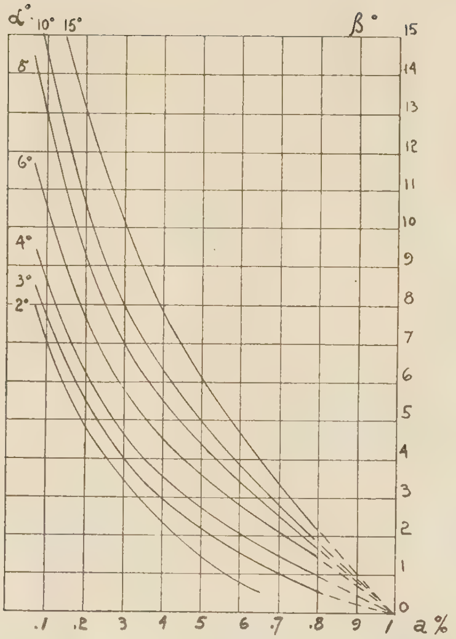

FIG. 11

spreading.) The angle of spreading of the impinging jet (γ_f), and the angle of spreading of the deviated jet (γ_r), should be clearly distinguished.

The plate was investigated at numerous impact angles. Fig. 9, for example, gives the appearances for an angle of 3 deg. The principal difference in these cases (angle 10 deg Fig. 8, and 3 deg Fig. 9) is that the curvatures of the cross sections are far more pronounced at 3 deg than at 10 deg. The cross section of a large jet deflection at 3 deg impact angle shows a cavity. The side ends of the deviated branch adhere to the plate; and under such conditions, cavitation is practically inevitable.

Figs. 8 and 9 indicate the angle of deviation β, i.e., the angle of the deviated jet, in relation to its initial direction. This angle can be calculated theoretically with relation to the reaction, and the following equation gives the value

$$a = \frac{1 - \cos \alpha}{1 - \cos (\alpha + \beta)} \qquad [4]$$

In this equation α is the impact angle; β the angle of deviation, and a is the per cent area of jet deflection. (The complete jet has 100 per cent area and the impinging branch, the area $1 - a$ in per cent.)

Fig. 10 indicates the dependency of the deviation β as a function of jet deflection a for 10 deg impact angle. Line 1 gives the theoretical calculations according to Equation [4], and line 2 gives the measurements with plate A, according to Fig. 8. The values are not completely identical, but it is impracticable to discuss the differences here.

RESULTS OF TESTS

In general, it has been established that the connection between deflected and impinging jets is uniform. Fig. 11 gives a complete diagram for plate A which shows all the deviations from 2 deg to 15 deg impact angles for all impinging jet areas.

As previously mentioned, plate A was selected from a variety of test plates. For example, in order to indicate the effects on another plate, Fig. 12 shows the conditions with jet impinging on plate B (Fig. 7). This was one of the first plates examined and has been developed from the lip forms used ordinarily.

In relation to the difference between plates (see enlarged sections in Fig. 7), the following comments are in order: While plate B is extremely thick in comparison with plate A, this alone does not constitute the essential difference. The edge angle on plate A is far smaller than on plate B. Moreover, the front edge angle δ_f on plate A is identical with the rear edge angle δ_r, whereas plate B has no rear edge angle at all.

By comparing the figures for plate B with those for plate A, it becomes evident that the results obtained with plate A are superior to those in the case of plate B. Both the impinging and deviated jets at plate B show disturbances which are not feasible under any conditions. To clarify these relations, the deviation for plate B under 10 deg impact angle is also noted in Fig. 8.

The principal differences in the impact conditions between plates A and B can best be defined as follows: Small impinging jets on plate B are not feasible, as the impinging jet will be deflected and sprayed.

With the same impact angle, the deviation for plate B is far greater than for plate A. The spreading of the jet is less favorable on plate B and is more developed.

A plate thinner than plate B, but with the same characteristic form, produces results analogous to plate B, with other limiting factors.

At this point, the influence of these investigations on the flow on a Pelton bucket should be discussed. We conceive from Figs. 8 to 12, inclusive, that the jet upon encountering a bucket has a tendency to deflect, this being forced to take another path, as presumed by the simple theory with the single water filament.

It is obvious that the relative paths a', b', c', in Fig. 1, are correct only in so far as they represent that portion of the jet which has not been deflected. The movement of that part of the jet which will be deflected is extremely complicated owing to certain cross sections being influenced by various deflections.

To make these relations as clear as possible, an attempt was made to develop a kinetographic representation. This has been accomplished in Fig. 13, by showing the wheel in a series of successive positions.

To simplify the representation, certain abstractions and suppositions were made as follows: The wheel should have 20 buckets (center angle 18 deg) similar to the case of Fig. 1 (buckets M, N, O). The lip of the bucket should have an edge, according to plate A, perpendicular to the splitter, so that the liquid will be caught up directly in the plane perpendicular to the plane of reproduction. The jet should be divided unequally (for practical reasons) into single segments. Their origin is as follows: The wheel will be presented in successive positions 10–20 (to 2 deg apart), affording uniform revolution.

The cross section 11 of the jet is determined by the end point of the bucket N in position 11 of the wheel. If bucket N arrives at position 12, the cross section 11 of the jet arrives simultaneously at the position indicated. In position 12, the end point of the bucket N determines the position of the cross section 12 in the jet, and so on.

This arrangement gives no uniform partition for the jet, but simplifies the representation. It is obvious that bucket O arrives at the location of bucket N in the successive ninth position. Positions 11 and 19 are identical.

The inclination of the lip was chosen of such a form that in the extreme position the relative velocity is exactly the direction of the lip entrance. It is assumed that if the lip enters into the jet, there will be a tendency toward deflection only for the critical cross section of the jet. The movement is now unsteady and it is therefore impossible to compute the exact influence on particles which are not in immediate proximity to the lip. A further sim-

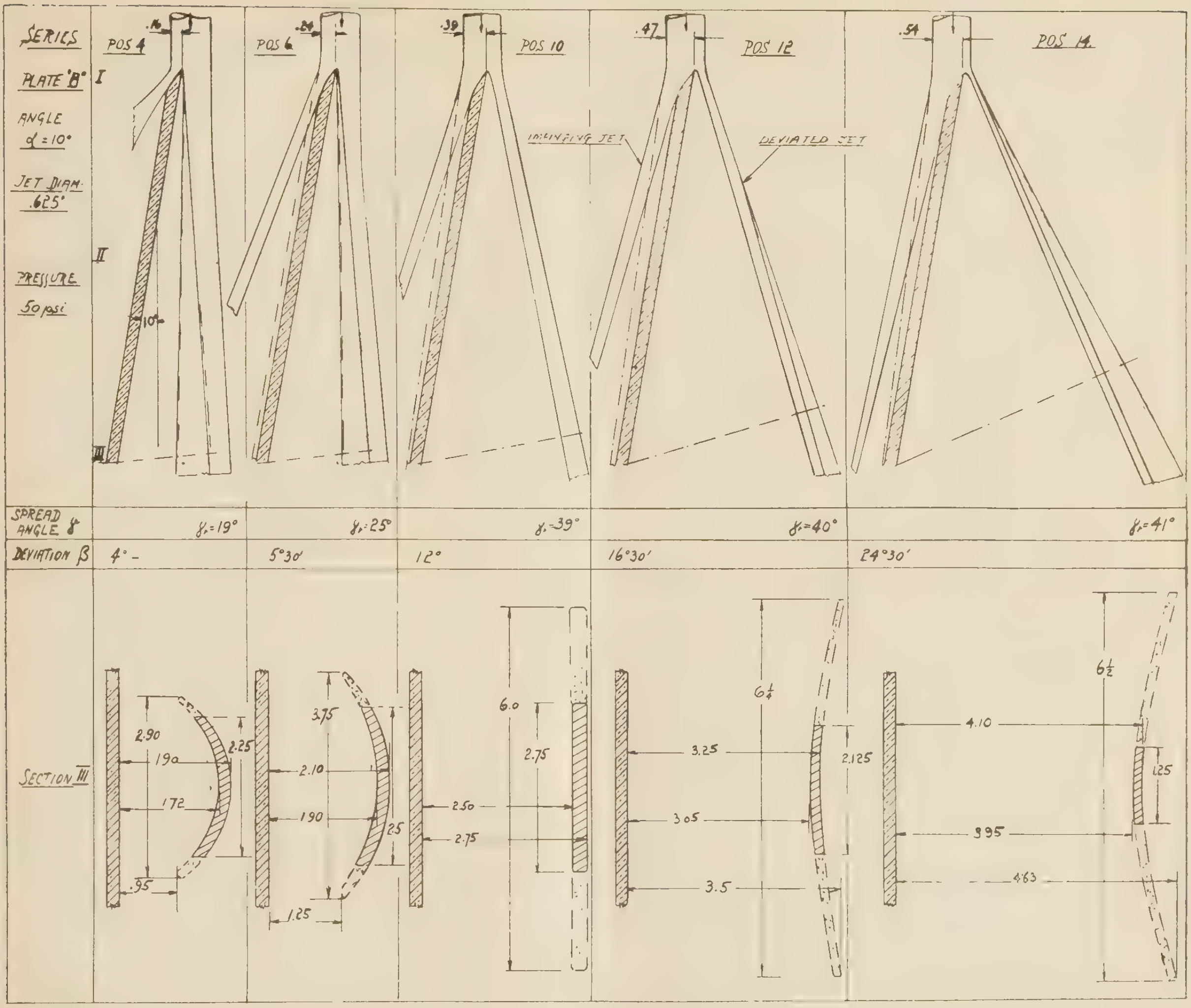

FIG. 12

plification can be made by disregarding the appearances of the width. We can therefore abstract the spreading and change of the cross section of the deviated jet. We assume further that the angle of deviation β in the relative system can be transferred unchanged in the absolute system. Therefore we co-ordinate to the influenced cross sections a deviation of the direction of the absolute speed according to the diagram, Fig. 11.

The affected cross sections of the jet are therefore not moving in the original direction; on the contrary, they will have individually different directions.

The kinetographic representation of the jet parts which form the admission for bucket M was made in this manner.

Fig. 13 represents a succession of positions of the three buckets M, N, O, and the relative position of the jet segments. Each second segment of the jet is hatched in the direction of motion. This figure represents clearly momentary positions, and it will be seen how a part of the jet between the cross sections 14–19 will be deflected so that it will not reach bucket M.

To find the loss, the volume of the part which does not reach the bucket is computed and divided by the volume of the jet per pitch length. The loss was found to be 5.8 per cent, which is excessive. An improvement can be effected in part by a greater number of buckets, and partly by increasing the wheel diameter.

These considerations have been abstracted from the form of the deflected jet. It has been established, Figs. 9 and 10, that there is a spread resulting in the form of the deviated jet being more or less rectangular, the longer side oblique to the splitter. It should be borne in mind that the deflected jet will be caught by the following bucket so that it will now be an impinging jet. The impact of such a rectangular jet on a bucket is certainly not favorable. The outer water filaments are likely to splash directly on the bottom of the bucket, guidance through the splitter thus being lost.

It is impossible to effect any material improvement of these particular conditions by changing the splitter. The lip, on the other hand, should be modified to avoid the appearance of such deviated and compressed jets. This is, in fact, the reason for the existence of special lip forms.

Relative to the impinging jet, it can be established that there is proper guidance only if the edge angle of the lip is very small. Sizable edge angles are inclined to produce a complete spray of the impinging jet. It has been demonstrated, furthermore, that the spread of the impinging jet is quite small if the impact angle is likewise small. It would be advantageous to arrange a bucket

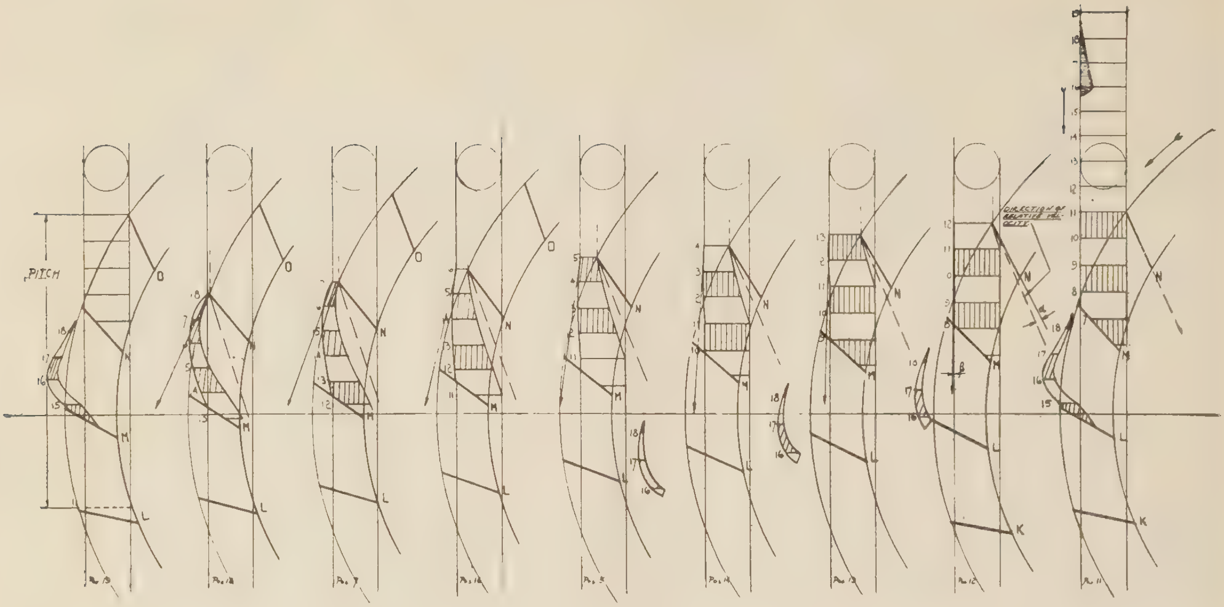

Fig. 13

so that small impact angles would be produced, but there are two reasons why this should not be done, as follows:

1 A very small impact angle at a given peripheral velocity results, at a greater peripheral velocity, in a jet impingement on the underside of the lip. This should be avoided because it decreases the efficiency.

2 Cavitation will inevitably be produced with smaller impact angles at higher heads.

On the other hand, a small edge angle on the lip is difficult to maintain, because such an edge will not stand up under high heads. Further, an uneven edge invariably causes cavitation; and it is therefore necessary to make the wall of the lip adequately thick.

It is obvious that the straight lip form does not give very favorable results (see Fig. 13). There are, however, various methods, forming other lips without the disadvantages just outlined.

As mentioned previously, the variety of the lips is so far-reaching that it is impossible to find a combination which can be regarded as best in every respect.

It develops that the method with the stationary bucket makes possible the study of all questions and problems connected with the form of the lip.

It has been said that the discharge on the Pelton bucket is influenced to a great extent by the entrance conditions, that is, by the form of the lip and of the splitter. We know now in what sense the lip is working, and that through the lip edge a spread of the impinging jet results.

The splitter also causes a spreading of the jet, and it can be stated generally that the spreading will be increased with greater splitter angle. Both edges, the splitter and the lip edge, have therefore a combined influence on the impinging jet. The form of the bucket should be a natural transition without sudden rupture in the curvature of the surface.

We assume that losses at the entrance are produced chiefly with the deflected jet, while losses at the entrance of the directly impinging jet are very rare, with suitable arrangement provided.

The losses of the deflected jet consist in not reaching the following bucket and in an impinging of a compressed jet on the adjacent bucket. By this means, the influence upon efficiency,

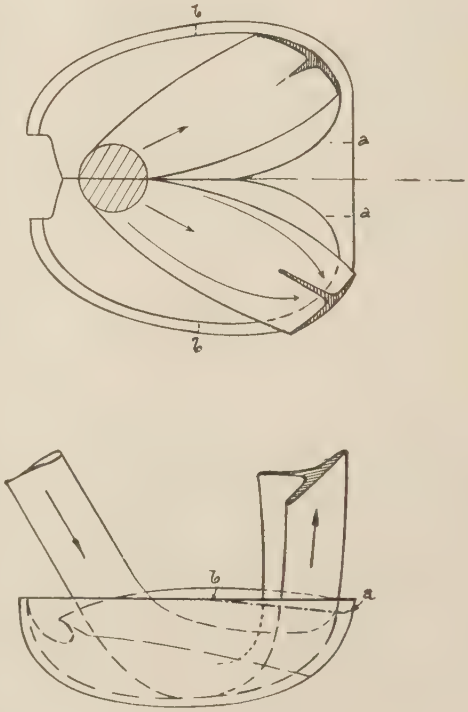

Fig. 14

both at the entrance and the discharge, can be established, resulting in considerable reduction of losses, if not completely eliminating them by proper arrangement. The final consideration is the possibility of avoidable losses during flow on the bucket itself.

It seems improbable that such losses could exist; however, with the method described, a hydraulic loss was found which caused further efficiency decrease. A runner which was built for a plant

according to a previous model did not show the efficiency expected. The bucket had smooth curvatures and had only slight differences in comparison with the model, which should have increased the efficiency.

Upon testing the bucket in the fixed position, this bucket showed a singular form of discharge. The cross section of the discharge was not flat or a more or less rectangular strip. On the contrary, it had a T-shaped form, Fig. 14.

It was established that this T-shaped cross section was produced by two streams, both originating on the splitter and working against each other. This was a steady appearance on the bucket, starting from a certain impact angle.

It was obvious that under such circumstances the discharging water came into strong interference with the preceding bucket. In this case it was possible to improve the efficiency largely by altering the discharge edge of the bucket alone. The bucket part on the rim between the points a and b (Fig. 14) was cut away as marked with dot-and-dash lines. However, a better correction would have involved a change in the curvature of the bucket.

Further investigations proved that such T-shaped discharge is not uncommon. In this case, the T-shaped form was a steady one, in another case it was found that such T-shaped form was produced by the continued succession of jet parts impinging under different angles.

These conditions can be studied easily in the case of the fixed bucket. It is doubtful whether it would have been possible to see them on a revolving wheel.

Upon becoming acquainted with the facts and knowing where to give our attention, it will be possible, with proper arrangements, to see these conditions on a revolving wheel with stroboscopic instruments.

Finally, there should be mentioned, among the avoidable losses, a disturbance which arises from the outflow, but which has its real cause in the enclosing housing. Sometimes the discharging water impinges against the walls of the housing or elsewhere and is deflected, after which it returns to the wheel, and in this way the power of the turbine is reduced.

In certain cases, the discharging water will be taken by the air rotating with the wheel, thus producing similar disturbances.

These losses cannot be traced back to the flow on the turbine; but they have their cause in the specific construction of the turbine itself, i.e., housing, air passages, baffle-plate arrangement, etc.

To avoid such conditions only one rule has to be observed. The outflow of the water from the bucket should be completely free of foreign matter which would interfere with the outflow; and it is advantageous to make arrangements whereby the outflow tendency can be increased.

Having established the relative outflow conditions on a bucket, it is possible to estimate the absolute flow and to decide whether interference is to be expected, and if such conditions exist, how to deal with them.

TABLE 1 ANALYSIS OF PELTON WHEEL LOSSES

1 Nozzle:
  Friction in nozzle and on needle
2 Wheel:
  (a) *Mechanical losses:*
    1 Friction in bearings
    2 Air resistance of turning wheel
  (b) *Hydraulic losses:*
    (a) Unavoidable losses
      1 Entrance; mixing loss
      2 On bucket; friction of flow on bucket
      3 Discharge; normal discharge loss
    (b) Avoidable losses:
      1 Entrance; deviation of jet on lip with outside pressing of jet parts. Spray-formation compression of jet parts with unfavorable impact on adjacent bucket
      2 On bucket; crossing of two streams; disturbances
      3 Discharge; tangential discharge; interference with preceding bucket; deflection of discharged jet on wheel housing and disturbances resulting therefrom.

With the method of the fixed bucket, the author has been enabled to study the flow on the bucket and to analyze the losses which determine the efficiency of the wheel. It is natural that this method alone is not sufficient to determine the efficiency which can be established only by testing the performance of the wheel. The losses on the bucket having been established, it is practical to summarize them for further investigation, as in Table 1.

Discussion

S. Logan Kerr.[2] This paper is of particular interest to the writer, as it fell to his lot to make a research investigation of some unhappy results of impulse-wheel performances when he was still in the hydraulic-turbine industry. Now after an absence of some 7 years, keeping meanwhile the position of "sitting on the lid of the Hydraulic Turbine Test Code Committee," the perspective attendant to a detached view of the "art" may be helpful as a contribution to the discussion of this paper.

Some 15 or 16 years ago, the writer had the privilege of visiting the plants of the principal manufacturers of turbines in Europe. They were most co-operative and put at his disposal much information which showed their superior performance, individually and collectively, with special attention to their improved performance in relation to the state of the art in America.

It was also his privilege to have entreé into the plants of the users of these superlative turbine units. Much "sales talk" was deflated thereby. But some things defined the debunking process; one in particular was the efficiency of impulse wheels.

Such "impossible" arrangements as a vertical shaft unit with a test efficiency of 90 per cent (with a penalty clause in the contract and a certified copy of the test results), and horizontal units of medium and large size that attained efficiencies of 88 to 91 per cent, seemed to be greatly at variance with reported "ceiling efficiencies" of 85 to 87 per cent in American practice, to say nothing of the horrible examples that slipped into the middle "seventies."

What was the answer to these major differences in performance?

There were many answers to that question given in writing, in polite conversation, and sometimes in conversation that was not so polite. Perhaps this view in retrospect will serve to bring the discussion of this controversial question down to a sound basis of an open-minded consideration of engineering and technical aspects and leave out completely the side issues of personal opinions and commercial issues, tradition, and other obscuring factors. Let us examine some fundamentals:

1 What is the efficiency of an impulse turbine? Is it not measured on the same basis as all other hydraulic prime movers as set forth in the A.S.M.E. Power Test Code for such equipment? Or should there be some other basis? The writer's committee[3] is vitally interested in knowing.

2 What elements determine the performance of an impulse turbine? Here we find an essential difference between the reaction turbine and the impulse turbine. In the reaction turbine, we have a single combined unit which permits various deviations in the individual elements that affect each other but can be compensated for in the design basis used.

In the impulse turbine, however, we find two units, the nozzle, and the bucket in series. The jet discharged by the nozzle must be solid and free from dispersion and distortion or else the most perfect bucket in the world cannot produce acceptable efficiencies. If the most perfect jet is impinged on a bucket not suited to it, the bucket cannot develop its best performance.

[2] United Engineers & Constructors, Inc., Philadelphia, Pa. Mem. A.S.M.E.

[3] Committee 18, Hydraulic Prime Movers, A.S.M.E.

We are then faced with two requirements: An efficient nozzle, delivering a solid compact jet to the bucket; and an efficient bucket capable of receiving it and converting this energy to the best possible degree.

4 The author gives a careful analysis and discussion of the elements of loss in the buckets upon receiving a good homogeneous jet of water from the nozzle. The writer recommends this paper; it merits careful study. It is the work of a distinguished engineer and scientist. Think well before taking issue, and then take issue only when full and accurate information to refute this exposition in part or in whole has been developed.

5 Make sure that the nozzle is free from rapid accelerations followed by retardations of flow; be sure that abnormally high velocities are eliminated from the approach passages of the nozzle; be sure that abrupt and unnecessary changes in direction are avoided in the nozzle and the approach to the nozzle; in brief, be sure that the most efficient bucket that can be designed has a good, solid, high-efficiency jet impinging upon it and can start its own work with a chance to contribute without impairment to the maximum over-all efficiency of the unit as a whole.

6 When these objectives have been attained, then test the scale model under conditions where there is sufficient turbulence to simulate accurately the behavior of the jet under full-scale velocity conditions of the design head, not the available test head for the model.

7 When these conditions have been thoroughly studied and complied with, then many of the mysterious discrepancies between the impulse-wheel laboratory work and the field performance will disappear, just as they did many years ago on the reaction turbine when the basic fundamentals of the relation between the model and full-sized-unit performance were reduced to an academic problem of slide-rule computation.

The impulse turbine has still to overcome the obstacles of insufficient relationship between experimental laboratory and full field performance, both as to the degree of efficiency attained at the maximum point and the ability to reproduce in the field the performance attained in the laboratory as to shape of curve and as to many other characteristics.

L. F. Moody.[4] It will be generally agreed that the Pelton wheel, while it has been developed into a thoroughly reliable and satisfactory prime mover, still has ample possibilities for further improvement. To hold that it has reached its ultimate state and that the room for betterment is too narrow for profitable investigation would represent a defeatist attitude not justified by analysis. Gains in efficiency of the order of some 5 per cent or more are easily conceivable. and when translated into values of generated power would justify intensive research.

As pointed out by the author, Pelton-wheel efficiencies have lagged considerably below those of reaction turbines; and it should be remembered in making this comparison that the installation efficiency of a Pelton wheel, which is the figure to be compared with the reaction-turbine performance, should include the head lost in the free fall from the buckets to the tail water, an element which is included in the reaction-turbine figures. Future development may show the practicability of regaining this loss, in spite of mechanical objections such as the closed outlet with shaft seals, ejector means for taking care of air accumulations, and float control of the water level.

The listing (Table 1 of the paper) of hydraulic factors involved in the Pelton-wheel action emphasizes the complexity of the flow in the bucket and the unattractiveness of attempting a completely theoretical analysis. In the hydraulic field, we have been long converted to the experimental method in attacking such problems. A mere comparison of over-all results with progressive changes in design is, however, a slow process, and a method of isolating parts of the problem and attacking them separately is greatly to be desired. The author's method of research should be of real value in giving us a deeper insight into the nature of the flow in the Pelton bucket.

[4] Professor of Hydraulic Engineering, Princeton University, Princeton, N. J. Mem. A.S.M.E.

Probably many have thought of using stationary buckets in experimental research on these wheels, or have actually made such experiments. The writer must admit, however, that before reading this paper he was inclined to think that conditions in the running wheel would be so different from those in the experiment that the results would be of doubtful value. The author shows convincingly that the differences are actually of a low order of magnitude; and that by the use of a dummy bucket, cut away to permit the jet to enter the preceding bucket at the correct relative direction, a close approximation to actual conditions can be secured. The author has thus introduced a new technique which should be of marked value in this field.

Without some method of this kind, it is difficult to know where to start in trying to improve bucket design; with the use of this method many unknown phenomena in the flow can be brought to light, the flow conditions can be determined at least qualitatively and directions of improvement shown.

The author has mentioned that not only can sources of loss of efficiency be detected, but that cavitation tendencies can also be discovered in some cases. One possible source of cavitation not included in the paper is the presence of a vortex core in the jet, which upon impinging on the bucket would produce all the essentials of cavitation.

The writer takes the opportunity to urge that more study be directed to Pelton-wheel cavitation and that statistical data be collected, as was done by the National Electric Light Association some years ago for reaction turbines, to show the general trend of the tendency toward pitting as a function of specific speed. With such data, we could plot curves of the Thoma cavitation factor sigma with respect to specific speed. For Pelton wheels having free discharge into the atmosphere, the static draft head term H_s disappears from the formula $\sigma = \dfrac{H_b - H_s}{H}$; giving simply $\sigma = H_b/H$, the ratio of the height of the barometric water column, at the elevation of the plant, to the effective head. With definite experience data plotted in this way we could more intelligently judge of the range of heads and speeds limiting the Pelton-wheel field.

Author's Closure

The author greatly appreciates the generous comments from such well-qualified sources as Mr. S. Logan Kerr and Prof. L. F. Moody.

Mr. Kerr states concisely the general principles which are absolutely necessary to obtain high efficiencies in Pelton wheels. Lower efficiencies in Pelton wheels can be attained very easily. One proof is that the old impulse wheel was built even a century ago in a primitive wooden form, sometimes with 75 per cent efficiency. This circumstance might be the reason for a certain superficial similarity between test and field arrangement. This neglect of nozzle arrangement and housing formation explains discrepancies between test and field performance.

The author agrees entirely with the remarks of Professor Moody, especially need of statistical survey concerning the cases of cavitation of Pelton wheels. It is obvious that cavitation is unavoidable in all kinds of turbines under certain conditions at a relatively high speed which is desired for economical reasons, and we urgently need to establish these limitations in order that cavitation may be avoided.

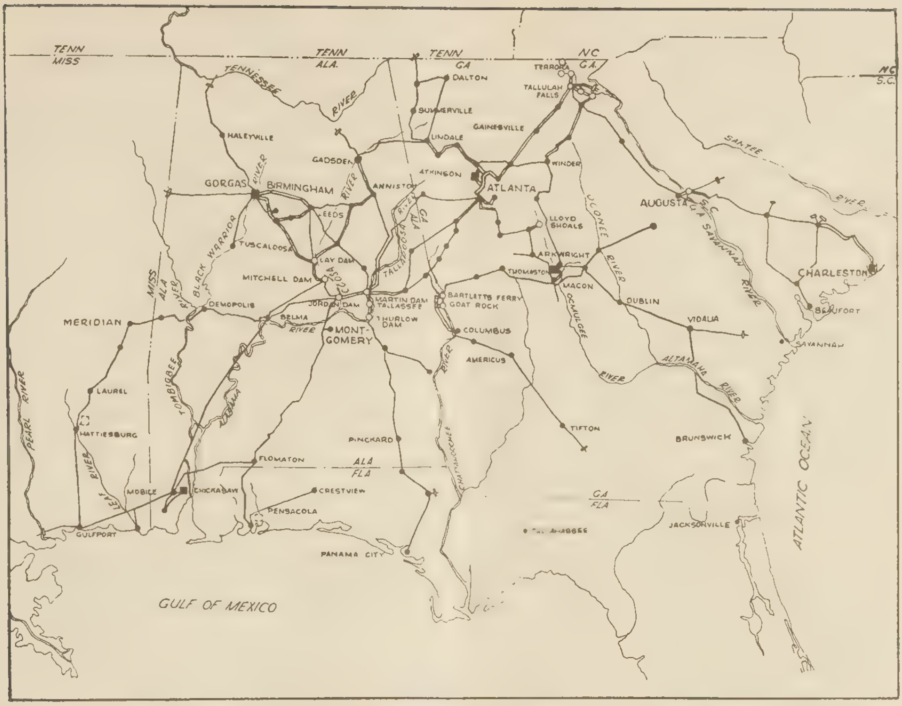

Fig. 1 Extent of Territory Served by Southeastern Group of the Commonwealth & Southern Corporation

Range of Operation of Steam Plants in a Combined System of Steam and Hydro

By A. T. HUTCHINS[1] and HOWARD DURYEA[2]

The balance between hydro- and steam-plant power generation on the southeastern group of The Commonwealth & Southern Corporation during the postdepression period and later the period of peak war demands is explained in this paper for an understanding of the system's approach to the installation of steam plants. The discussion includes such data as area of the system, loads, plant capacities, and such pertinent factors as ratio of steam to hydro generation, the relation of steam to hydro capacity, etc. The unit principle is used in the steam-plant installations, i.e., one boiler supplies all steam requirements for one turbine unit. Details of the boilers, auxiliary equipment, operating experience, and results attained are comprehensively treated.

The problem of fitting large and efficient steam units into utility loads during the depression years from 1930 through 1935 on the larger systems of the country has been much discussed. On the southeastern group system of The Commonwealth & Southern Corporation many of the steam plants were idle for long periods, and this was accentuated by the existence of hydro capacities that were relatively large in proportion to the total load. Seasonal changes in river flows necessitated wide variations in steam-plant loading to supplement hydro generation when flows were low and to utilize hydro generation and avoid wasting water during high flows. Postdepression and particularly the present high war loads are in sharp contrast to the foregoing conditions, as they require maximum continuous output from the more efficient steam plants and greatly increased generation from higher-cost plants.

Extent of Southeastern Group of System

The discussion of such operations is obviously more interesting if there is a general understanding of a system, its area, loads, and plant capacities, together with other pertinent factors, such as ratio of steam to hydro generation, the relation of steam to hydro capacity, etc. The small map , Fig. 1, has been prepared to show the extent of the territory served by the southeastern group of The Commonwealth & Southern Corporation. Because of lack of space it does not show all existing steam and hydroelectric plants, nor all of the transmission lines and substations.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

The data presented in this paper refer to the system shown in Fig. 1, corrections having been made for periods prior to 1940, when Tennessee Electric Power Company was included in this group. The present system includes South Carolina Power Company, Georgia Power Company, Alabama Power Company, Gulf Power Company, and Mississippi Power Company, listed in order from east to west. Again moving from east to west, the rivers supplying water to hydroplants are the Savannah, between Georgia and South Carolina; the Tallulah and Chattooga, whose confluence forms the Tugalo in northeast Georgia; the Oconee and Ocmulgee in north central Georgia; the Flint and Chattahoochee in southwest Georgia; and the Tallapoosa and Coosa in Alabama. Located as these rivers are in the southern foothills of the Appalachian range and above the fall line, they afford advantageous sites for hydro plant developments, starting with elevations of 1846 ft above sea level at the Burton site on the Tallulah River in northeast Georgia, with progressively lower elevations as one moves southwest.

Fig. 2 shows total load, expressed in million kilowatthours

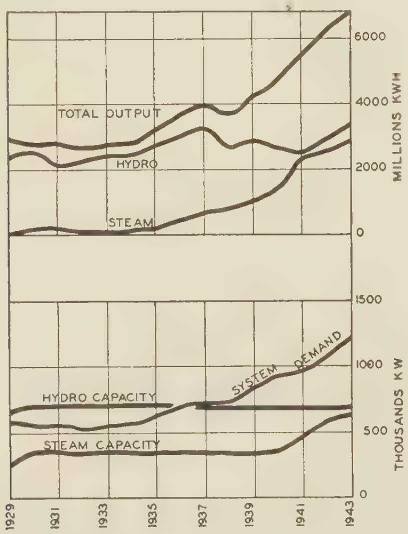

Fig. 2 Total Annual Load on System, Relation of Hydro and Steam Generation, Peak Hour Maximum Demands, and Installed Capacity of Steam and Hydro Power

annually, together with hydro and steam generation, and also peak hour maximum demands expressed in kilowatts, together with installed capacity of hydro and steam sources. Purchased energy is not shown because of lack of space and consequent confusion. The system has purchased large amounts of energy from outside sources, and the co-ordinated companies have purchased by-product generation from industrial concerns. For a few years one of them operated a plant to supply steam to an industrial customer, and thus secured a block of by-product energy. Purchases are based on requirements to supplement steam and hydro sources, availability, and relative cost.

In discussing the graphs, Fig. 2, depression loads are apparent from 1929 through 1934, followed by moderate increases in 1937. Loads near the close of 1937 and 1938 decreased, following which they again increased and assumed the present sharp upward slope characteristic of war production.

The lower section of Fig. 2 shows that there have been no increases in installed hydro capacity since 1930, and that steam capacity also remained practically constant from 1930 through 1940. This condition reflects the depression loads and the eventual loading of both hydro and steam plants until the war loads approached the existing capacity, about 1939, and resulted in the construction of additional steam facilities.

Fig. 1 shows the location of six hydro plants in extreme northeast Georgia; three on the Chattahoochee River above Columbus, Georgia, three on the Tallapoosa River, and three on the Coosa River in Alabama. In each of these groups the plants are in series so that water released from the upper plants is successively used by those below.

The total capacity of run-of-river plants is around 375,000 kw and that of the two principal storage groups is 350,000 kw, making the total installed hydro capacity some 725,000 kw. However, the annual generation is roughly two thirds from run-of-river plants and one third from storage plants. The total equivalent energy in storage is 450,000,000 kwhr, with some additional 15,000,000 kwhr in run-of-river ponds.

Referring to Fig. 2, curve "Steam," it will be noted that the output from steam plants was very low for the first 10 years shown. In fact, the composite plant factor for the steam capacity, as shown in the lower part of this figure, is approximately 10 per cent for this period. As previously mentioned, a load drop occurred in 1938, following which the sharp uptrend beginning in 1939, started a new era of steam-plant construction and operation in marked contrast to the previous periods. In 1939, the first unit of those shown in Table 1 was purchased and its initial operation, together with that of other added units, is given.

TABLE 1 STEAM PLANTS ADDED TO SYSTEM

Plant	Unit no.	Kilowatt rating	Throttle pressure, psi	Steam temperature, deg F	Revolutions, rpm	Generator cooling	Initial operation
Arkwright	1	40000	850	900	3600	Hydrogen	April, 1941
	2	40000	850	900	3600	Hydrogen	May, 1942
	3	40000	850	900	3600	Hydrogen	Nov., 1943
Chickasaw	1	40000	850	900	3600	Hydrogen	April, 1941
	2	40000	850	900	3600	Hydrogen	Mar., 1943
Atkinson	2	60000	425	750	1800	Air	Sept., 1941
Gorgas	5	60000	850	900	3600	Hydrogen	Dec., 1943

Although generating and transmission facilities, as well as distribution systems, are owned by the operating companies in the several states, planning studies for additions, design engineering, and subsequent direction of operations of power sources are co-ordinated by a service company mutually owned by the operating companies. The personnel of the service company for the southeastern group is located in Birmingham. By means of the co-ordination effected by the service company, the ultimate aim that power-producing facilities be most advantageously located, designed, maintained, and operated so as to produce the maximum amount of energy at lowest cost, and that each of the operating companies shall benefit therefrom is attained. In other words, all power-producing facilities are operated as a single group to provide capacity and energy for the total load. This method of operation takes advantage of all diversities of loads, stream flows, unit outages for inspection, maintenance, and emergencies that may occur between the individual operating companies' systems, as well as within the systems of each of them. Any action that materially reduces the area served and the facilities available to produce power would tend to lessen the economic value of the group.

System Ranks High in Performance

An indication that the size of this system tends toward economy in operation is its performance in comparison with other systems east of the Mississippi River. Of a load of 89 billion kwhr east of the Mississippi River in 1940, 44 per cent was generated by seven large generating and transmission systems that have been developed and operated under common ownership similar to the southern group of The Commonwealth & Southern Corporation. These seven systems generated an average of 4430 kwhr per kw

of dependable generating-plant capacity. This system stood third among these seven groups with a generation of 4413 kwhr per kw of dependable generating capacity.

The remaining load east of the Mississippi River was developed by a large number of smaller companies and systems. For the same year, these companies generated 3590 kwhr per kw of dependable generating-plant capacity. This is a rather striking comparison when it is realized that in this respect the seven systems under common ownership showed an increase in use factor of 23 per cent, as compared with the other systems.

Three factors undoubtedly contribute to these results:

(a) Diversity of peaks reserves the surplus capacity.

(b) Better planning, due to engineering staffs made practicable with size of operations.

(c) Greater use of interchange agreements where dispatching is handled by one co-ordinated organization.

The operation of all generating plants and the interchange of energy between companies within the group and with adjoining concerns is under the supervision of a central supervisory group of the service company, who work with the load-dispatching forces of the individual companies. Hourly load estimates for the entire group are maintained constantly and are projected into the future to provide hourly load schedules for hydro groups and steam units during critical daily and week-end periods. Arrangements are made to load and unload steam units so as to utilize, supplement, or conserve hydro flows; provide for outages for inspection, maintenance, and emergencies of steam and hydro units; insure sufficient free hydro-regulating capacity for tie-line control, spinning reserves, and minimum flows when and where required. Arrangements are made to receive energy from outside sources, to make deliveries to them, and to synchronize these transactions with hydro and steam operations on the system.

Unit Principle Used in Steam Plants

Since the short time peaks of load throughout the year are carried by the hydro plants, the units installed, as shown in Table 1, were all designed on the basis of carrying prescheduled loads. The extensive transmission system, interconnecting the load areas with their steam plants, makes conditions ideal for use of the unit principle of design, in which one boiler supplies all the steam requirements for one turbine. One exception among the units shown in Table 1 is unit No. 2 at Plant Atkinson, where there was some capacity available in existing boilers. Because of the importance of the load in the local area a cross connection was made between the steam lines of the two units at the Chickasaw Plant. It may be noted here that at the Gorgas Plant the steam conditions for No. 4 unit are 425 psi and 750 deg and No. 5 unit 850 psi and 900 deg, and no pressure-reducing and desuperheating equipment was installed.

The early plants in the 900-psi class were designed for 900 psi in the boiler drum and 850 psi at the turbine throttle. The design of superheaters to secure 900 F total steam temperature resulted in approximately 50 psi drop across the superheater at full load. When operated with a margin of 30 psi under popping pressure, and 20 psi drop through valves and piping, the pressure at the turbine approached 800 psi. In 1939, the requirement for safety valves designed to operate between 900 and 1500 psi was realized, and the purchase of the safety valves for the first boiler of the units, listed in Table 1, was delayed until such a valve was offered. Since that date boilers in this class have usually been built for 950 to 1000 psi at the boiler drum. This has resulted in securing the minimum of safety-valve operation.

All of these units in the 900-psi class are designed for four-stage extraction for feedwater heating, one heater being the direct-contact type arranged for very complete deaeration of the feedwater.

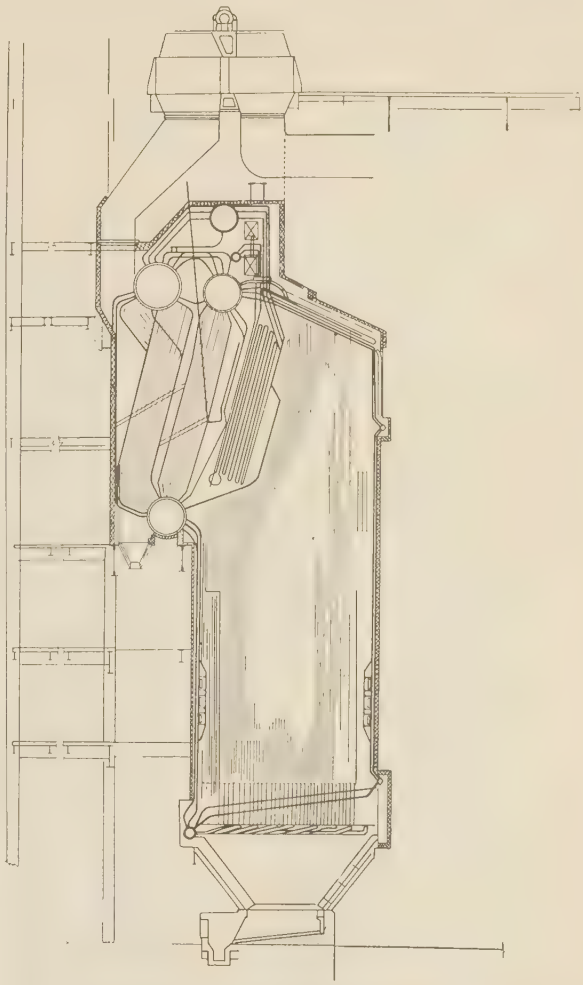

Fig. 3 Cross Section Through Steam Unit Rated at 400,000 Lb Per Hr, at 950-Psi Drum Pressure

Feedwater temperatures are approximately 375 F. Obviously, with the unit system, availability of the boiler for continuous operation over long periods is highly essential. Conventional types were selected and furnace design provided for reasonable heat liberation on the basis of four waterwalls and a dry ash pit.

Fig. 3 is a cross section of a unit rated at 400,000 lb per hr, 950 psi drum pressure, and 900 F. Three of these units were installed. The furnace of this unit has fin tubes on four sides and is tangentially fired with twelve burners, three at each corner. These three boilers and all others built with the units in Table 1 are each equipped with three pulverizers.

Fig. 4 is a cross section of one boiler built for the same capacity and steam conditions and differs primarily in the furnace bottom and the side-wall tubes. This arrangement of the furnace bottom is intended to be safe against damage to the hearth tubes from falling slag, which occurred in the other design, and also affords better protection for the dry-ash pit. The wall tubes are tangent tubes, 3 in. OD on $3^1/_8$-in. centers, which are connected to the headers with a forging which joins two tubes to a $3^1/_4$-in-OD nipple, which in turn is rolled into the header. Obviously, this results in a cooler furnace, which has proved to be less subject to slagging.

Fig. 5 is a cross section of a boiler built for the same steam con-

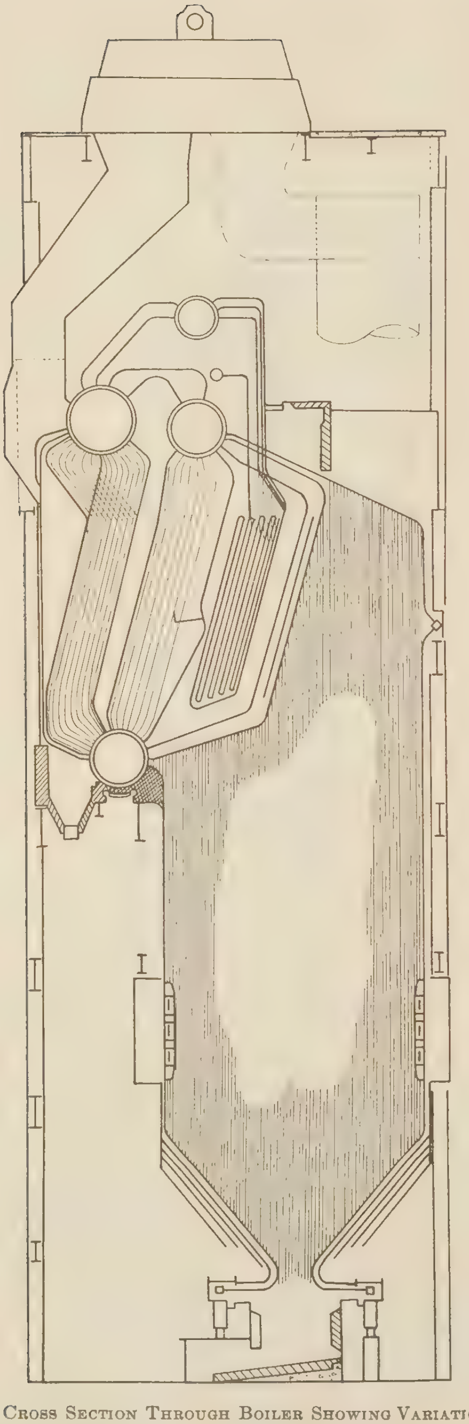

FIG. 4 CROSS SECTION THROUGH BOILER SHOWING VARIATIONS IN FURNACE BOTTOM AND SIDE-WALL TUBES

ditions but with a capacity of 600,000 lb per hr. The wall tubes and furnace bottom follow the pattern of Fig. 4; the design embodies an economizer and omits the dry drum. Similar to those just mentioned, this boiler is tangentially fired and all five units represented by Figs. 3, 4, and 5 are equipped with the regenerative-type air preheaters.

Fig. 6 is a cross section of another boiler of the 900-psi class and 400,000 lb per hr capacity. This unit is fired with six circular burners which are fitted with rings for burning gas, has a relatively large economizer and tubular air preheater. This unit has been in service only a short time, and it is too soon to report on several features of operation based on inspections, but tests show that carry-over from this boiler is a fraction of 1 ppm at rated load and with boiler water having from 340 to 835 ppm dissolved solids. This unit is installed with the primary-air fan ahead of the pulverizers, and comparative results will be studied with interest.

The reason for the selection of boilers with economizers designed to secure 2 to $2^1/_2$ per cent increased efficiency above that for the boilers, shown in Figs. 3 and 4, becomes evident when it is noted that these are the most recent units, will be assigned a more nearly base load, and the price increase was relatively small.

All of these units are arranged with by-pass dampers for control of superheat temperature, so that from 60 per cent to 100 per cent rating 860 F total steam temperature may be maintained. These dampers are manually controlled and for the type of load carried this has proved satisfactory.

Although one might not select the tangentially fired furnace for burning gas as fuel, a number of these have been fitted with gas

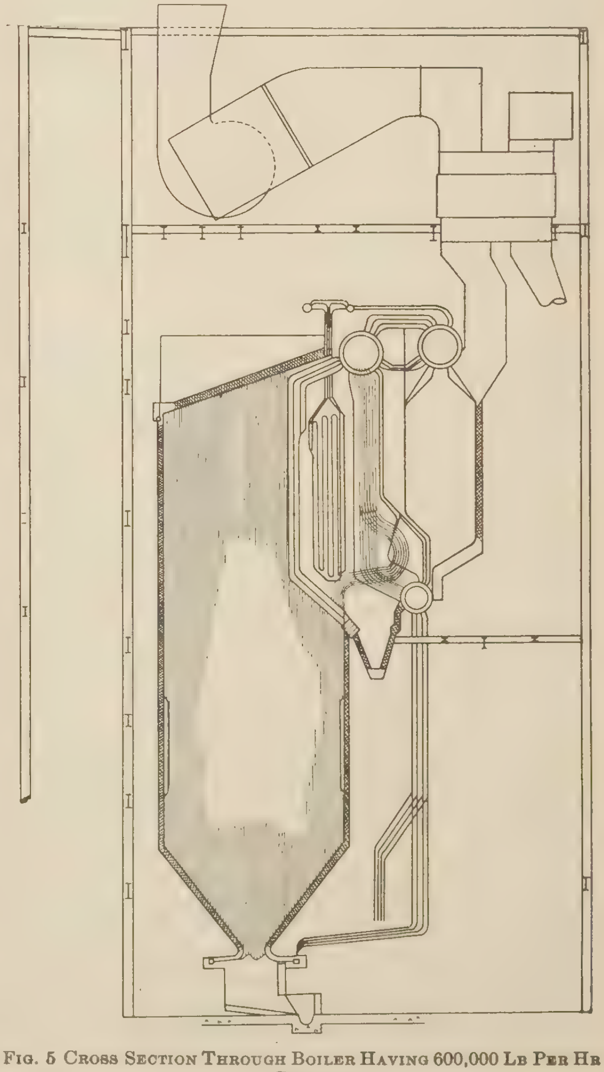

FIG. 5 CROSS SECTION THROUGH BOILER HAVING 600,000 LB PER HR CAPACITY

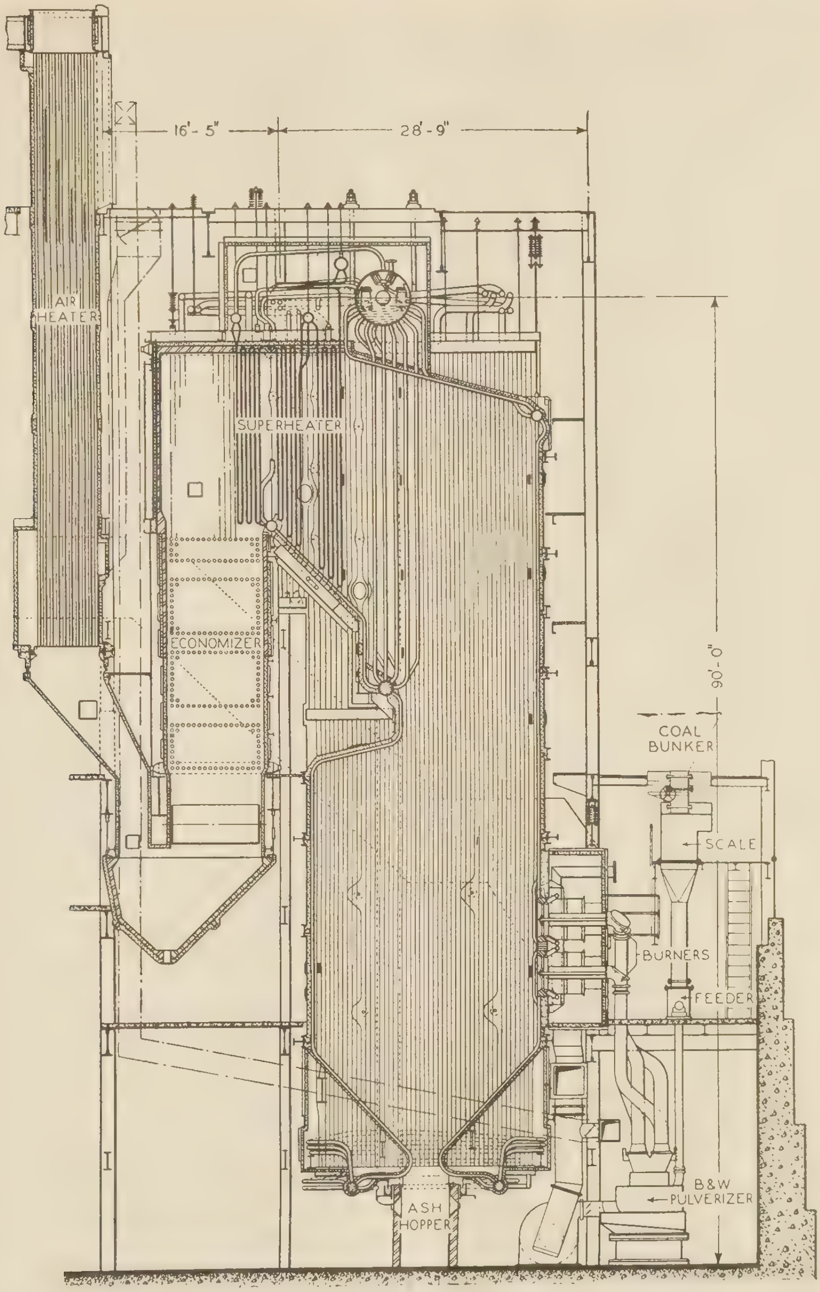

Fig. 6 Cross Section Through Boiler of 900-Psi Class

burners and are performing fairly well. It is interesting to note that during summer months off-peak or "when available" supply of gas to plants Arkwright and Atkinson has approached 2 billion cu ft per month. The availability of burners and pulverizers installed with these units has been practically 100 per cent with no emergency outage. For the first 150,000 tons per mill, when pulverizing coal having a grindability of 45 to 50, there have been no replacements with the exception of exhauster-fan blades.

Operating Experience With Steam Units

In studying the performance of modern boilers, it is perfectly evident that the difficulty with handhole gaskets had caused considerable outage and particular effort was made in improving the design of previous units to reduce the extent of waterwall headers to an absolute minimum and thus eliminate handholes with their plates and gaskets. This was carried to the point of necessitating extremely long tubes, which were fabricated by welding, and required special arrangements for shipping. The cross sections of all these boilers clearly show this feature in the wall tubes. It is evident that a failure in one of these long tubes calls for repair by welding and this has been done successfully.

A study of the superheater of any of these boilers shows how

important it is that all portions of these tubes be free of faults and that no solids shall be deposited by the steam. In all cases tubes are made up with welded joints and at the outlet-header nipples are welded in and stresses relieved before shipping. None of these superheaters is drainable and great care must be taken to see that all condensed steam is evaporated before the tubes are subjected to gases at normal operating temperatures. One failure in the superheater of one of these boilers and of similar installations has been experienced, due to the presence of water in certain tubes, thus preventing the flow of steam and consequent overheating. In regard to the time element to secure safe starting, it may be noted that in general the elapsed time between the first indication of pressure rise in the boiler drum when starting from cold and the time for closing vent valves on the superheater outlet when steam is being supplied to the turbine is approximately $2^1/_2$ hours.

Other failures of superheater elements include erosion, due to leaking baffle, and one case of corrosion. The case of corrosion occurred near the upper end of the superheater element on the inlet side, and since there is no evidence of high temperature it must be assumed that the failure was due to oxygen attack which had been concentrated due to condensate flowing into the tube, or during the shutdown period when this element of the tube was nearly filled with water. This is mentioned because of the emphasis which it places on the necessity either for completely filling the superheater elements with water properly treated, or completely drying these elements when the unit is out of service. This latter is not easy of accomplishment.

Automatic combustion control providing only for the control of fuel and air to the pulverizers, secondary-air supply, and induced-draft-fan operation has been highly satisfactory.

With one exception the pumps for these units are all motor-driven with constant-speed motors, and the feedwater-control element consists of a single valve. Something over 2 years of service from one of these valves, which is provided with good hydraulic characteristics, has resulted in an almost perfect record, and the valve is now reported as being tight enough for shutoff purposes so that the boiler may be banked with a full pressure on the pump side of this valve. In contrast with requirements for boilers supplying steam at a rapidly fluctuating rate where three-element control is used, these valves operate from drum level only. By using a direct-contact heater for deaerating the feedwater and as a storage supply ahead of the boiler feed pump, one high-pressure closed heater has been eliminated. On the other hand, power for pumping increases somewhat, and we have found that pure water at 320 F attacks the steel in the pump more rapidly at points of high turbulence than at 235 F so frequently used. The cure for this has been coating such areas with a high-grade alloy metal or some recirculation of boiler salines.

Although the history of these units is not long enough to be conclusive in regard to carry-over, No. 1 unit at Chickasaw and No. 1 unit at Arkwright were each inspected after 2 years of operation and were found to be relatively free of blade deposits. In fact, this condition was such that continued operation throughout another year would have been satisfactory in this respect. The subject of carry-over embraces boiler performance, quality of feedwater, and finally quality of steam. To secure feedwater with low content of dissolved solids, make-up is secured from evaporators and special attention given to securing tight condensers.

Since the various manufacturers' recommendations were used in securing condenser tubes to the tube sheet, two methods were used. In one case, tubes are rolled at both ends, elongated by heating before rolling-in, and bowed slightly to take care of expansion. The other method calls for rolling at one end and packing at the other end of the tube, which was highly polished to avoid dragging the packing due to any movement caused by expansion. Both methods have proved highly satisfactory.

Starting with high-grade condensate secured from these condensers, water is supplied direct to the boilers without addition of chemicals but with thorough deaeration. Chemicals are supplied individually to each boiler through a chemical feed pump, and the initial operation of all boilers used the well-known phosphate treatment. Enough sodium sulphite was used to maintain from 10 to 15 ppm in the boiler water in order to take care of any residual oxygen or that picked up during shutdown periods. It may be noted here that the record of sulphite used in the case of 450 lb pressure and 850 lb pressure shows a considerable discrepancy, the easiest explanation being that in the higher pressure some of the sodium sulphite is broken up. About 15 or 20 ppm PO_4 in solution is maintained in the boiler water and pH of approximately 11. Under these conditions it was found that the first three boilers installed accumulated a considerable amount of fine black magnetic oxide in the boiler water. This oxide was present in quantities sufficient to cause considerable concern, and it should be noted here that the cure for it was considerably delayed due to the necessity of maintaining all boilers in operation. Some variation in boiler-water treatment was undertaken but without conclusive favorable results. Although no boiler failures have resulted from this type of corrosion, its prevention will be eagerly sought.

Turbogenerators and Fuel-Storage Methods

It will be noted from the data in Table 1, that six of the seven turbogenerator units installed in the last 4 years are equipped with hydrogen cooling for the generators. This has been highly satisfactory, hydrogen use has been less than specified by the manufacturer, and the control equipment has been trouble-free.

All of the 3600-rpm units in this group were built after the developments which provide for mounting the stator iron within a fairly rigid frame so that the 7200-cycle vibration, which has been objectionable in some earlier designs, was practically eliminated. The performance of the exciters on these units has been fairly satisfactory. The manufacturers supplied each turbine with pressure-relief diaphragms on the exhaust casing and automatic throttle closing in case of high back pressure, so that none of these units was equipped with atmospheric relief valves. This equipment has proved satisfactory on these and one 1929 unit.

At the time these plants were being designed, the use of tractors with bulldozers and carryalls for storing and reclaiming coal was gaining favor rapidly and after a test at Plant Atkinson, this equipment was put into use at the four plants: Atkinson, Arkwright, Chickasaw, and Gorgas. The advantages of using this equipment for storing and reclaiming coal are the wide range of operations permissible, low first cost, and the ability to store coal to any height of pile, as much as 30 to 40 ft. Over 200,000 tons have been stored in one pile with this equipment. Furthermore, the dense packing resulting reduces spontaneous combustion to a minimum; in fact, there have been no fires in such piles.

It is impossible to go into detail of all operating characteristics of these units, but it may be said that all changes in equipment worked out through co-operation with the manufacturer during the design of these plants have been quite satisfactory.

The longest forced outages have resulted from the damage to one superheater, previously noted, and the delay in putting two generators into regular service due to hot spots developing in the stator iron, which called for extensive rebuilding by the manufacturer. Erosion of induced-draft fans has caused repeated short outages, and plans are now under way for installing dust collectors ahead of these fans at one plant.

Shortage of manpower has prevented compilation of data to determine closely the performance of these units; such checks as have been made show they are surpassing design performance.

The Co-Ordinated Operation of Hydro and Steam Capacity in Electric Power Systems

By G. W. SPAULDING,[1] BALTIMORE, MD.

Two decades or more ago, numerous technical papers were written on the then popular subject of hydro *versus* steam power, but more recent discussions, presented both in society journals and in other engineering publications, have emphasized the advantages of the co-ordinated use of hydro *and* steam. These later papers have explored and charted that wide range of existing conditions where co-ordinated operation of hydro and steam will be more economical than use of either alone; and today the general principles involved in such co-ordinated operation are understood and applied in many regional power systems. In the first part of this paper, the historical development and the present status of co-ordinated operation in a specific regional power system are reviewed. This is followed by discussion of a possible approach to the use of pondage at run-of-river hydro plants as part of the system reserves, such use being predicated upon the probabilities of noncoincidence of maximum system loads, minimum river flows, and severe forced outage of steam capacity.

The highly diversified industrial area in the Middle Atlantic States, including New Jersey, Delaware, Eastern Pennsylvania, Maryland, and the District of Columbia, is served by seventeen electric utility systems with power-generating resources of over 4,500,000 kw. The individual operating systems of the area, controlled by more than a dozen independent interests, have through the last decade been interconnected by high-voltage transmission lines to form an integrated regional system. Effective co-ordination of operation of the combined hydro and steam resources of this region has been established. The four companies operating in the southern portion of this area completely co-ordinated their hydro and steam operations over 10 years ago, while three of the larger companies in the northern portion began such operation at an even earlier date. Interconnection between the northern and southern parts of the area began about 1931, and over-all co-ordination has subsequently developed. Very recently, interconnections have been strengthened between the northern end of this regional power group and other regional systems in New York State and through them with power systems in New England.

Co-operative studies on a regional basis were initiated in the late 30's. These studies were gradually expanded to assure fullest utilization of the hydro and steam resources in the carrying of increased loads, in the scheduling of maintenance, and in meeting emergency outages of equipment. As a consequence, the production of power at the more economical steam-generating stations, in general without regard to location or company ownership, has resulted in operating economies and more effective utilization of available generating capacity. In addition to the co-ordinated scheduling and operation of capacity, there has been a continuing interchange of energy on an economy basis which has made possible, in the last 2 years, substantial conservation of much needed coal and fuel oil and in relief of fuel transportation.

Regional Load and Capacity Conditions

This regional power area is predominantly urban in character. The electrical-load requirements are typical of what might be expected in metropolitan areas in which are located large and diversified industries. The power resources of this area consist of approximately 85 per cent steam capacity and 15 per cent hydroelectric capacity. Of the effective hydro capacity of about 670,000 kw, nearly 90 per cent is installed in run-of-river plants along the lower reaches of the Susquehanna River, about 10 per cent is supplied from storage projects in the Delaware River drainage basin, while the balance is made up of numerous small hydro installations.

Since there is no upriver storage on the Susquehanna, the plants on this river are essentially run-of-river plants, the pondage at the plants being relatively small in comparison with installed capacities. The Susquehanna is subject to wide variations in natural flow. The flood stages of the river usually occur in the spring of the year, and the resultant reductions of head and capacity rarely adversely affect system capacity requirements, but the experienced low flows in the late summer, fall, and winter do have an important effect on the system capacity requirements. It is in the treatment of this capacity situation that this large interconnected group of utilities may be different from other power systems.

There has been co-ordinated hydro and steam operation in certain of the interconnected systems through varying forms of power contracts for over three decades. Because of this fact, most of their steam-capacity installations have been designed for sustained or base-load operations, that is, the turbines and more particularly the boilers have been designed and rated for operation for 12 or more hours a day, which operation is required with low river flow when the peak portion of the load is carried by the hydro plants. On other power systems in the area, experiencing generally the same daily and seasonal load shape, but with relatively less hydro capacity, the steam capacity has, through the years, been designed to carry a large portion of the daily peak loads, normally of short duration, and cannot be operated at maximum output for sustained periods. In co-ordinated operation, the normal limitation of this peak steam capacity must be recognized.

Under present-day conditions, with high-load-factor wartime loads, and absence of sharp evening peaks resulting from extension of daylight-saving time, and, until recently, curtailment of lighting in coastal areas, it has been necessary and desirable to use all steam capacity on a more or less sustained basis. This means that those plants designed for peak operation must now be operated for longer periods at reduced loading. The sharp peak loads, or what is left of them, are scheduled to be carried by hydro under critical capacity conditions.

Essential Factors in a Co-Ordinated Regional Power System

There are obviously several essentials to the complete and

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full co-ordination of the combined generating facilities of a regional power system, in addition to the necessary physical facilities, particularly when the various operating systems in the power region are controlled by independent interests. These are as follows:

1 The first essential is the availability of power contracts or interconnection agreements between the several companies of the area which are sufficiently flexible to cover the wide range of required operation. Fortunately, there were such flexible interconnection agreements in this regional area, dating back at least 5 years. There must, of course, be an incentive for co-ordination, and this can best be provided within the framework of the power agreements.

2 Considerable operating experience is necessary before the fullest advantage from co-ordinated operation can be obtained. Our co-operative studies were expanded in both scope and detail in the light of experience gained during the several years preceding the present increased war loads, such experience establishing not only the basis for determining the amount of capacity gained from co-ordinated operation, but also the routine planning and scheduling of operation and of maintenance.

3 A procedure for day-to-day direction and control of co-ordinated operation is essential. Experience has indicated, when other essentials are provided for, that the maximum advantages can be obtained without a top regional power dispatcher, so long as there is a co-operative effort on the part of the several system dispatchers to work in the interest of over-all operating economy. This co-ordinated operation, which has been of such material aid in the war effort in this regional area, has been accomplished without encroachment upon the individual responsibilities of the many operating companies.

Joint Operating Problems Not Difficult to Handle

The operating problems of system frequency control, time variation, and tie-line controls have never been difficult, largely because of the capacity of the interconnecting transmission network. This consists primarily of two high-voltage 220-kv transmission systems, themselves interconnected by a grid of double-circuit 66, 110, and 132-kv lines. There are no automatic frequency controls or tie-line loading controls on the regional system. The responsibility for frequency control is assigned, from time to time, to one of the five larger systems.

Telemetering is used extensively in the several dispatching offices to give indications of tie lines, system loads, and station generations.

Problems of policy are considered by a "Steering Committee," consisting of the top operating executives of each operating company in the region, but experience has indicated that once the general principles are agreed upon, frequent meetings of this Steering Committee are not necessary. The detailed operating and accounting problems are handled through so-called "Operating Committees," on which each company or group of companies is represented by operating men.

One of the purposes of the continuing co-operative studies is the determination of the maximum firm hydro capacity available from the run-of-river hydro plants when operated as a co-ordinated part of the regional system capacity. Our problem was not one of determining whether hydro or steam capacity should be provided to serve the increasing system load, but rather how much more war load could be carried on the regional system by complete co-ordination of the three large run-of-river hydro plants and the available steam capacity. These hydro plants are owned by different utilities and, although geographically close together, are electrically connected with different parts of the regional system. While there were found to be minor limitations in the power that could be interchanged over the regional network, particularly between the northern and southern sections, these limitations did not impose any serious restrictions on the complete utilization of the available power resources. A detailed discussion of some of the general factors that enter into our studies of firm hydro capacity may be of interest in connection with further considerations of the economies of run-of-river hydroelectric developments.

Hydro Installation

There are four run-of-river hydroelectric plants within a 50-mile stretch in the lower reaches of the Susquehanna River adjacent to tidewater. Data on these plants are given in Table 1.

The operation of run-of-river hydro plants in a hydro-steam system has been frequently explained, and it is sufficient here to call attention only to the low ratio of hydro to steam capacity in this regional system. Under normal operating conditions, the Safe Harbor pond can be used for weekly regulations of low flows; but there are other conditions under which such operation is either not desirable or not possible. In general, the use of the Safe Harbor pondage differs for three possible conditions of operation, as follows:

1 When there is ample system capacity, i.e., an excess of reserve capability, the Safe Harbor pond elevation is scheduled to provide for most economical use of hydro energy on the combined system load. The use of pondage is so scheduled as to have the forebay at top elevation Monday morning.

2 When, because of low river flow or reduced steam reserve capability, the capacity situation appears critical, the Safe Harbor forebay is maintained as near top elevation as possible, at least during the first part of the week, in order that maximum pondage will be available in the event of forced loss of steam capacity, i.e., for emergency use.

3 When, because of low river flow, coincident with loss of steam capacity due to forced outages, there is a deficiency in steam-generating capacity on the system, i.e., an emergency exists, the Safe Harbor pondage may be drawn on to obtain the needed energy from the three lower plants. It should be recognized that the only alternative to such operation would be to curtail system load. As soon as steam reserves are again available, hydro generation is curtailed to restore the forebay to normal.

Load Shape and Maintenance Requirements

The 15-min integrated peak loads for each week in a wartime

TABLE 1 HYDRO PLANTS ON LOWER SUSQUEHANNA RIVER

Miles from tide-water	Plant	Year of initial operation	Number of units	Head, ft	Maximum plant discharge, cfs	Effective plant capacity, kw	Pondage, in terms of available energy,[a] kwhr
50	York Haven	1904	20	20	18000	20000	...
27	Safe Harbor	1931	7	55	65000	230000	10500000
19	Holtwood	1910	10	51	32000	104000	1200000
4	Conowingo	1928	7	89	44000	252000	5300000
				215		606000	17000000

[a] Energy that can be generated from useful pondage, including both generation at site and at downstream plants.

and a prewar year are shown in Fig. 1. The critical capacity conditions might be expected, from these data alone, to occur in December. In an all-steam system that conclusion would be obvious; but in a hydro steam system without storage, where hydro capacity varies from month to month depending upon river flow and reaches minimum values about September, the late summer loads may be the most critical. Maintenance requirements and condensing-water temperature effects on steam capacity also must be considered in this connection.

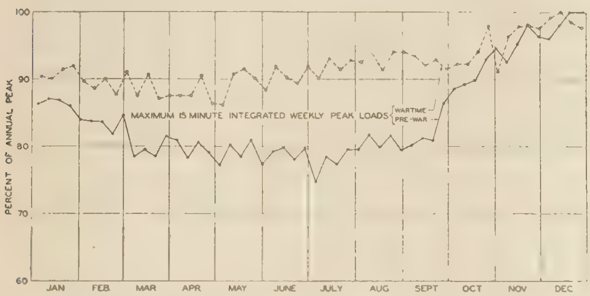

Fig. 1 Annual Load Shapes of Regional Power System

The valleys in the prewar annual load curves of the regional area and to a lesser extent in the wartime load curve, together with the seasonal variations in the amount of available hydro capacity, control to a large extent the scheduling of steam-unit maintenance; but because it is normally possible to schedule most maintenance at times other than the expected time of occurrence of system peak load, maintenance usually has little or no effect on system capacity requirements.

River-Flow Experience

For many years, computations of firm hydro capacity have been based upon the most adverse river flows then of record, determining for such flows the hydro capacity that would be firm on maximum expected loads with pondage used over some specified period of time, determined by the assumed length of simultaneous forced outage of steam reserves. The improbability of the coincidence of all these conditions has led to a growing feeling that such assumptions are extremely "conservative."

The wartime need for using maximum-capacity resources gave added impetus to several earlier studies relating to the applicability to capacity computations of river flows somewhat greater than the minimum flows of record, but still coincident with the maximum system peak loads, and to the use of Safe Harbor pondage over a shorter period of time, which in turn is related to more recent experience on the probable length of simultaneous forced outages of steam units.

Prior to the increase in load resulting from war, the minimum weekly flows of record for each month were used in capacity computations; but currently the effective hydro capacity is computed for river flows that can be expected to be equaled or exceeded in about 93 per cent of the years (based upon flow records from 1891 to date). For the river flows thus selected for each low-flow month, the effective hydro capacity is determined for the maximum weekly load estimated for that month, with pondage used on a weekly cycle of drawdown and refill. This hydro capacity is then combined with steam capacity, adjusted for circulating-water temperatures, and for scheduled maintenance in arriving at the minimum total effective system capacity for that month, i.e., for load-carrying capacity and for system reserves. The excess of this computed capacity over the estimated system load is the capacity available for reserves and for new loads; the latter being of particular interest to the companies themselves and to the government agencies responsible for the location of war industries.

In the foregoing procedure, there are certain hydro resources which have not been fully evaluated. Additional hydro capacity is available for something less than 93 per cent of the time, resulting from river flows in excess of those used in capacity computations. Such capacity is valuable for the release of steam units for maintenance work in addition to that scheduled in advance, and much maintenance of this nature has recently been necessary. Additional hydro capacity is also available from the short-time use of pondage, when required by the simultaneous forced outage of a large number of steam units, even with stream flows lower than those used in capacity computations. The extent to which a system would be willing to rely on such capacity as firm would be influenced by the forced outage record of its own or similar steam-generating equipment, and upon the amount of other reserve capacity then available on the system.

Perhaps someone will eventually evaluate and properly combine all the pertinent factors by the application of a probability theory. As this theoretical solution now seems impracticable, an attempt has been made to approximate the possible use of pondage in the light of the forced-outage experience of steam-generating equipment, the ideal objective still being that a combined hydro and steam system should have substantially the same degree of reliability as an all-steam system.

Forced-Outage Experience of Steam Capacity

Forced or emergency outages of steam turbines and boilers are of relatively infrequent occurrence; and only over a period

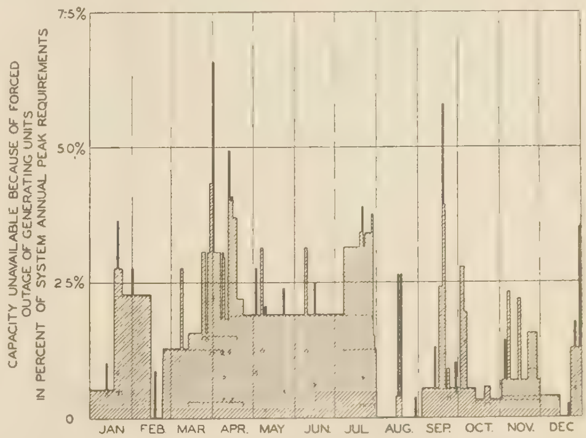

Fig. 2 Actual Forced Outages of Generating Capacity, 1943
(Exclusive of all scheduled outages or losses of capacity due to causes outside of the generating plants.)

of years could a system accumulate sufficient data to determine reasonably well the average forced-outage probabilities of its generating units. These probability factors may differ between systems in which there are different types of units or different operating requirements and maintenance practices. Although conclusions regarding either detailed differences or similarities in the probability factors must await collection of additional data, the general fact that simultaneous outages of a large number of generating units are infrequent and of short duration seems to be well established. This fact is fundamental in the determination of required system reserve capacity and the possible use that can be made of pondage.

The term "forced, or emergency outage" is sufficiently descriptive so that no precise definition is here required. We are interested only in the outages of facilities which have a bearing

upon the capacity resources of the power system, and we are not here concerned with scheduled outages for inspection, where such outages can be scheduled in advance.

It will be noted in Fig. 2 that a number of fairly long steam-unit outages were experienced on the regional system in 1943. Three of these prolonged outages existed concurrently on April 1, at the time of the largest total forced outage of capacity during the year. Yet, more than one half of this total simultaneous outage only lasted from 1 to 3 days. In spite of these several long outages, the average forced-outage time for this year was somewhat better than might have been expected from our own earlier experience; but such variations would be quite normal in year-to-year operation of any system.

Most of the individual and all of the more serious simultaneous outages, shown in Fig. 2, were of short duration. The absence in this one year of long severe simultaneous outages of capacity does not prove that such outages cannot exist, but the probabilities are decidedly against them. Data for earlier years generally confirm those shown for 1943, and give weight to the opinion that severe simultaneous forced outages of steam capacity are infrequent and of short duration. This point is important in connection with the use of pondage as firm capacity.

Use of Pondage for Short-Time Capacity

The pondage, which system operations have indicated as usable by the lower Susquehanna River plants, is equivalent to 17,000,000 kwhr. This is nearly $2^1/_2$ times the weekly hydro energy available from the lowest 7-day average river flow recorded in the month of December. Some portion of this pondage might be used (not to be confused with normal weekly use of pondage where the forebay is returned to top elevation on each Monday morning) as system reserve capacity on those infrequent occasions when severe multiple forced outages exist coincident with peak week loads and minimum flows. Such pondage operation is illustrated in Fig. 3, for the conditions assumed in capacity computations in contrast with conditions more likely to be met in actual operation.

The diagram at the left in Fig. 3 shows the assumed simultaneous outage of steam capacity equal to the ordinate Y. The energy made available by use of only 40 per cent of the available pondage, during the first 5 days of a period beginning Monday morning, would permit hydro capacity to replace an amount of steam capacity equal to the largest steam unit on the system (135 mw), or to the aggregate capacity of several smaller units. This capacity is indicated by the dimension X. (It is here assumed that there would be sufficient installed capacity at the hydro plants to permit such operation, which, however, is not presently the case.) By the following Monday morning, X capacity, out of the total forced out, is assumed to be returned to service. At that time about 60 per cent of the stored energy is still available as pondage, and this condition would be maintained until additional amounts of steam capacity are returned to service to permit gradual refilling of the ponds. The pondage will also be refilled in part as higher river flows are experienced or lighter system loads are encountered, both of which will have an appreciable influence, since it had been assumed from the beginning that such use of pondage coincided with maximum weekly loads and minimum river flow.

The diagram at the right in Fig. 3 assumes simultaneous outage of steam capacity such as was experienced by this system during the middle of April, 1943, the maximum of the simultaneous outage again being equal to Y capacity. In both diagrams the hatched areas indicate the capacity X and X_1 that could be obtained by drawing down the ponds of these three run-of-river hydro plants, while the cross-hatched portion represents capacity, which had been forced out at the beginning of the period but is returned to service, and the approximate time to bring the forebay back to top elevation. A comparison of the assumed and the more probable outage condition indicates that the assumption shown at the left is, indeed, conservative in the light of forced-outage experience; but it is an approach to the use of pondage for at least a portion of the system reserve.

It is necessary to emphasize that the amount of hydro capacity which may be available from drawdown cannot be considered as replacing the entire system reserve requirements. Obviously, it is impossible for pondage to be used as protection

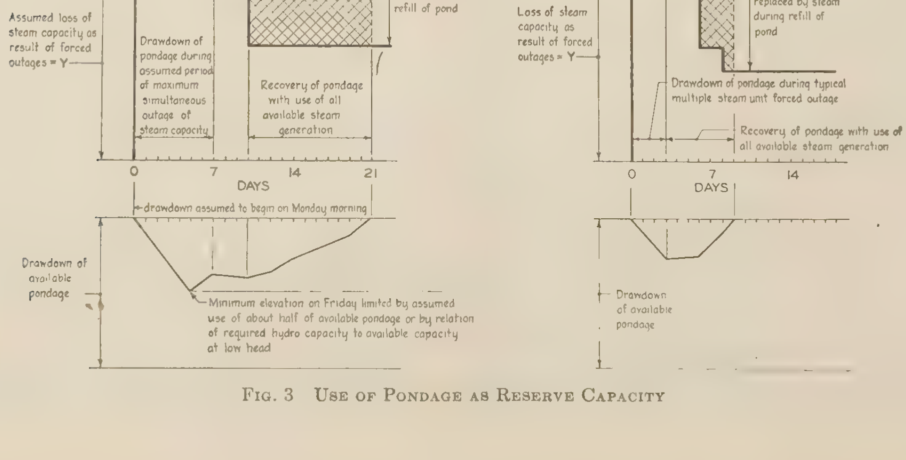

Fig. 3 Use of Pondage as Reserve Capacity

against all outages of steam capacity, for these are almost continuously existent; in low flow the available pondage would soon be exhausted as opportunities for refill would be inadequate. Neither could pondage provide protection against errors in load estimates which must be made several years in advance, nor could it provide capacity to cover scheduled maintenance where required during the peak load period of the year. However, under normal conditions, pondage might be considered for about one third of the total required system reserve.

Under present-day conditions, with necessity of meeting war loads with less system reserve than would be provided in normal times, together with the realization that any new loads added in the area might likely be of high load factor, it has been felt that the use of pondage should be limited in capacity estimates to that used on a weekly recovery cycle. It is appreciated that there is in storage, hydro energy available as emergency reserve.

Conclusion

The co-ordinated operation of all the interconnected utilities in this regional system, initiated by the utilities themselves for economic reasons even before the war clouds in Europe gathered, has proved of immense value in meeting electric-power demands of the national war effort. There has been considerable satisfaction in the realization that the saving of critical fuel and its transportation has been accomplished, and that the area has been able to absorb increased war loads because of such co-ordinated operation on a regional-system basis.

Evaluating Importance of the Physical and Chemical Properties of Fly Ash in Creating Commercial Outlets for the Material

By C. M. WEINHEIMER,[1] DETROIT, MICH.

The paper presents the reasons for emphasizing the physical properties rather than the chemical constituents in outlining an investigational program for fly-ash disposal and outlines the procedure followed in determining the potential consumers of this material. The results of a laboratory and field investigational program are presented and various methods and equipment for modifying the ash to meet special requirements are discussed. Commercial markets and other possible uses for fly ash are discussed together with the handling equipment and the storage facilities necessary in using bulk fly ash. Finally, certain conclusions are presented regarding the general problem of fly-ash disposal.

Any company which operates pulverized-coal-burning equipment so located that collection of the fly ash is desirable and has available low-cost, adjacent land on which to dump it, is indeed fortunate. The company with which the author is associated is now in that enviable position; the long-time outlook, however, may not be so favorable. It is with this thought in mind that it has been considered advisable from time to time to sponsor work having as its purpose the development of revenue-producing outlets for fly ash. It is self-evident that any fly ash disposed of at a profit is not only a source of revenue in itself but at the same time eliminates disposal charges. It is conceivable that the profit derived from the sale of a portion of the fly ash might cover the cost of disposal of that portion for which there is no market.

In approaching the fly-ash-disposal problem, it can be considered that the value of the material lies in its physical and chemical properties, or in its chemical constituents.

The best hope for a satisfactory outlet seemed to be in capitalizing on the finely divided state in which all fly ash exists, and the major part of the investigational work was conducted with this property in mind. Nevertheless, it was felt that a certain amount of information on the chemical characteristics of fly ash would be necessary in order that we might talk intelligently with prospective customers, and, therefore, rather extensive chemical analyses were made of fly ash "as collected" and of a number of sieve fractions. In this connection it is well to make a distinction between chemical composition and chemical properties.

Laboratory Investigation

A laboratory program was undertaken, the object of which was to determine how fly ash compared physically and chemically with the materials that the ash might replace. Some of the physical and chemical properties and chemical constituents of fly ash are presented in Tables 1 and 2. Although most of this information is of a commonplace nature and is self-explanatory, a few of the properties, together with some not shown in the tables, warrant a brief discussion.

Physical Properties

Particle Size. In dealing with a material as fine as most fly ash, it is usually desirable to know more about the particle-size distribution than it is possible to learn from a sieve analysis. There are numerous ways of determining particle size in the subsieve-size range and a few of them as applied to fly ash will be discussed.

It was found that different technicians, using the hydrometer method, described by Biddle and Klein,[2] and the same suspension medium, could arrive at reproducible results within 5 per cent. However, when different suspension mediums were used, a wide divergence in results was obtained. The detailed results of this investigation are shown in Table 3.

Subsequent results obtained, using a Roller particle-size analyzer, indicate that the surface area ranges from 3500 to 4100 sq cm per g. Apparently, better dispersion is obtained when air is used for the suspension medium rather than water or kerosene.

In order to avoid the necessity for determining particle-size distribution by any one of the methods suitable for such determinations in the subsieve size ranges, in case such equipment was not available, an investigation was made of the possibility of making sieve analyses and obtaining the distribution of the finer particles by means of curves on logarithmic-probability paper. With the procedures used in this investigation, air elutriation and dry sieving, the results were not very encouraging.

Color. The color of fly ash is affected to a great extent by the carbon or coke particles. Even in those ashes that contain only a very small amount of carbon, the color is still some shade of gray. The results reported in Tables 1 and 2 show that the carbon content is much higher in the coarse than the fine-sieve fractions, yet the finest-sieve fraction with approximately one half the carbon content of the original material is reported as having the same color as the original material. In another case, where the carbon content was reduced to a little over 1 per cent, the color was distinctly gray. An explanation of this phenomenon is found in the fact that color is influenced to a greater extent by surface area than by weight. Thus in spite of the lower percentage by weight of carbon in the fine-sieve fractions, the surface area is probably much greater and the coloring effect more pronounced.

Fig. 1 shows two materials mixed in such a manner as to demonstrate this principle. These bottles contain mixtures of light and dark powders. The percentages by weight of the two powders is the same in each case, but the dark powder in the dark-appearing sample is of a much finer particle size than the dark powder in the light-appearing sample.

Particle Shape and Structure. There has always been con-

[1] Research Department, The Detroit Edison Company.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

[2] "A Hydrometer Method for Determining the Fineness of Portland-Puzzolan Cements," by S. B. Biddle, Jr., and A. Klein, Proceedings of the A.S.T.M., vol. 36, Part 2, 1936, p. 310.

TABLE 1 CHEMICAL CONSTITUENTS OF PULVERIZED-COAL FLY ASH

Constituents	Ash as a whole, per cent	Per cent by weight of sample as collected, dried — Sieve fractions (Tyler Standard): On 35	On 100	On 150	On 200	On 325	On 400	Through 400
Iron as Fe_2O_3	9.10	4.40	4.50	5.50	6.05	7.90	8.15	9.07
Carbon as C	10.64	76.30	60.50	31.60	24.75	16.36	13.49	5.20
Magnesium as MgO	0.83	0.19	0.22	0.24	0.81	0.83	0.86	1.09
Calcium as CaO	2.38	0.63	0.64	1.17	1.37	1.46	1.48	2.38
Aluminum as Al_2O_3	26.90	5.71	10.53	20.79	23.43	25.40	26.56	28.31
Sulphur as SO_3	1.28	0.60	0.70	1.00	0.92	0.86	0.58	1.20
Titanium as TiO_2	1.35	0.45	0.60	0.95	1.10	1.20	1.30	1.47
Carbonate as CO_3	0.01	0.03	0.01	0.01	0.02	0.01	0.01	0.004
Silicon as SiO_2	45.93	8.42	17.46	35.95	39.23	42.21	44.59	48.83
Phosphorus as P_2O_5	Trace	Trace	Trace	Trace	Trace	Trace	Trace	Trace
Undetermined	1.58	3.27	4.84	2.79	2.32	3.77	2.98	2.45

TABLE 2 PHYSICAL PROPERTIES

	Ash as a whole, per cent	Sieve fractions (Tyler Standard): On 35	On 100	On 150	On 200	On 325	On 400	Through 400
Separated fractions retained on each sieve, per cent of sample sieved	....	0.1	3.3	3.0	4.6	8.6	7.0	73.2
Sieve fractions; expressed as cumulative per cent of sample sieved	....	0.1	3.4	6.4	11.0	19.6	26.6	....
Moisture, per cent by weight of dry sample	0.28	1.04	1.06	0.50	0.49	0.43	0.20	0.24
Solubility in water at 70 F, per cent by weight of dry sample	1.45[b]	3.44[a]	1.90[a]	1.37[a]	1.86[b]	2.33[b]	2.02[b]	1.51[b]
Solubility in water at 180 F, per cent by weight of dry sample	1.53[b]	3.19[a]	2.03[a]	1.55[a]	1.83[b]	2.35[b]	2.13[b]	1.55[b]
pH value of water in contact with ash 4 hr at 212 F	7.95[c]	2.75	3.25	5.90	4.00	3.50	3.95	8.00
Effect of dry ash on iron and lead	None	None	None	None	None	None	None	None
Effect of wet ash on iron, loss of weight in mg per sq cm exposed	1.94	2.56	6.48	2.62	1.53	2.63	0.82	0.73
Effect of wet ash on lead, loss of weight in mg per sq cm exposed	0.33	4.80	4.34	1.74	1.01	1.61	0.39	0.29
Settling rate in kerosene, per cent faster than asbestine	15	....	....	.....	.....	....	13	10
Linseed-oil absorption, ml of oil per 100 g of ash	....	....	....	.....	.....	.....	92	46
Water preferential over asphalt oil, per cent in water phase	....	....	....	.....	.....	25 (On 325 to Through 400)		
Specific gravity, apparent	2.28	1.75	1.82	1.87	1.97	2.00	2.15	2.38
Unit weight, lb per cu ft	45.7	18.63	17.68	23.17	21.94	33.98	33.64	47.80
Color of fractions as sieved	Gray	Black	Black	Black and white	Black and white	Dark gray	Dark gray	Gray
Color of fractions burned free of carbon	Buff	Red-brown	Brown	Light brown	Dark buff	Buff	Buff	Light buff
Pyrometric cone equivalent:								
Initial deformation, deg F	2620	2430	2730	2730	2680	2680	2680	2620
Softening point, deg F	2690	2530	2820	2830	2750	2760	2750	2700
Initial fluidity, deg F	2800	2620	2840+	2840+	2810	2820	2810	2800

[a] Dissolved portion contained an appreciable amount of bituminous oil.
[b] Upon evaporation the solid matter consisted chiefly of gypsum.
[c] The pH value of pure water is 7.0 at 20 C.

TABLE 3 SPECIFIC SURFACES OF VARIOUS MATERIALS TESTED IN KEROSENE AND WATER MEDIUMS

Sample tested	Specific surface (sq cm per g) in medium used in test: Kerosene (+ cp oleic acid)	Water: TDA[a]	Saponin[a]
Portland cement (accepted value 2630)	2550	..	..
Sample B fly ash, as collected:			
Through 100-mesh Tyler sieve	1121	2332	2343
Through 325-mesh Tyler sieve	1097	2672	2705
Sample B fly ash, carbon-free:			
Through 100-mesh Tyler sieve	..	1398	..
Through 325-mesh Tyler sieve	2510	1777	1647
Weathered fly ash:			
Through 100-mesh Tyler sieve	1380	1789	..
Through 325-mesh Tyler sieve	1830	2326	.
Limestone dust:			
Through 100-mesh Tyler sieve	1196	1571	.

[a] Dispersing agents.

siderable verbal if not written discussion about the physical structure of fly-ash particles. From visual microscopic examinations of the ash itself and a study of photomicrographs, it is possible to make the following statements:

The carbon exists as irregular, porous, cokelike particles. The noncombustible particles generally have a spherical shape. A portion of the noncombustible particles, although apparently a small percentage, are thin-walled hollow glass spheres. Most of the dark-colored particles are cokelike material. The other dark-colored particles ordinarily spherical are believed to contain most of the iron compounds. Photomicrographs in Figs. 2 and 3 clearly show the physical condition of certain size fractions of fly ash and also how the condition is affected by burning out the carbon.

Thermal-Insulation Properties

The knowledge that some of the particles of fly ash are hollow glass balls invariably brings forth the idea that the loose ash

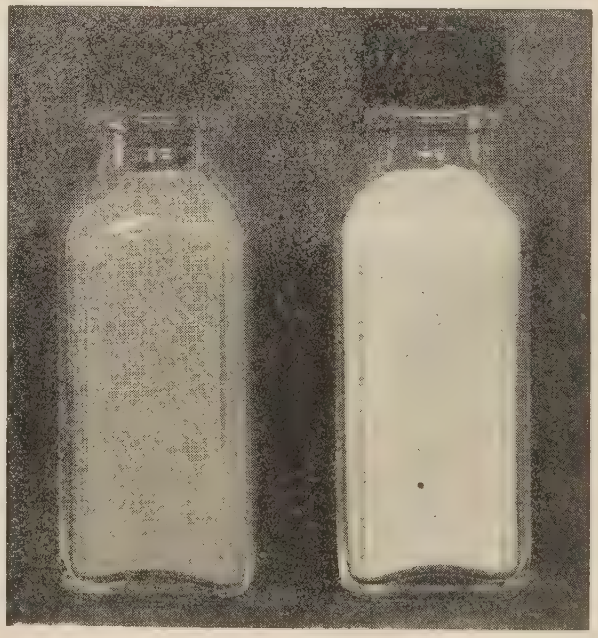

Fig. 1 Influence of Particle Size on Color

Dark Powder	Light Powder
90 per cent by weight of —150 + 200 mesh	90 per cent by weight of —150 + 200 mesh
10 per cent by weight of —400 mesh	10 per cent by weight of —150 + 200 mesh

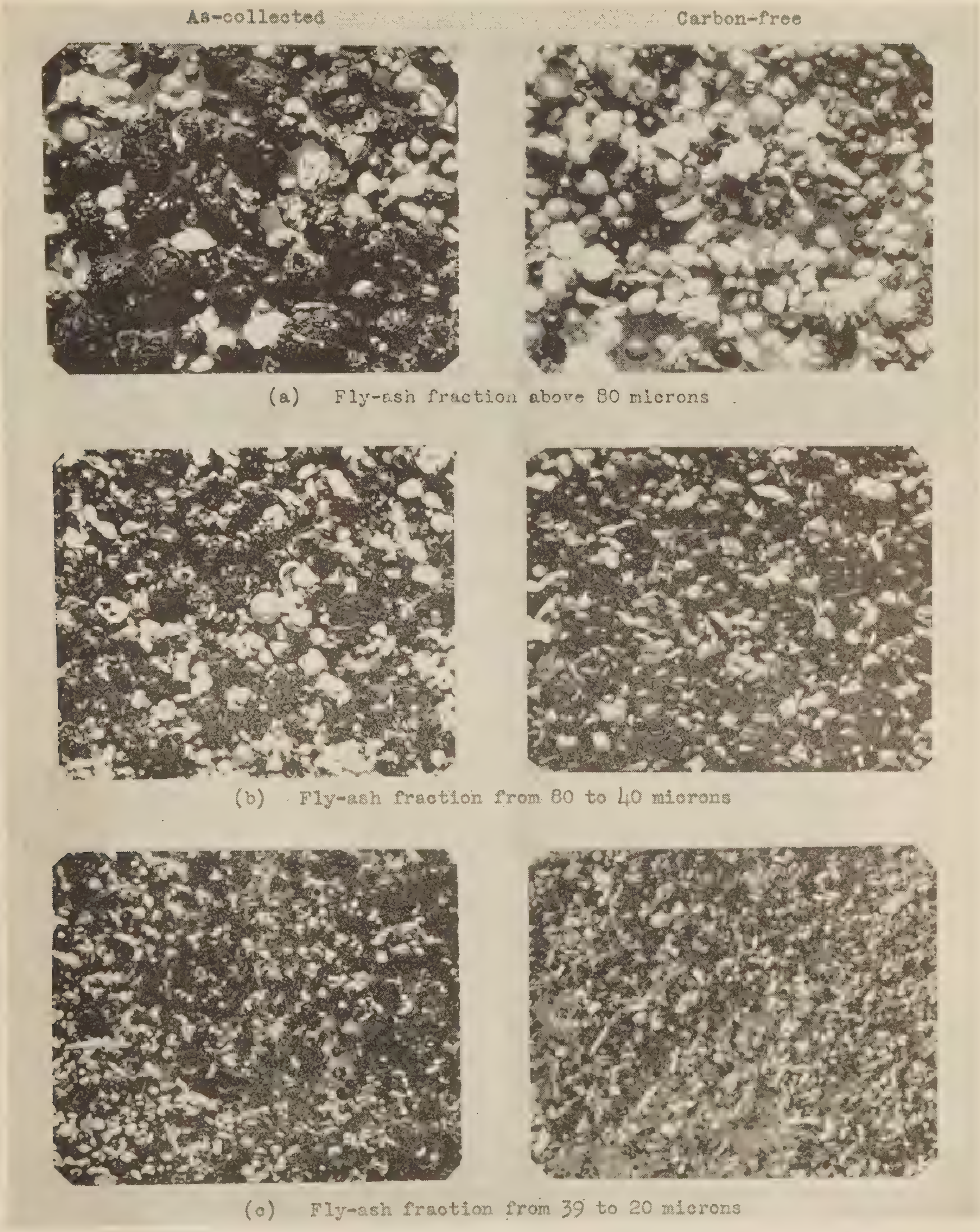

Fig. 2 Physical Structure of Fly Ash in the "As-Collected" and "Carbon-Free" Condition; ×33

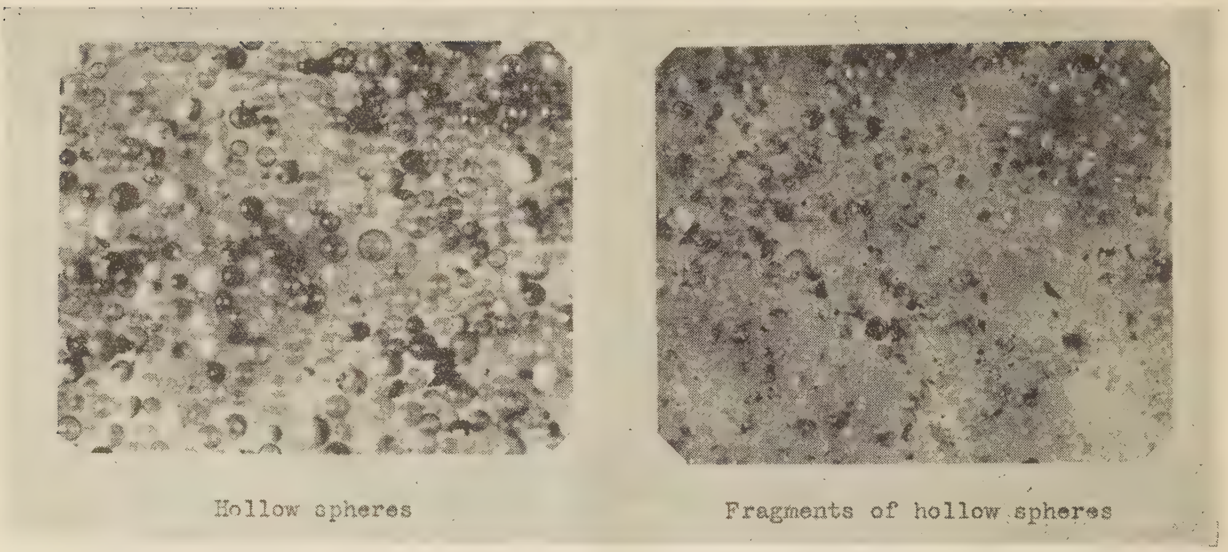

Fig. 3 Example of Hollow-Sphere Particles in Fly Ash; ×33

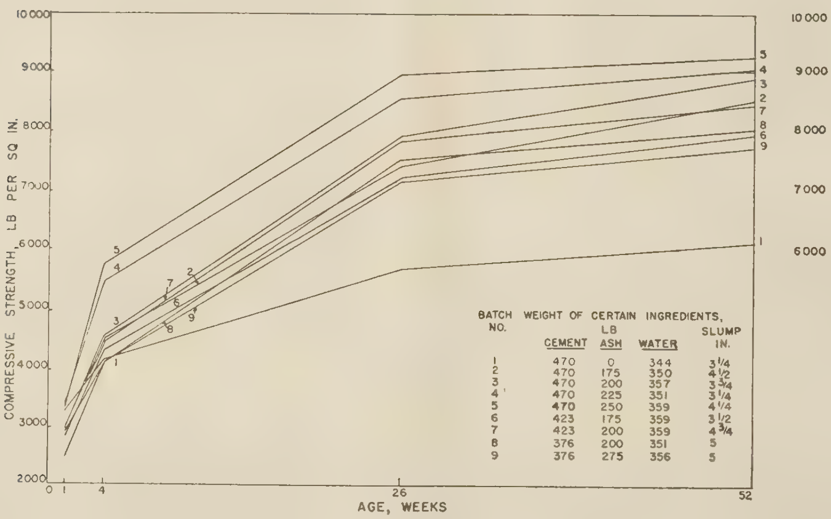

Fig. 4 Effect of Various Cement and Ash Contents on Compressive Strength of Slag-Sand-Ash Concrete

should make a good thermal insulator. These hollow spheres are very friable, however; hence it is quite probable that a great many of them are broken in the course of normal handling. In spite of this apparent weakness, it was decided to obtain the thermal conductivity value k, Btu per sq ft per hr per in. thickness per deg F, for bulk fly ash. The values obtained in the investigation ranged from 0.74 at 75 F mean temperature to 1.04 at 356 F mean temperature. Apparently the k value is appreciably affected by the degree to which the ash is consolidated. This is clearly illustrated by the following data:

Density, lb per cu ft	46.7	47.5	50.1	50.0
Mean temperature, deg F	74.9	75.0	74.6	76.0
k value	0.740	0.768	0.863	0.868

A test made on an 8-in. wall of Cottrell block 5 ft square revealed the following information:

Type of insulation	U^a
None	0.383
Rock wool	0.159
Bulk fly ash	0.179

[a] Over-all conductivity coefficient corrected to a 15-mph wind velocity.

In a number of houses constructed of Cottrell block, fly ash has been used as insulation with apparent satisfaction.

Inasmuch as widespread use of fly ash as a loose insulation material would not be made without information on its hygroscopic properties, a limited amount of work was done in this connection. With a surrounding atmosphere having a relative humidity of approximately 36 per cent, it was found that the ash absorbed 0.15 per cent of its weight in 20 min. The rate of absorption decreased rapidly after the initial period and at 160 hr had reached an equilibrium state at 0.48 per cent.

TABLE 4 RESULTS OF AN INDUSTRIAL SURVEY OF FLY-ASH DISPOSAL

Industry	Material previously used	Can ash be used— Technically	Commercially	Remarks
Acoustic				
dum dum	Slag dust	Questionable	Yes	Too light in weight
Agriculture				
fertilizer	Spent sand	No	No	Too fine; spent foundry sand delivered for $0.40 per ton
Soil-amendment agent	Unknown	Yes	Yes	Probably would not produce any revenue
Alumina	Bauxite	Yes	No	Uneconomic under present conditions
Building block:				
Cement block	Cement	Yes	No	Detroit market demands light-colored block
Cinder block	...	Yes	Yes	Improved quality of block
Cottrell block	...	Yes	...	...
Glazed block	...	Yes	Yes	Glazed, colored tile for decorative work
Building brick:				
Fired	Clay	No	No	Unsatisfactory product
Sand lime	Sand	No	No	Unsatisfactory product, dark color
Ceramic industry:				
Drain tile	Clay	Yes	No	No improvement in quality of product and cost of manufacture increased
Hollow tile	Clay	Yes	No	
Flower pots	Clay	Yes	No	
Concrete:				
Transit mixed	Cement	Yes	Yes	Improves quality, reduces cost
Cinder ash	...	Yes	Yes	Lightweight concrete
Floor tile:				
Asphalt tile	Clay	Yes	No	Shipping distance too far. Color limitation
Foundry:				
Core sand	Sand	No	No	Too fine; makes a dense core
Parting sand	Limestone dust	Yes	Yes	Aluminum and magnesium foundries
Facing sand	Sand	No	Yes	Softening point too low for steel
Hot tops	Calcined clay	Unknown	No	Believed to be unsatisfactory
Gaskets	Clay	Unknown	Yes	Believed to be unsatisfactory
Paint	Extender	Questionable	Yes	Lacks covering power; difficult to disperse
Paper	Clay	No	Yes	Color, no matting properties
Plastics	Diatomaceous earth	Questionable	Yes	Poor quality of product
Polishing:				
Buffing compound	Pumice	No	Yes	Does not produce desired finish
Tooth paste	Whiting	No	Yes	Too hard
Portland cement:				
Raw material	Clay	Questionable	Yes	High alumina content is objectionable
Purification:				
Water filter	Sand	Questionable	...	Market small; properly graded material required
Oil	Fuller's earth	No	Yes	Does not produce clarification
Filter aid	Diatomaceous earth	No	...	Does not function as such; wrong type of particle
Zeolite	Zeolites	Yes	No	Small fraction satisfactory; low capacity
Putty:				
Furnace cement	Whiting	...	...	No information was obtainable
Sash putty	Whiting	Questionable	...	Dark color objectionable
Caulking compound	Asbestine	Questionable	...	Dark color objectionable
Road:				
Asphalt	Limestone	Yes	Yes	Lower cost
Concrete	Cement	Yes	Yes	Lower cost; improved product
Oil aggregate	Limestone	Yes	Yes	Lower cost
Roofing	Filler	Unknown	No	No market
Roofing	Slate granules	No	Yes	Too fine
Rubber	Clay	Questionable	Yes	Seems to have possibilities
Soap mechanic's	Pumice	Unknown	Yes	Small market; dark-colored soap
Thermal insulation:				
Insulating cement	Calcined clay	Yes	Yes	Improved product
Loose fill	Mineral wool or vermiculite	Yes	No	Can be used only in the cells of masonry building blocks

Since the thermal insulating value k is very sensitive with respect to the unit weight, the maximum compressibility of the bulk fly ash is important. Again, there are a number of variables, such as original condition of the ash and dimensions of the test specimen, that must be considered in interpreting the results.

Using a cylinder 13.54 in. diam (1 sq ft) and 1 ft high, it was found that with rodded ash the over-all height could be reduced 1.10 in. under a total load of 300 lb. With aerated ash, this reduction increased to 2.15 in. These results indicate that even under a loading of 300 psf, the unit weight would not be more than 48.5 lb per cu ft. If no vibration were present, this might be so, but unit weights of 54.7 lb per cu ft have been produced by vibration alone without any external pressure. On the basis of these data, it appears probable that loose ash in a wall or similar container would acquire a unit weight of at least 50 lb per cu ft.

Chemical Properties

As previously stated, most of the investigational work dealt with the physical properties of fly ash. All of these have been reported and the unusual properties discussed. Earlier in the paper a rather elaborate chemical analysis of fly ash was presented. Fly ash has a few chemical properties, independent of the chemical composition, that warrant discussion.

Certain types of materials are known as "pozzolanas." A "pozzolana" can be defined as a siliceous material that, although not cementitious in itself, contains constituents that at ordinary temperatures will combine with lime in the presence of water to form compounds that have a low solubility and possess cementing properties. Since volcanic tuffs, burnt clays and shales, and pumicite are pozzolanas, it is logical that fly ash should have pozzolanic properties inasmuch as all these materials have a similar chemical composition and have been subjected to a

calcination temperature. Evidence that fly ash is a pozzolana is shown in an indirect manner later on in the paper where concretes containing fly ash are discussed. If increase in compressive strength of concrete is a measure of pozzolanic activity, then the results, shown in Fig. 4, clearly demonstrate that this fly ash is a pozzolana or has pozzolanic properties.

Sales Program

Following the accumulation of these rather complete data on the properties of fly ash, a canvass was made of the potential users located within the area of economic shipment. If possible, a personal call was made to discuss the matter with both the purchasing and technical departments. If information was requested on specific properties of fly ash that had not already been obtained, an attempt was made to furnish it to the prospective customer either by work in our own research laboratory or by sponsoring investigational work in an outside organization. If there was any possibility of substituting ash for a material being used by the customer, a sample of ash was provided. Often these interviews resulted in requests to supply ash in a modified form. Wherever possible this was done. The methods employed to provide the processed ash will be described in detail later. A subsequent attempt was made to learn the results of the experiments using the ash. Sometimes, however, it was impossible to get any worth-while information. The results of this sales program are given in Table 4. The conclusions to be drawn are as follows:

1 There are only a few industries that can use the quantity of ash produced by a large steam power plant.

2 The only reason for attempting to develop the small-consumer market is for greater diversification of sales and to improve the yearly load factor inasmuch as the requirements of the large consumer are apt to be very seasonal.

3 In general, the ceramic industry is not an outlet for ash because ash does not improve the product and clay can be delivered in the plant at a cost below that at which it would be possible to supply ash, except under unusual circumstances.

Modifying Processes

Sometimes it appeared that a prospective customer might be able to use ash if it were modified in some way. The usual requirement was that certain coarse fractions or certain chemical ingredients be eliminated. These cases led to investigations of commercial ways for modifying the ash by some process.

Air Classification. It was found that, by passing ash through an air classifier, a filler material could be obtained which would meet the physical requirements of the rubber and plastics industries. The results of air classification are shown in Table 5.

In doing this work no adjustments were made to the air classifier other than those possible on standard equipment. It is believed that possibly a larger recovery of fines in one processing could be obtained with a small amount of experimentation in adjustments. It should be noted that (1) the fines of Run A meet the physical requirements for rubber and plastic fillers, (2) the tailings of Run A meet the physical requirements for asphalt filler, (3) the fines of Run B meet the requirements for asphalt filler, and (4) inasmuch as the tailings from Run B contain approximately 20 per cent carbon, they might be returned to the furnace as fuel. Since these tailings represent less than 15 per cent of the ash produced, it is doubtful that the dust-collecting equipment would be overloaded by the additional burden.

Carbon Removal. Freedom from carbon is a requirement in many instances, chiefly because of the color effect mentioned previously.

Several ways of removing carbon or reducing the carbon content were investigated but none was found to be satisfactory from both the technical and the economic viewpoints. While none of the modern flotation methods was investigated, some elementary work on this type of processing was done in the laboratory, based on information from Germany, in which it was claimed that the carbon content, which was high in the case discussed, could be reduced by a flotation method using an oil-water emulsion. Our limited work indicated that (1) with carbon contents of approximately 10 per cent no carbon reduction was obtainable, (2) the loss of the flotation medium was excessive, (3) the final product would not have been satisfactory for the intended use even had the carbon content been satisfactory.

The production of carbon-free ash by the continuous-ignition process proved to be technically possible but not commercially

TABLE 5 SOME PROPERTIES OF AIR-CLASSIFIED FLY ASH

Sample		As collected	Run A		Run B[a]	
Sieve Analysis (Tyler Sieves)						
Sieve fraction		Per cent by weight				
			Fines	Tailings	Fines	Tailings
	Retained on 16 mesh	0.0	0.0	0.0	0.0	0.0
Passing 16 mesh	35	0.1	0.0	0.8	0.0	1.2
35	100	3.3	0.0	6.2	0.8	18.9
100	150	3.0	0.1	5.3	1.8	14.5
150	200	4.6	0.3	7.0	3.9	16.5
200	325	8.6	1.3	11.3	9.1	17.8
325	400	7.0	2.8	9.0	9.0	8.9
400		73.2	94.8	59.6	74.6	21.2
	Unaccounted for	0.2	0.7	0.8	0.8	1.0
	Total	100.0	100.0	100.0	100.0	100.0
Per cent recovered in each fraction			45.6	54.4	76.7	23.3
Chemical Properties						
Carbon content, per cent		10.64	7.68	13.75	11.67	19.28
pH value		7.95	7.80			

[a] Run B is a reclassification of the tailings of run A.

TABLE 6 RESULTS OF CARBON REMOVAL BY SOLUTION METHOD[a]

Sample no.	Quantity of fly ash digested	Quantity of digestive agent used,[b] ml	Carbon in sample, per cent by weight	Ash dissolved, per cent by weight of sample	Carbon removed, per cent by weight of sample	Carbon remaining, per cent by weight of sample	Carbon removed, per cent by weight of total carbon
1	5	50	21.82	12.45	6.60	15.22	30.24
2	5	100	21.82	19.69	13.61	8.20	62.40
3	5	200	21.82	26.67	20.37	1.43	93.40
4	5	300	21.82	27.96	21.36	0.46	97.89
5	5	400	21.82	33.84	21.44	0.38	98.30

[a] Time of digestion varied from 15 min for the smallest quantity of solution to 30 min for the largest quantity.
[b] Chromic acid.

practical. Using an externally fired, rotating, inclined kiln, it was possible to reduce the carbon content from approximately 6 per cent to less than 0.5 per cent. The kiln was operated at approximately 1500 F, and the output was 25 lb of ash per hr. It was found that the introduction of a small quantity of air at low velocity greatly aided the carbon reduction. However, the high fuel cost and limited output possible from this type of equipment prohibit its becoming commercially useful. Approximately 1500 lb of ash were processed in this equipment.

Another attempt was made in the laboratory to remove the carbon by mixing air and ash, both of which had been preheated, in an externally heated chamber. For operating temperatures up to 1800 F, it appears impossible to reduce the carbon content below 2 per cent with this arrangement of equipment.

Further efforts were made to remove the carbon by solution methods. While a number of acids could be used as the reagent to perform this function, our work was limited to the use of cleaning solution (chromic acid). The results are given in Table 6. These results show that, in addition to carbon, other ingredients, probably iron, are removed by this treatment. The fact that the residue retains its white color even after heating to 1800 F in an oxidizing atmosphere, further substantiates the conjecture that the iron compounds are removed. Because of the high cost of processing and the substantial loss of material by this digestive process, it is believed to be uneconomical to treat fly ash with acids for carbon removal.

A final attempt to remove the carbon was made using a contact potential separator at the U. S. Bureau of Mines. Starting with a material containing 7.32 per cent of carbon, two passes through the separator produced a material containing only 1.3 per cent of carbon. The yield of low-carbon material was approximately 55 per cent of the ash charged. Combining certain fractions of high-carbon material and reprocessing them would undoubtedly increase the yield of low-carbon ash without substantially increasing the carbon content. This method appears to offer the possibility of producing a low-carbon and extremely fine material in one processing. In spite of the low carbon content, the material still possessed a light-gray color which is in distinct contrast to the buff-colored material of similar carbon content produced by burning the carbon. This process seems to have definite commercial possibilities for producing a low-carbon fly ash.

Iron-Compound Removal. The general chemical composition of the ash indicated that it could be used as a raw material in the manufacture of glass if the iron compounds were removed. With this in mind, various processes aiming to reduce the iron content were investigated, but none was found which accomplished the purpose.

As previously mentioned, under the process for reducing the carbon content by treating the ash with chromic acid, the iron as well as the carbon was removed. The cost of treatment in this process made such processing economically unsound even if the process yielded a satisfactory material. A subsequent laboratory investigation disclosed that solution methods using hydrochloric, sulphuric, and chromic acids failed in all cases to reduce the iron content below 6 per cent. The original material contained 9.10 per cent iron in the form of an oxide.

Failing to produce a material of low iron content by the solution method, commercial magnetic-processing equipment was investigated. In no case was it possible to produce a material with an iron content of less than 5 per cent.

On the basis of the equipment and processes now available it appears impossible to process fly ash to produce a material containing less than 1 per cent iron which is the maximum permissible for use in the glass industry.

Commercial Markets

As a result of this extensive amount of laboratory and field investigational work, what can be stated regarding the prospects of developing commercial markets that will yield some revenue and use sufficient tonnage to take a substantial portion of the fly-ash production?

Concrete. That certain fly ashes improve the quality of concrete is no longer a debatable question, and the real problem in the use of them in this field is one of economics and sales resistance. In order to obtain data on the value of fly ash in concrete made with the cements and aggregates available in the Detroit area, considerable laboratory and field investigational work has been conducted. This work, part of which was previously discussed, confirmed the findings obtained by other investigators that fly ash has pozzolanic properties.

The chief benefits to be derived from using fly ash in concrete containing pebbles, crushed slag, or stone as the coarse aggregate are (1) greater long-time strength, (2) improved workability, (3) greater resistance to attack by ground waters, (4) lower permeability, (5) lower cost, (6) less bleeding.

Pebble-sand and slag-sand concretes containing fly ash and cinder-ash concrete have been used satisfactorily on a number of projects of the author's company.

The type of concrete and manner of use are as follows:

Type of Concrete	Use
Pebble-sand-ash	Roadways, building structures, machine foundations, cable ducts
Slag-sand-ash	Roadway
Cinder-ash	Lightweight concrete

The fact that the use of fly ash in pebble-sand concrete makes possible a decrease in cement content without sacrificing strength is clearly shown in Fig. 5. As would be expected, because of the

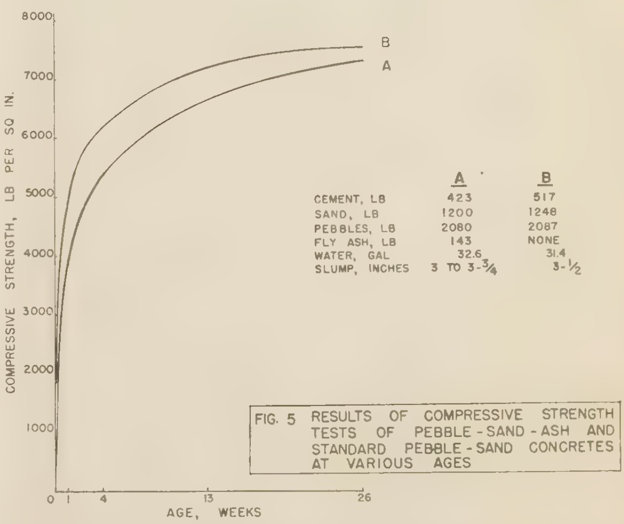

Fig. 5 Results of Compressive Strength Tests of Pebble-Sand-Ash and Standard Pebble-Sand Concretes at Various Ages

lower cement content, the pebble-sand-ash concrete had a lower early strength. However, after 6 months aging, both types of concrete reached approximately the same compressive strength value of 7400 psi, which is excellent concrete. Although in this case it took approximately 6 months for the two concretes to arrive at the same compressive-strength value, other tests have

shown that this point may be reached in a minimum of 28 days and a maximum of 1 year.

In an article[3] by F. R. McMillan and T. C Powers, a method is given for determining the relative values of admixtures and Portland cement in producing workability in concrete. Fly ash was one of the materials tested and in this case was approximately 11 per cent more effective than Portland cement in improving workability. No information was given as to the properties of the fly ash, but it is believed that its efficacy would be considerably influenced by the surface area, specific gravity, and possibly the carbon content. That the addition of finely divided materials is an accepted method for improving workability is evidenced by a statement taken verbatim from the article mentioned: "For example, if it is desired to bring about an improvement in workability, that may be done by enriching the mix, by correction of aggregate graduation, or by the use of almost any one of a host of available finely powdered materials."

Obviously, any material which not only is more efficient than Portland cement in producing workability but also contributes other valuable properties to the concrete should be given serious consideration, especially since the material is available at a fraction of the cost of Portland cement.

The presence of jagged and irregular particles in crushed-stone and to a greater extent in crushed-slag coarse aggregate leads to harshness and poor workability in concretes. The advantages, both as to workability and compressive strength, of using fly ash in slag concrete are shown in Fig. 4.

Recent permeability tests, conducted in the course of this investigational work on fly ash in concrete, were inconclusive in that it was not possible at pressures up to 90 psi to permeate sufficient water for accurate measurement. C. F. Ramseyer, formerly of the Chicago District Electric Generating Corporation, was more successful in his experimental work. In an article[4] reporting his results, curves are presented which show the relative permeabilities of different concrete mixes. The total water-loss permeation through a 28-day-old test specimen made from 6-bag per cu yd concrete was 340 cc at the end of a 7-day test period. The loss during the same period of time on a similar specimen, made from concrete containing 5 bags of Portland cement and 1 bag of fly ash per cu yd was 50 cc, a ratio of nearly 7 to 1. Low permeability is especially desirable in concretes exposed to frost action and ground waters.

From time to time there has been considerable discussion, both written and verbal, concerning the resistance of concretes containing fly ash to the deteriorating effects of alternate freezing and thawing. Some of the investigational work has been carried out on small specimens and it was necessary to screen out of the test-concrete mix part of the coarse aggregate. In order to approach field conditions as closely as possible, freezing-and-thawing tests were run on test specimens consisting of 8-in. concrete cubes from which no aggregate had been screened. The appearance of two tested cubes of pebble-sand-ash concrete is shown in Fig. 6. The results of this test and others indicate no reason to believe that concretes containing fly ash are particularly sensitive to freezing and thawing.

In certain locations, such as caissons or footings, substructures, and underground-cable duct, the resistance of concrete to the ground waters is very important. A great deal of investigational work has been carried on especially in Europe on the effect of ground waters on concrete. In a book[5] by F. M. Lea and C. H. Desch, a summary is given of the work of a number of investigators who have conducted experimental work on the effect pozzolanas have in increasing the resistance of concrete to the action of ground waters. In general, the results are favorable. In order to determine the effect of sulphur water on the concrete caissons of a powerhouse, an experiment was started by the company, with which the author is associated, approximately 15 years ago in which various concretes were exposed to the particular sulphur water in question. An analysis of this water indicates that, in addition to hydrogen sulphide, it contains sulphates commonly found in ground and sea water and is practically neutral, that is, neither acidic nor basic. Some of the concrete cylinders exposed to this sulphur water contain fly ash. The last progress report[6] on this test contains the following conclusions:

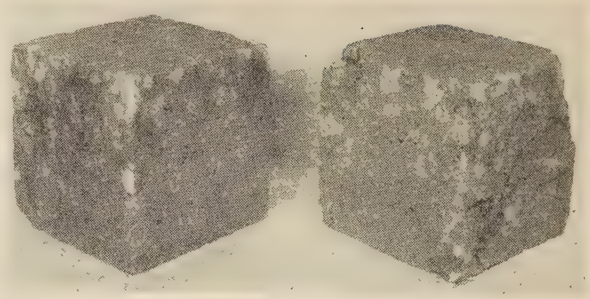

Fig. 6 8-Inch Concrete Cubes After Subjection to 225 Cycles of Freezing and Thawing
(4 1/2 bags cement, 150 lb fly ash, 36 gal water per cu yd concrete.)

1 The concrete should have a high cement content.

2 The concrete should have a minimum slump consistent with good placeability for a dense concrete.

3 The addition of fly ash up to 35 per cent by weight of the cement content is desirable.

This discussion regarding concrete has dealt with improving the quality of the product. In approaching a prospective user of fly ash, a sales argument that shows him how he can reduce his costs has much more appeal than one concerning improvement of his product. This is natural in that he is now selling his product and, therefore, believes it must be good enough for the job. The chance of making any substantial saving in the cost of concrete other than by a reduction in cement content is remote. Cement is a comparatively cheap material. Therefore, it is essential that there be a sufficient differential between the cost of cement and of fly ash to make it worth while to handle an extra material. Part of the delivered cost of fly ash is in the handling; this matter is discussed later on.

All the foregoing discussion has dealt with a standard concrete modified by the addition of fly ash. A new lightweight concrete is now available that uses fly ash in large quantities. It is known as cinder-ash concrete[7] and can be used wherever lightweight concrete is acceptable. All the floors and the roof in a six-story office building of a volume of 2,000,000 cu ft are constructed of this material. Before proceeding with the use of cinder-ash concrete in this structure, an extensive investigation was made of properties of the concrete. The results of this investigation are given in Table 7.

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[3] "A Method of Evaluating Admixtures," by F. R. McMillan and T. C. Powers, *Journal of the American Concrete Institute*, vol. 5, March–April, 1934, pp. 325–344.

[4] "Solving the Fly-Ash Problem," by C. F. Ramseyer, *Electric Light & Power*, vol. 17, February, 1939, pp. 44–47, 66.

[5] "The Chemistry of Cement and Concrete," by F. M. Lea and C. H. Desch, E. Arnold & Company, London, 1935.

[6] "Concrete Exposed to Sulphur Water," by J. S. Nelles (Progress Report on Tests), Proceedings of the American Concrete Institute, vol. 37, 1941 pp. 441–452.

[7] U. S. Patent 2,250,107.

TABLE 7 RESULTS OF INVESTIGATION ON LIGHTWEIGHT CONCRETE

Physical Properties of a $5^1/_2$-Bag Cinder-Ash Concrete

Weight, lb per cu ft	100
Modulus of elasticity (3000 psi compressive strength):	
At 20 per cent of compressive strength	$1.6–2.0 \times 10^6$
At 50 per cent of compressive strength	$1.5–1.7 \times 10^6$
Compressive strength, psi:	
At 7 days	1600
At 28 days	3000
Slump, in	6–7

Typical 1-Cu-Yd Batch Design Data

Ingredient	Weight, lb
Cement ($5^1/_2$ bags)	517
Ash (dry)	500
Cinders (24 cu ft)	1450
Total water (63 gal)	525

Sieve Analysis of Cinder Aggregate

Sieve fraction, retained on	Cumulative per cent by weight
$1^1/_2$ in.	1.70
$^3/_4$ in.	17.40
$^3/_8$ in.	41.70
No. 4	60.00
No. 8	72.50
No. 14	80.80
No. 28	87.00
No. 48	91.50
No. 100	100.00

The extensive use of this type of concrete would create a large market for fly ash because of the high ash content. The ash supplied for the cinder-ash concrete used in the building mentioned had about 25 per cent moisture by weight of dry ash. This in itself may seem a bit unusual and contrary to accepted practice. Early in this paper, it was stated that fly ash was a pozzolana and, according to the definition, a pozzolana was not cementitious in itself. Therefore there seems to be no reason that moist ash could not be used in all types of concrete. An investigation of this deduction disclosed that as good concrete can be produced with moist fly ash as with dry. However, it should be pointed out that the problems encountered in handling moist fly ash are quite formidable and any shipping advantages obtained from using the moist ash may be offset by local handling difficulties.

Bitumastic Road Fillers. Trinidad asphalt was at one time about the only bitumastic material available for road construction. Now, however, by-product materials from the oil refineries and the coke ovens provide a large amount of binder material. Trinidad asphalt is in fact a mixture of asphalt and mineral filler. From the chemical composition and the appearance of the particles, this filler evidently is pumice which is of volcanic origin. In the course of our investigational work in connection with the use of fly ash as a mineral filler in asphalt topping, an incomplete chemical analysis was made of the mineral filler in Trinidad asphalt. A comparison of certain chemical ingredients of fly-ash and Trinidad-asphalt-mineral filler is given in Table 8.

TABLE 8 PRINCIPAL INGREDIENTS OF ASPHALT FILLERS

Material	Composition, per cent by weight: SiO_2	Al_2O_3	Fe_2O_3	Total
Fly ash, as collected	45.9	26.9	9.1	81.9
Fly ash, burned free of carbon	51.4	30.1	10.2	91.7
Trinidad-asphalt filler	67.6	18.9	6.4	92.9

The use of fly ash as a mineral filler in bitumastic road construction is now definitely established in the Detroit area. This use was arrived at, however, only after an extensive period of experimentation and by overcoming a considerable amount of sales resistance. The fly ash is being used as a mineral filler in both asphalt and oil-aggregate or black-top types of roads.

The original installation of asphalt topping or wearing surface was installed on a heavily traveled thoroughfare in 1931, and it is still in use. Since 1939, approximately 1,325,000 sq yd of fly-ash-asphalt topping has been installed. Design data typical of this material are as follows:

Ingredient	Per cent by weight
Sand	76.7
Fly ash	13.0
Asphalt	10.3
Total	100.0

This mixture, when tested in a Hubbard-Field stability tester, gives a stability value of 1925 lb.

Oil-aggregate or black-top roads differ from asphalt topping in that the binding agent in the former is a slow-curing asphaltic oil, while in the latter it is asphalt. There is also a considerable difference in particle size and gradation of the aggregate, the oil aggregate having much larger particles and less mineral filler. Oil absorption by the filler is an important item in the cost and should be determined. The amount of absorption would undoubtedly be different for various ashes, depending probably upon the surface area and carbon content.

Rubber. In both of the large outlets for fly ash previously discussed, the delivery demands fluctuate widely and seasonally, there being practically no demand in the winter months. An attempt to improve this yearly load factor led to a rather extensive investigation of the possibility of substituting fly ash in some form for the clays and whiting now used as inert fillers in certain rubber products. It was realized that the "as-collected" fly ash would not meet the fineness requirements for rubber filler, but the air-classification process previously described did permit the recovery of approximately 50 per cent of the ash in a satisfactory particle size for use as a rubber filler. The estimated cost of classification was sufficiently low to make the process a commercial possibility. As previously mentioned, a considerable amount of ash was air-classified and a portion of this was burned free of carbon. Quantities of these materials were supplied for test purposes to a rubber company whose products were mechanical goods, such as floor mats, pedal covers, running-board mats, etc. The resulting rubber product was reported as having a tensile strength 10 per cent lower than that of the present material and of a darker color.

For the type of product being made, the limited color range did not seem particularly important to the author but it was just another hurdle to be overcome in breaking down the sales resistance. Considering the fact that, on the first trial, rubber was produced whose only limitations were color and a reduction of 10 per cent in the tensile strength, it does not seem that the door is completely closed yet in this field. Any company that is interested in the fly-ash-disposal problem should investigate the possibility of substituting processed fly ash for the present inert filler in low-cost mechanical rubber goods, because it affords a profitable outlet with a good yearly load factor and may also provide the rubber company with a satisfactory filler at a price advantage.

Handling Facilities

Because fly ash is usually sold in competition with low-cost materials, the desirable way of handling the ash is in bulk shipments. In some instances the ash must be handled in bulk because of the handling equipment available, while in others bulk use is necessary from an economic standpoint.

Fly ash has a very unfavorable reputation with respect to handling properties. In part, this reputation is deserved owing to the combination of small particle size and low specific gravity. When air becomes entrained, a light, fluffy, bulky product results, which flows very readily and leaks through very small openings. On the other hand when no air is entrained or when the material is allowed to settle for a long period of time, the ash is likely to pack hard in the bin and bridge over.

A large part of this poor reputation may be attributable to the attempts that have been made to handle fly ash with equipment

that was never intended to handle such fine material. Nevertheless, it is often the case that the equipment is already installed and, if fly ash is to be used, it must be handled with available machinery.

In using fly ash as a mineral filler in asphalt topping it was necessary to devise some means for handling the ash, as otherwise it would not be used even though the topping was technically satisfactory. A portable Fuller-Kinyon pump had been used for unloading bulk limestone dust from boxcars. This equipment, as built, could not be used to handle bulk fly ash. The problem was solved by delivering the ash in bulk-cement trucks, providing an adapter, and modifying the pump. This combination is shown in Fig. 7. However, it was found that trucks, which

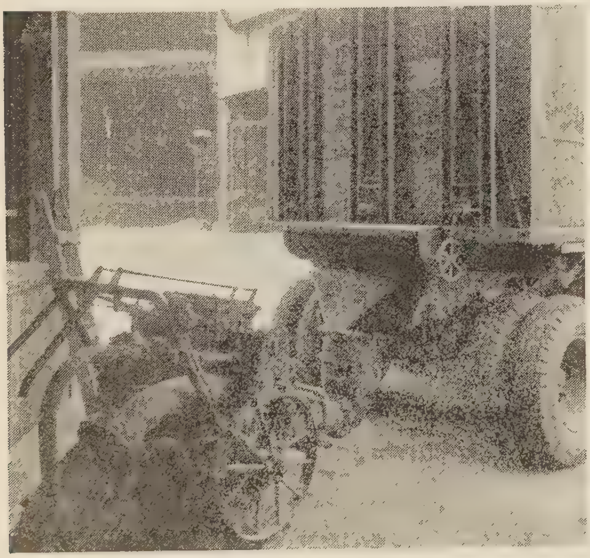

Fig. 7 Unloading Device for Bulk Fly Ash

were satisfactory for bulk cement, were not satisfactory for fly ash. The stop-gate construction at the rear had to be improved and holes in the body, that were not troublesome when the trucks were used for cement, had to be sealed.

At another asphalt plant, the ash is delivered in bulk cement cars and unloaded by means of a stationary Fuller-Kinyon pump located in a pit beneath the tracks. It is also possible to deliver ash by truck to this same plant, as shown in Fig. 8. If a portable Fuller-Kinyon pump is available, it can be converted to handle ash from the bulk-cement cars.

At the power plants, both the trucks and the cars are loaded from overhead bins through a rubber hose that is dropped into open hatches in the top of the conveyance. Between the bin and the hose line there should be installed a star-feeder to facilitate control of the rate of flow through the hose. A small vacuum breaker at the top of the hose line produces a much more uniform flow of ash than is obtainable if the hose line is sealed tightly. Because of the low specific gravity of fly ash and its tendency to aerate readily, the capacity by weight of bulk-cement conveyances is reduced when carrying the ash. This tends to increase the hauling charges. Laboratory investigations showed that by vibrating the receptacle with a commercial electric vibrator the unit weight of ash could be increased from 39.9 to 54.7 lb per cu ft, or 37.1 per cent, which would approach the bulk weight of uncompacted cement.

At a transit-mixed concrete plant and at an oil-aggregate batching plant, the ash is being handled successfully in bucket conveyers. The rate of discharge is somewhat slower than when handling cement or limestone dust. Precautions must be taken to keep the ash from bridging over in the feeding hopper to the conveyer, and better results are obtained if the speed of the bucket conveyer is kept below that recommended for use when handling cement.

At this point it is well to consider storage facilities. Previously it has been mentioned that the ash demand is seasonal for concrete and roads. Since both of these uses are dependent to some extent upon the weather conditions, there is a considerable fluctuation in day-to-day requirements. Also, if a contractor for a large project were considering the use of fly ash, it would be necessary to assure him that the ash could be delivered at the rate necessary to meet his requirements. These daily requirements might be considerably above the average daily ash output of the power plants. This means that adequate storage facilities must be provided, either by the producer or the consumer of the ash.

Of course, fly ash can always be furnished in paper bags similar to those used for Portland cement, but of a greater volume if it is desired to have the same weight of fly ash as cement. As a general rule bagged ash is supplied only to the small consumer. If a fly-ash producer should be so fortunate as to develop a large

Fig. 8 Conveying, Unloading, and Storage Facilities for Bulk Fly Ash at an Asphalt Plant
(Unloading bulk fly ash from a truck with the same unloading unit as used with a bulk cement car.)

number of small customers, it might be well to consider a delivery unit that would be self-contained, that is, the customer would provide the bin, piping, and possibly compressed air, and the remaining equipment necessary to discharge the ash into the bin would be a part of the truck or semitrailer. Such an arrangement would eliminate the expensive bagging operation, cut the cost to the consumer, and decrease the handling cost.

In interviewing a prospective customer, it is essential that one be able to discuss intelligently means of handling the fly ash. This is important because in a great many cases the bad reputation of ash is already known to the customer, and, unless proper facilities are provided for handling the material, the results will be unsatisfactory.

Other Possible Uses

Being of such fine particle size, fly ash has not been considered a satisfactory ground-fill material. Nevertheless, the experience of the author's company with several fills along the water front indicates that, if properly handled, filled-in land of this sort can be made capable of sustaining coal piles 35 ft high. In one case, the fill, which in places contained 15 ft of fly ash, part of which was submerged in water, has sunk about 1 ft after several years of storage of coal.

An attempt was made to produce a satisfactory fill material by combining fly ash with an impure limestone that is available in large quantities as a by-product from the soda-ash and acetylene-gas industries. The production of a suitable fill material was unsuccessful because of the difficulties encountered in mixing the two ingredients mechanically. However, a material was produced that was quite comparable, in resistance to disintegration and compressive strength, to well-designed soil-cement stabilization mixtures.

In Table 4, dealing with possible uses of fly ash, reference is made to the use of fly ash as a filler in agricultural fertilizer, and the reason for its being unsuccessful. Fly ash, however, does produce certain beneficial effects in heavy clay soils. That is, it is a soil-amendment agent. Such an agent may not have any fertilizing value but does benefit the soil by improving its mechanical properties. When thoroughly mixed with heavy clays, fly ash, being a highly vitrified material, tends to decrease the tendency to bake hard, crack, and ball up, and to increase the capacity to hold moisture. Heavy clays may require an addition of 50 per cent by weight of fly ash in order accomplish to the desired results. It is realized that this market offers very little opportunity for revenue, but it might eliminate haulage charges for dumping. Since the fall and winter months are the ideal time to place the ash on farmland, such an outlet would fit in nicely with the bitumastic road and concrete requirements.

Conclusions

As a result of this extended investigation, conducted both in the laboratory and in the field, it appears possible to arrive at certain general conclusions regarding the fly-ash market.

1 There are only a few industries that can use the quantity of ash produced by a large steam power plant.

2 The ceramic industry, which is a large consumer of clays, is not likely to become a user of fly ash because of the cheapness with which clays are delivered to the ceramic plants and because the ash does not generally have the proper properties to improve the quality of the product.

3 Certain types of fly ash are satisfactory as mineral fillers in bitumastic road construction.

4 Certain types of fly ash improve the quality of standard concrete, make possible a new lightweight concrete, and also open the way to a possible reduction in the cost of concrete.

5 The seasonal load factor of the mineral-filler and concrete markets makes it desirable to develop a year-round outlet of large tonnage such as in the fields of rubber or plastic fillers.

6 Regardless of the merits of fly ash as a material, it cannot be sold commercially unless the customer's handling problem is satisfactorily solved.

7 There will be no fly-ash-disposal problem if the sales resisance of the concrete industry and the bitumastic road industry can be overcome.

Radio-Frequency Technology in Wood Application

By G. F. RUSSELL[1] and J. W. MANN,[2] TACOMA, WASH.

As a basis for understanding the principles involved in applying radio-frequency heating to the resin-bonding of wood, the author explains certain fundamentals of radio theory which are involved. A particular quality of the radio-frequency field is its selectivity which results from the tendency of the field to follow the most conductive path between electrode extremities. Originally operations of bonding materials by this means placed alternate layers of wood and glue in the field so that the glue lines were established parallel with the electrodes causing the field to be perpendicular to the glue line. More recently it has been established that the normal resin-glue line is more conductive than the wood with which it is associated, and the technique of "parallel bonding" in which the field is parallel with the glue line has been developed. Details of the process, application of radio-frequency heating to specific cases, most efficient electrode arrangements, and results attained are treated comprehensively.

Technical Background

CERTAIN fundamentals of radio theory are essential to a proper understanding of radio-frequency heating effects. An electromagnetic field of force evidences itself through rapid alternations of charge on electrodes which constitute elements of the output circuit of a radio-frequency generator. When one electrode is charged highly negative, the opposite electrode is relatively highly positive. The next half-cycle results in a reversal of this relationship with the negatively charged electrode becoming the positive, and the positive becoming the negative. The electrodes with their connecting inductance constitute a simple oscillatory circuit. The space between the electrodes constitutes the part of the circuit wherein the dielectric is placed for heating.

When air is the dielectric between the electrodes, no loss takes place in the area. When other dielectrics are placed between the electrodes, however, a loss will result. When an energy-absorbing dielectric is introduced into a radio field of force, a phase shift takes place according to the standard laws

$$E\, I \cos\theta$$

which measures the total of real power. If wood of a certain density is introduced, a certain resistance will result. If wood of another density is introduced another phase shift will result. The energy absorption of the dielectric employed in the field has a major bearing upon the resultant phase shift.

Between the electrodes acting as boundaries of the capacitance portion of the circuit, there exists a half-standing wave of electromagnetic energy. This half-standing wave is a balance and a supplement to the half-standing wave resident in the inductance portion of the same circuit. When no resistance exists between the electrodes, there is a 90-deg out-of-phase relationship between voltage and current in the capacitive area established by the electrode boundaries, while the out-of-phase relationship in the inductance portion is in proportion to its ohmic resistance.

Radio-frequency waves are analogous to sound and similar to light waves. Each has the peculiar quality either of standing or traveling. The fundamental difference involved here is between the standing and the traveling wave. It is this difference which led to the discovery of and proof that there exists a half-standing wave in the capacitance portion of an oscillating circuit, and a half-standing wave in the inductance portion.

In electrical engineering language where there is no energy absorption in a circuit operating on 60 cycles, a 90-deg out-of-phase relationship exists between current and voltage. When resistance is introduced into this 60-cycle circuit, a phase shift results, and power loss may be calculated effectively by the vector method which shows total average power expended. This vector engineering is satisfactory when a traveling wave is considered to be active in the circuit study. It fails to show the placement of power loss when a circuit involving a standing wave is considered, because it only shows a total of the whole effect. The fundamental difference between a traveling wave as studied by vector diagrams and the standing wave which must be studied by chart diagrams is that the traveling wave averages its effect over a long distance, while the standing wave does not average its effect but in another manner maintains fixed and identifiable positions of energy conversion. The difference between the traveling-wave concept and that of the standing wave is observed to be the difference between averages of the whole effect and instantaneous values as fixed positions of power loss.

To return now to the half-standing wave between the electrodes of an oscillating circuit, it can be noted that the values are not averages, as would be the case of a vector analysis, but result in fixed but varying effects throughout the area wherein resistance is effective on the field of force. When a problem of heat placement is considered, the vector method of analysis might lead to the belief that heat distribution, as a result of natural forces, would be uniform. In this new approach, namely, the analysis of heat distribution by standing-wave instantaneous values, placement of the heat effect in its nonuniform distribution is almost exactly determinable.

Most investigators dealing with this problem of dielectric heating have discovered that greater heat exists in the center of a package of dielectric than on the outer edges of such a package. On the basis of analyzing the effect through vectors and averages, it naturally might be assumed that the less heating effect on the outer edges of the package is caused by radiation through the electrodes and through the outer atmosphere where there is a gradient of heat between the block and outside area. When the analysis of heat distribution is made by standing-wave method, the fallacy of even heat distribution in the radio-frequency field of force becomes apparent. Several simple steps show nonuniformity of heating effect in the radio-frequency field, the

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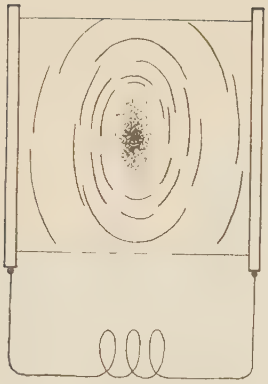

Fig. 1

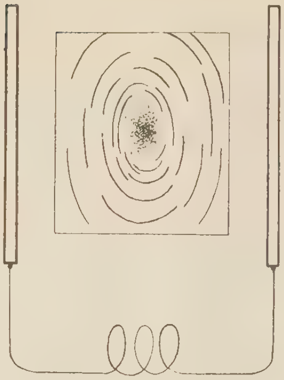

Fig. 2

Fig. 3

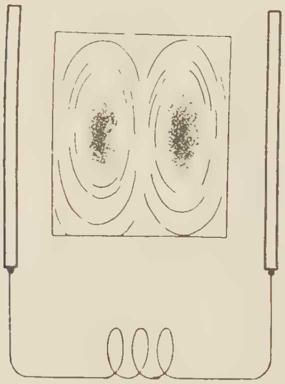

Fig. 4

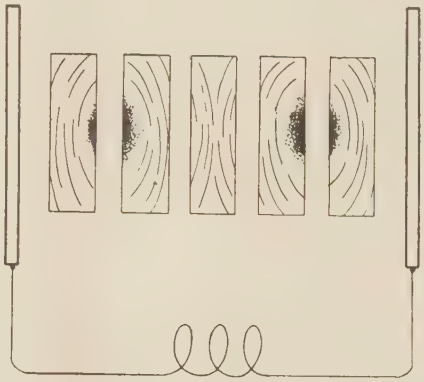

Fig. 5

first being to burn a single block in the center with electrodes contacting the outer face of the block as in Fig. 1. A second step is to separate the electrodes from the side of the block and burn it in a similar manner, as in Fig. 2. If radiation from the outer edges were the proper answer to the uneven burned effect of the dielectric, this could readily be proved by placing three blocks of dielectric in the field of force and having each separated from the other by an equal air space. This effect results in the center of the three pieces of dielectric being charred to an extreme, and the inside faces of the outer two grading down, leaving the outside faces of the outer two blocks unaffected by charring, as in Fig. 3. If the radio-frequency field of force were uniform, the three blocks would each be burned, respectively, at their centers.

It might be argued, at this point, that radiation still has its effect even on the three separated blocks and this, in part should not be denied, and so our studies should be carried a step further and show the effect of smaller phase shifts than occur in the examples of Figs. 1, 2, and 3. Producing this result shows a double burned area, with one burned spot on the left of the center and one on the right of the center, with the center and the two outsides remaining unaffected by charring, as in Fig. 4. The burned areas are the direct results of a shift in phase of less than that which causes the center burn in the other blocks. The conclusion of the series of experiments brings into the field five separated blocks with burned areas in the blocks next to the center one and also next to the outside ones, as in Fig. 5, reproducing the pattern shown in the single block, Fig. 4. All of the burns shown in the five illustrations are the result of double burned areas; but in the first three the two spots are merged into what appears as a single spot, the phase shift being greater and in phase relationship between the electrical components being closer than in the last two.

When a phase angle approaches 90 deg, but some energy absorption is effective, the two burned areas will be found at one quarter and three quarters of its thickness. When the phase angle approaches 0 deg and maximum energy absorption is effective, the two burned areas merge to form a single peaked area at the center of the thickness. In this latter case, heat distribution approaches the square of the instantaneous values of a half-standing wave of current. Returning therefore to the original formula for power, it can be said that $E\ I \cos\theta$, which shows total power does not show the distribution of a radio-frequency field of force, and that the relationship of power distribution to dielectric materials varies in uniformity with the adsorptive qualities of the load, and the effectiveness of the dielectric to produce a resistive effect.

The foregoing discussion is offered as in opposition to the uniformity theory, not to conclude that all that has gone before is wrong, but to see another side of the matter which results from a study of the physics of wave structure and electron motion, with the admonition to think twice before too much is assumed, based upon the vector engineering of 60-cycle alternating current.

The fact that the heating effect of the radio field can be predetermined; that its heat effect can be controlled; that its effect through the thickness of the material, while not uniform, is instantaneous and simultaneous makes it a new and useful tool for industry in many wood, plastic, and other applications. However, the primary concern in this discussion is with adaptations

to wood treatment and glue-setting technique in wood which stem from some of its unique, nonuniform, and selective qualities.

Gluing Technique

A particular quality of the radio-frequency field is its selectivity. This results from the tendency of the field to follow the most conductive path between electrode extremities.

Original operations of bonding materials with radio frequency placed alternate layers of wood and glue in the field so that the glue lines were established parallel with the electrodes. This means simply that the established lines of force between the electrodes would be perpendicular to the respective glue lines. There being no one path between the electrodes, under this situation, which would be any more conductive than any one other path, it was perfectly natural to bring the entire package of wood and glue up to the critical setting temperature of the glue for purposes of bonding the mass. It has been incorrectly assumed that it made little difference whether the glue lines were perpendicular to or parallel to the lines of force.

Parallel Bonding. Now it has been determined, however, that the normal resin-glue line is much more conductive than the wood with which it is associated, so we can adapt this knowledge of conductivity and selectivity to gluing technique and place the glue lines in the field so that they substantially parallel the main lines of force rather than exist perpendicular to them. This technique of gluing is called "parallel bonding" and differs materially from the older technique of "perpendicular bonding" in that it allows the concentration of the entire effect of the field on the glue joint, avoiding the necessity of wasting much heat in bringing the wood up to the same critical temperature as is required to set the resin glue. In some cases it had been found that in thick laminations only a small portion of the heat from the radio field is required to set a given package by the parallel-bonding method as was required to set an equal passage by the perpendicular process.

It should be pointed out however that, as the critical setting temperature of the resin is increased to a point approaching 212 F, the effect of parallel bonding diminishes, and that if the critical setting temperature of the resin is substantially above the boiling point as is the case of high-temperature phenol resins, little if any advantage remains through the use of parallel-bonding methods, as compared with perpendicular methods. This is due in part to the fact that much energy will be expended from the hot glue lines by conduction to the wood itself before the temperature reaches 212 F, and also in part to the fact that the moisture in the wood will be acted upon equally with or to the exclusion of the conductivity of the glue lines above 212 F. The selective effect of the radio field therefore has led to interesting and advantageous methods of gluing wood and the treatment of other materials.

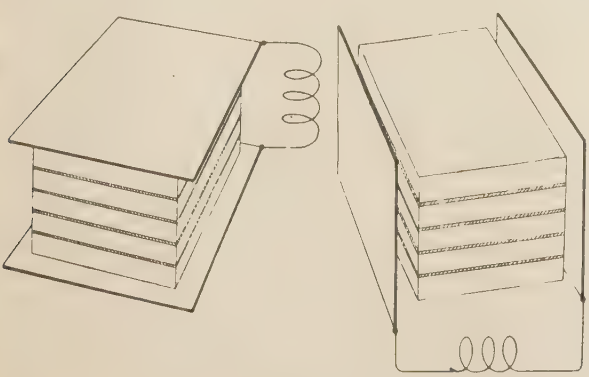

Fig. 6

Resins. This selectivity may best be utilized by parallel bonding techniques, using ureas, melamines, and the very new types of phenol resins, which are in time, cold-setting, or at least set below the boiling point. Heat will accelerate practically any resin adhesive, so if a cold-setting resin is used heat will accelerate its chemical reaction and complete polymerization in rapid order. Rather than spend time describing glues, our primary interest is in applications of varying techniques and electrode arrangements. It is one job to create a radio-frequency oscillator, but it is still another to apply it properly to a gluing operation. Fig. 6 illustrates the two general types of electrode arrangement used in most operations; the sandwich or perpendicular method and the parallel bonding method. The differences between these types has been described but some general principles can be illustrated from them.

Applying the Radio-Frequency Oscillator

In both methods, pressure is applied to the top and bottom of the package, the electrodes in the perpendicular method being between the package and the pressure sources. Surplus glue may squeeze out, fall downward toward the bottom plate electrode, and if the spread is not very carefully controlled to prevent this squeeze-out, the glue excess will start a high-frequency arc and carbonize a path between the electrodes which not only is dangerous as it reflects back into the oscillator or amplifier, but it may easily ruin the entire batch of material. In parallel bonding, since the electrodes do not have to contact the work, squeeze-out as a danger is entirely avoided.

In determining which method to use for a specific job, frequency of the package, which is dependent upon its size, shape, moisture content, and glue is sometimes a determining factor. With a fixed-frequency oscillator, of course, less flexibility in package dimension and electrode placement is permissible than with a variable-frequency oscillator. The latter types are variable, even so, only over the range of their variable capacitances. The lamination of ten plies of veneer $^1/_8$ in. thick with $^1/_8$-in. cauls above and below in a size 7 in. $\times$ 14 in., for example, can be treated at different frequencies, but its fundamental, using the sandwich method with plattens above and below, is in the neighborhood of 25 meters or 12 megacycles. With parallel-bonding electrodes on either side of the same-size package, its frequency fundamental is 10 meters or 30 megacycles. In a scarfed joint the same size and thickness, the fundamental frequency is slightly higher, due to a lesser content of resin glue. Frequency of oscillation, or wave length, undoubtedly has its effect upon speed of setting and effective heat penetration and conversion, but the study by Von Hipple and Deitz was inconclusive in this regard. It has been said that the higher frequencies are more effective in heat generation than are the lower ones, which in general may be accepted as founded on some logic, but it is believed that power input and conversion efficiency of the oscillator have a greater effect upon heating than does the variation of frequency within limits, for example, between 10 and 50 megacycles. At the lower frequencies, which is to say, under 6 megacycles, some noticeable reduction in heat-conversion efficiencies should become apparent.

Almost any wood-bonding load can be matched, one way or another within the range of between 10 and 50 megacycles. Simpler apparatus at these wave lengths is just another reason to go higher and higher in oscillating frequency. But limitations of power, because of the capacitances of the vacuum tubes, prevent large packages being operated upon at very high frequencies. Some day these limitations will be breached, and progress in that direction is currently very rapid.

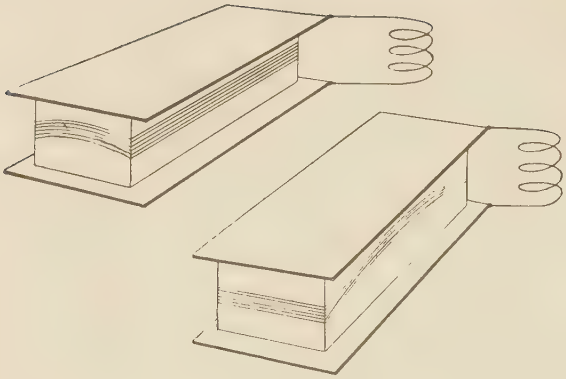

Fig. 7

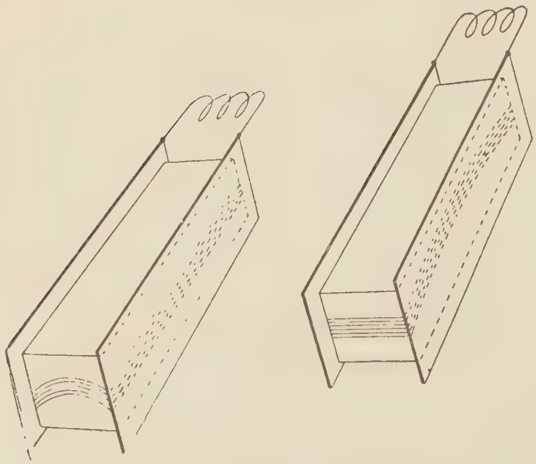

Fig. 8

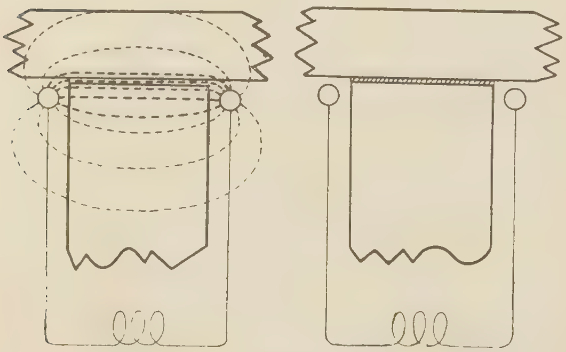

Fig. 9

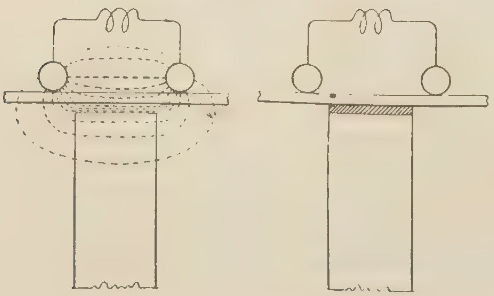

Fig. 10

Fig. 7 shows a perpendicular bonding setup which employs the advantages of and also approaches the effectiveness of parallel bonding. Here curved shapes may be bonded between male and female forms. Fig. 8 shows these two same curved shapes set up on a pure parallel-bonding method. The time of bonding using the same input power will be about one fifth as long with the parallel bonding means as with the perpendicular means, with a glue which sets below 212 F.

Fig. 9 shows an accessible butt joint with electrode arrangement and field distribution. This is a simple means of bonding joints quickly. Fig. 10 shows a similar application, but in the case of a thin skin abutting an inaccessible rib or other member. The field distribution shows how the energy seeks the conductive

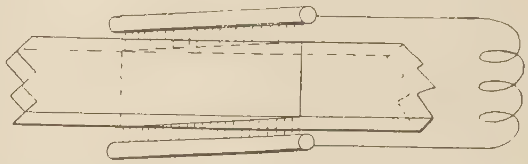

Fig. 11

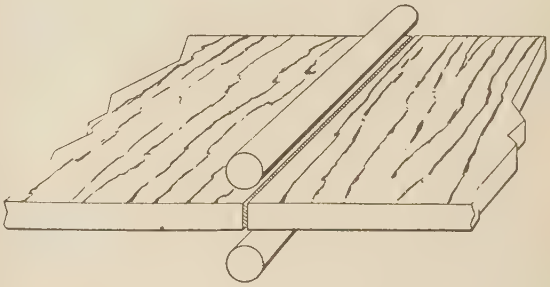

Fig. 12

glue line through the skin surface and accelerates its setting even through relatively thick sections. This application is one means of spot-welding wood or sewing thermoplastics. No matter what the accessibility of the joint, so long as no metal parts parallel the field of force and detract from the conductivity of the glue line, some method can usually be worked out to accomplish the job, and some system of electrode arrangement devised to supply properly radio-frequency energy to the joint.

Most Efficient Electrode Arrangements

A straight scarf, such as is employed in aircraft work shown in Fig. 11, when pressed from top and bottom, presents no problem when the scarf is under 6, 8, or even 12 in. in width. Rods or tubes placed along either side of the scarf, their centers aligned with the glue line and supported by standoff insulators, suffice for electrodes, either center or end fed. The field lines of force literally spit through the glue line in this setup but care should be taken either to space the electrodes away from direct contact with the glue lines, or to place a thin wood or other dielectric caul between them. An air gap when not too wide, say, $^1/_2$ or $^3/_4$ in. on either side of the scarf, will cause no material slowing of the reaction of the glue, as compared to using a thin caul and no air gap. A bar or tube rather than the conventional flat plate, as shown in Fig. 6, may be employed in order to concentrate emission from the surface closest to the glue line. Stray and therefore wasted field lines of force are minimized with rods, but are greatly increased when an electrode area of great width is presented to a single glue-line application.

The principle of presenting the minimum of electrode-surface

area to the glue line is well illustrated in Fig. 12, which shows a tube-electrode application to edge-glued lumber or veneer, employing the parallel bonding method. This means is employed on a continuous machine being developed to take the place of the tape splicer and also the hot-plate tapeless splicer used for core work in plywood, veneer, and furniture manufacture. It is anticipated that we will shortly see in operation such a machine running at a speed of between 50 and 100 fpm.

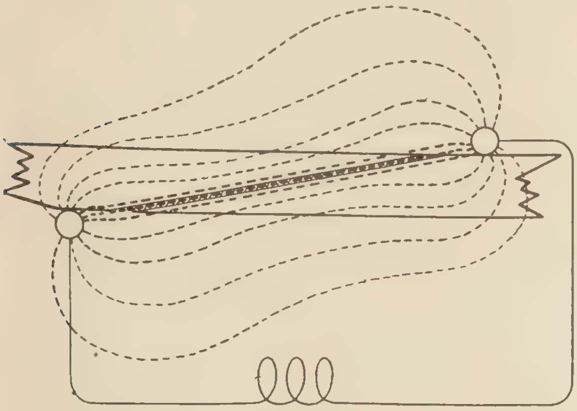

FIG. 13

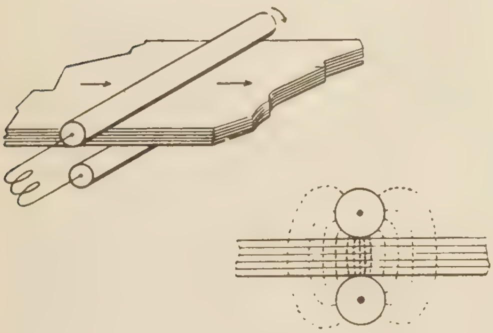

FIG. 14

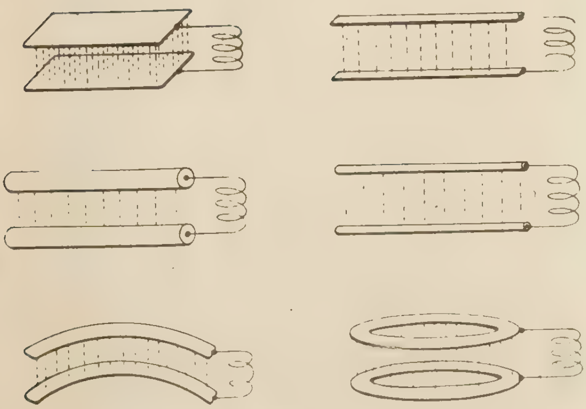

FIG. 15

Fig. 13 illustrates parallel bonding applied to a scarf joint which is very wide, for example, in the 4- or 8-ft side of a sheet of plywood. These joints, 10 or 12 in. through and 4 ft long, are such as those used on the bottom of PT boats. It would be impractical to apply an electrode arrangement such as shown in Fig. 11, because its frequency would be much too high to come within the range of any currently used radio-frequency machines. It might even reach 300 or 400 megacycles or even into the microwave strata. Long tubes can be used running the entire length of the scarf and still take advantage of the parallel bonding method. These electrodes, supported by standoff or other insulators, run the entire length of the scarfed joint. Using tubes 3 in. diam and 50 in. long, as shown in Fig. 13, a frequency of 15 megacycles, and a wave length of 20 meters, creates a nice operating load. When these electrodes are extended to longer lengths and the width of the scarf widened also, the frequency is maintained at 15 megacycles.

For certain operations such as bonding plywood on a continuous basis roller electrodes are employed successfully. With one roller above and another below the package, the established field lines operate perpendicular to the glue lines. Such an operation, as shown in Fig. 14, not only applies to plywood bonding, but to the drying of wood, the treating of cellulose-plastic compounds, the dehydration of food products, and many other uses.

By reference to the several illustrations, possible electrode arrangements are shown for several different uses. These are but a few, and to show others would be to repeat largely the lessons evolved from the various types of placement. The shape of electrodes may vary even more widely than their relative placement. Fig. 15 has six separate shapes, plates, strips, rollers, bars, curves and circles, all of which with equal efficiency may be adapted to their special uses.

It might be asked how the two-spot heating effect, or even that of the single spot affects gluing procedures. On loads of relatively light power absorption or power factor, as some call it, the two-area heating will be encountered in parallel bonding. In perpendicular bonding, no such light load is apt to be encountered. The two hottest areas, in a light resistive load, will spread their temperature rise rapidly to adjacent glue-line areas, and in a flash which is so fast as almost to prevent stop-watch timing, distribute enough heat to polymerize the resin in the entire glue line.

One very interesting case occurred in bonding a small scarf joint $1^1/_2$ in. wide and 4 in. long in $^1/_4$-in. spruce aircraft stock, using tubular electrodes, and the method shown in Fig. 11. A 2-sec exposure to the field of a generator having an output rating of 2200 Btu per hr cured the two outer edges and into the scarf some distance, but when broken immediately after release from the press an uncured center showed up about $^1/_2$ in. through the glue line. This indicates quite clearly the two-spot heating effect. When exposed 5 sec, the entire scarf-joint glue line was cured throughout. Normally one might not encounter usual evidences of the two-spot heating effect, but it is there nevertheless and should be considered in gluing light resistive loads.

An Introduction to Aircraft Hydraulic Systems

By HOWARD FIELD, JR.,[1] INGLEWOOD, CALIF.

This paper traces the evolution of hydraulic systems for airplanes, from the hand-actuated piston pump and "motor" piston, through the stage of motor-driven gear pumps, down to the present-day systems which derive their power directly from the airplane engine. Three general methods of pump control have been developed which permit rotation of the pump but prevent build-up in the hydraulic system, since the pump operates continuously. These methods include "the constant-pressure," the "manual by-pass," and the "automatic by-pass," all of which are in wide use. Special applications of hydraulic control are described in the "power-amplifier" or "load-feel" system, the "follow-up" system, "follow-up plus load-feel" system, and the "full-automatic" system.

THERE are many means of transmitting power from its source to the place where it is to be used, among which are electrical, steam, mechanical, hydraulic, etc. Each has advantages and disadvantages which become of greater or lesser importance depending upon the particular problem at hand.

Having no regard for relative importance, and in fact, the relative importance is not constant, the advantages of hydraulic systems in aircraft are as follows:

1 Light weight per horsepower required.

2 Low inertia of moving parts, making quick starts and stops possible without undue stresses being set up.

3 Controllability to any degree of accuracy and sensitivity required.

4 Great flexibility of installation not only because tubes can be run anywhere but also because the same system can develop great forces or light rapid motions.

5 Ability to withstand large amounts of abuse without complete breakdown.

6 Reliability greater than many other competing forms of power transmission.

Since the hydraulic system is a method of transmitting power, it is obvious that there must be a means of applying the power at one end of the chain, in other words, a pump. This pump may be either hand- or power-operated, the simplest form being a cylinder containing a piston which may be manually moved. Next, a line or tube to convey the power to the output end of the chain, and a motor to reconvert the hydraulic power to mechanical motion and force are required. This "motor" may likewise be a simple cylinder and piston. If this "motor" piston be arranged to be returned by a spring, and a reservoir to take care of leakage or volume changes be added, the result is the well-known automobile hydraulic brake. Many small airplanes use this identical system for brake actuation, as shown in Fig. 1. It will be noted that none of the diagrams in this paper makes any pretense of showing the actual physical embodiment of the circuits or devices. They are intended to be only the simplest possible representation of principles. In many cases, undesirable devices have been represented rather than more desirable but more complicated ones, which would be more confusing. In general, no packings have been shown, although they are obviously necessary in the actual devices.

By substituting an ordinary double-acting pump for the simple master cylinder, a double-acting cylinder for the brake cylinder, and adding another line and directional-control valve, the resulting system (diagrammatically shown in Fig. 2) is complete. This was the first system used in aircraft for actuating mechanisms other than brakes. There are many control-valve designs, but all accomplish the same end. These valves are usually known as "selector valves," or "4-way valves."

The next step was to add another unit to be operated from the same pump and reservoir. This can be done by having the second unit in series or in parallel with the first. Since the series arrangement requires the use of tail rods or different sizes of cylinders to provide identical motions it is not used for the following reasons:

1 Tail rods require extra packings.

2 Different sizes complicate manufacture, and they might be improperly installed.

3 There is no way to bleed the trapped portion of the circuit between the two cylinders.

4 Thermal changes in volume or slight leakages would throw the cylinders out of synchronism.

5 The pressure is divided between the two cylinders; hence it is necessary to make them large and consequently heavy.

There are ways to overcome faults 3 and 4, but series cylinders

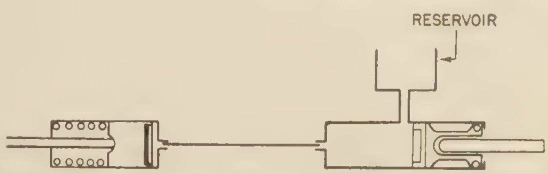

Fig. 1 Schematic Form of Simple Hydraulic-Brake System

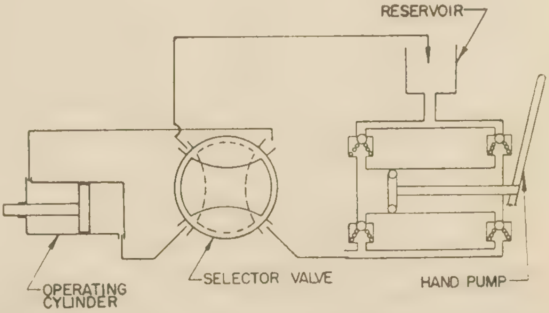

Fig. 2 Diagram of System in Which Double-Acting Pump and Cylinder With "Selector Valve" Are Substituted for Simple Form of Fig. 1

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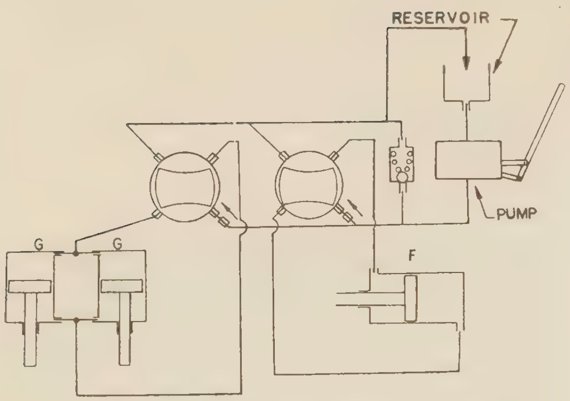

Fig. 3 System With Pressure-Relief Valve Added

are never used unless there are overwhelming reasons for so doing. All five of these difficulties are avoided by connecting the cylinders in parallel, although this introduces a new problem. The cylinder which, for any reason, moves easiest moves first. It is seldom that this causes actual difficulties and, where it does, corrective steps can be taken. For example, it may add aesthetic value to have both wheels of an airplane retract in unison, but it is not usually essential that they do so.

The addition of a relief valve to prevent excessive pressures from being built up in the pressure lines results in a circuit like Fig. 3, and consideration must be given to what would happen if the pilot should open both selector valves at the same time. In any system, there is a tendency for all potentials to seek a common level, and a hydraulic system is no exception. Assume that an airplane is coming in for a landing and the pilot pumps his flaps down; this may require 600 psi in the flap-actuating cylinder (cylinder F in Fig. 3). The pilot moves his landing-gear selector valve to "gear down." The gear falls part way by gravity, which requires no pressure, so the fluid runs out of the flap cylinder and into the landing-gear cylinders (cylinders G in Fig. 3). Since there is nothing holding the flap down, it moves to a trailing position, which reduces the lift and may result in a crash. Of course, if he had lowered his gear first, this would not have happened, but the simple addition of check valves at the fluid-pressure ports of the selector valves will prevent unwanted interflow between selector valves, although this introduces another difficulty.

Hydraulic fluid expands about $4^1/_2$ per cent per 100 deg F (much more than the metallic parts expand), and the check valves block off any chance of relieving excess pressures due to thermal expansion in the cylinders and lines through the relief valve. This can be overcome either by making the check valves "leaky" so they will permit the very slowly expanding fluid to pass, or by putting additional relief valves at each selector-valve pressure port. Circumstances dictate which method shall be used. The first would let the flaps creep up slowly (disadvantageous but not usually critical), while the second entails manufacture of more relief valves. Strangely enough, even though very simple, good relief valves are among the most difficult hydraulic parts to make.

General Power System

As airplanes grew in size, it became evident that some sort of power would have to be applied to hydraulic systems, since a pilot cannot be expected to produce more than about $^1/_6$ hp, and that only for a limited period of time. It was also true that as airplanes grew larger, the pilot had more to do and no longer had time to spend in purely manual labor. When airplane systems required 1 or 2 hp, it seemed a logical development to apply an electric motor to drive the hydraulic pump, which was generally of a positive-displacement type, such as a gear pump. This entailed the addition of a relief valve to provide a path for unused fluid to return to the reservoir.

Further experience showed that the pilot neglected entirely to turn off the motor when the operation was completed, and the system continued to run, heating up the oil. It then became necessary to connect a pressure-operated device to the switch so that, when a predetermined pressure was reached, the switch would operate automatically. Of course, the relief valve was set higher than the amount of pressure which was expected to be used. Fig. 4 shows the resulting power circuit. This automatic pressure "kick-out" device relieved the pilot but it had faults of its own. For example, the pressure had to be set very much higher than required for an operation or the switch would kick out when accelerations acted on the landing gear. Thus the system had to be designed for much higher pressures which shortened the pump life and made everything heavier.

No good solution was ever found for this condition because other factors increased the power requirements so fast that it became impracticable to draw the power from the airplane electrical

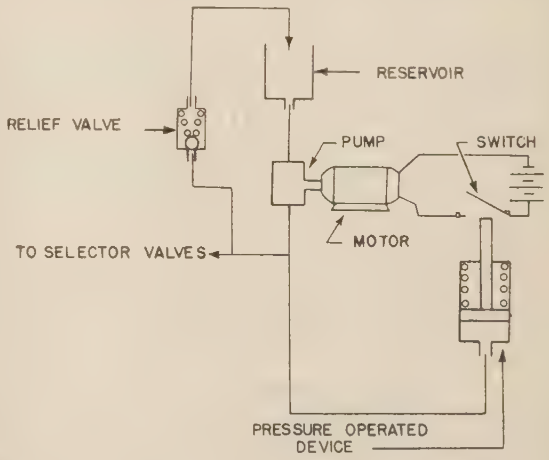

Fig. 4 Hydraulic Circuit With Electric Power Added

system. The obvious answer was to connect the pump to the engine, and thus to use the engine power directly, instead of indirectly through the storage battery and motor. With the pump driven by the engine, there was no such thing as turning it on when required unless some sort of clutch device were interposed between the engine and the pump. The idea of using the clutch was abandoned, as investigation showed that this would be cumbersome and heavy and would probably lead to service difficulties. Thus it was necessary for the pump to operate continuously whenever the engines were running, which meant that fluid was always being pumped.

By-Pass Methods of Relieving Hydraulic Systems

One method of relieving the hydraulic system and the pump when it was not required was to by-pass the fluid which was being pumped all the time the engine was in operation. This general scheme is the one used in almost every hydraulic system at the present time. This method of by-passing the pump may be subdivided into three general classes:

1 The class, known as the "constant-pressure" system, may in turn be subdivided into two classes:

(a) This class utilizes a relief valve to limit the pressure. In

this system, the fluid when otherwise not being used, constantly runs through this relief valve, which is undesirable, since the power put into the fluid by the engine-driven pump is wasted or turned into heat. Although this type of "constant-pressure" system is sometimes applied, it is limited to approximately $^1/_5$ hp because of the large heat-rejection problem.

(*b*) A variation of the "constant-pressure" system which is used to some extent, particularly in Europe, is one in which a variable-delivery pump is connected to the engine. This pump is so constructed that, as the pressure which it produces is increased, the amount of oil or fluid which it pumps is decreased. Thus when the pressure rises to some predetermined point, which is somewhat above that required for operation, the volume of fluid delivered falls to zero and no further pressure increase occurs. The pressure in the system remains at "cut-out" values, but since there is no volume being moved, the power absorption and consequent heat rejection is zero, except for the power required to overcome pump friction.

2 A method of relieving the hydraulic system, known as the "manual by-pass" system, is usually semiautomatic in that the pilot turns on the pressure, but it is relieved automatically. This method may again be subdivided as follows:

(*a*) An application, which uses a special by-pass valve for operation at some predetermined pressure, is utilized in the power system shown schematically in Fig. 5. The by-pass valve is constructed so that the pilot pushes the plunger "in" (to the left in Fig. 5), where it is latched. There it remains until the pressure in the release mechanism (shown as a cylinder and piston, restrained by a strong spring, in Fig. 5) builds up to a point where the latch is released, and the plunger is returned to an "open" position by the light spring behind it. The by-pass valve performs much the same function as the switch in an electrically operated system, in that the pilot closes it and it opens automatically. Unfortunately, it is subject to the same disadvantages as the electrically driven pump system because the pressure must be set at a higher point than is desirable, in order that the valve may not be operated by a surge in the hydraulic lines and thus stop operations which are not complete.

(*b*) Another scheme uses a special by-pass valve which opens a certain length of time after it is closed. This by-pass valve is shown in Fig. 6. This would connect to the general hydraulic system in the same manner as the by-pass valve in Fig. 5. In Fig. 6, a plunger is adapted to close the pressure line when it is pushed into its seat by the pilot. Then the pressure from the line exerts a force to push the piston to the right and withdraw the plunger, but an equal and opposite force is exerted by the fluid which runs through the check valve and acts against the right-hand side of the piston. Thus the hydraulic forces are in balance but the spring shown tends to move the piston to the right. In order to accomplish this, it is necessary that the fluid to the right of the piston escape to the return line. This is controlled by a properly chosen restrictor valve in the line connecting both ends of the valve. A number of trainer type airplanes use this scheme of operation.

(*c*) Yet another application of the "manual by-pass" system makes use of a special type of selector valve known as an "open-center" type. When in the neutral position, the fluid from the engine pump circulates through the valve and back to the reservoir. When any of the valves is displaced to operate some unit, this by-pass is shut off and the fluid is then directed to the proper operational device. The early versions of this open-center type depended upon the pilot returning the valve to neutral position when the desired operation was completed, and again the pilot forgot to do so, which left the oil at full pressure and being relieved through a pressure-relief valve. To avoid this difficulty, pressure-operated devices were arranged to return the valve to

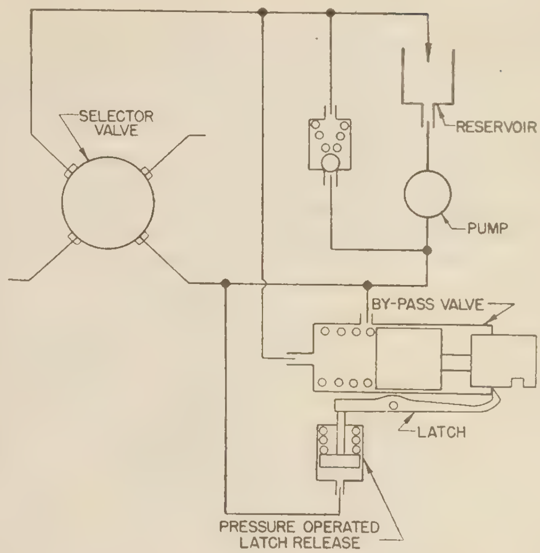

Fig. 5 Manual By-Pass System of Pump Relief With By-Pass Valve Operating at Predetermined Pressure

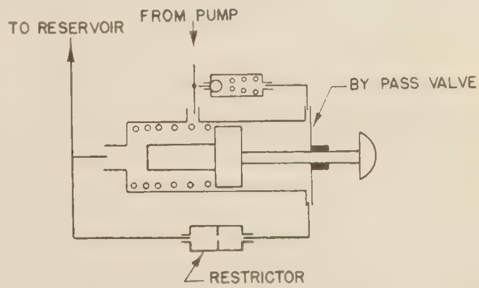

Fig. 6 By-Pass Valve Operating on Time-Lag Principle

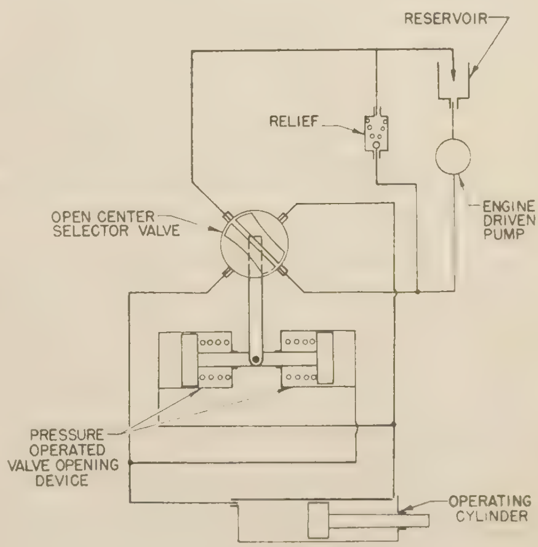

Fig. 7 Manual By-Pass System of Pump Relief Employing "Open-Center" Type of Selector Valve

neutral automatically when the end of an operation was reached. Schematically, this system is shown in Fig. 7. Here again, the pressure must be set higher than is desirable in order to keep the valve from being moved to neutral by pressure surges in the system. Although somewhat undesirable, this system is in use in some airplanes at the present time.

3 Another general method of relieving the pump is known as the "automatic by-pass" or "unloader" system. This method is now used on practically all airplanes built in the United States. The heart of the system is an "unloader" valve which is shown diagrammatically in Fig. 8. This valve is placed in a circuit with

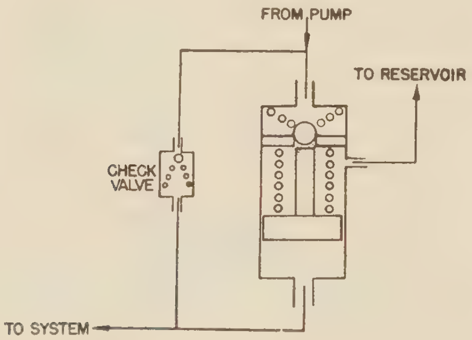

FIG. 8 DIAGRAMMATIC REPRESENTATION OF PUMP RELIEF BY MEANS OF "AUTOMATIC BY-PASS" OR "UNLOADER" SYSTEM

the fluid from the pump entering the valve as shown, and also flowing through the check valve and to the general hydraulic system. A small ball, seated in the partition across the valve body, forces the fluid to flow through the check valve and to the general system, since the ball prevents the oil from going through the valve seat and out through the side port and back to the reservoir. The pressure in the system is carried to the bottom of the piston which is shown as having an extension reaching the ball valve. The pressure acting below the piston tends to lift the ball from its seat. This is resisted by two forces, (1) the heavy spring whi h tends to push the piston downward and (2) the pressure acting on top of the ball tending to hold it to its seat. As the pressure increases, a point is reached where the pressure below the piston overcomes the spring and lifts the ball from its seat, thus losing the pressure which tended to hold it down, consequently the piston moves up rapidly opening the valve wide. Now, the fluid from the pump has free passage to the reservoir and, consequently the pressure drops. Obviously, the check valve, shown at the left, is required; if it were not for the check valve, the pressure would also drop in the rest of the system which would permit the piston again to return to its original position and thus close the ball valve, resulting in a system that was constantly opening and closing.

In this system an additional supply of fluid must be provided to react against the bottom of the piston in order to keep it moving in an upward direction. Without this extra fluid supplied below the piston, the piston would move up only to a point at which the ball would rise off its seat a sufficient amount for the pressure loss through it to reduce the pressure below the piston to a degree that would cause the valve to stay in that position. To overcome this, an elastic device, which is known as an accumulator, is used in the line to the system. Originally, accumulators were simply cylinder-piston combinations which were loaded by means of heavy springs, but, since operating pressures increased to the neighborhood of 1000 psi, this type of accumulator became impracticable because of the excessive spring weight. Later, accumulators were designed which used compressed air in place of the spring. This proved much lighter than the old spring-loaded type and has been in continuous use since then.

The air-loaded accumulator consists of a chamber, usually spherical in shape, which is divided at its diameter by a diaphragm of flexible material. The chamber on one side of the diaphragm is pumped up to some initial-charge pressure, such as 400 psi, through an air valve, the diaphragm serving to separate the air from the oil which is admitted to the other side of the accumulator. Obviously, as the pressure in the fluid is increased, the air is reduced in volume so that it functions in the same manner as did the spring in the earlier design; but the air is practically weightless.

There have been numerous difficulties with this type of accumulator, most of which have been due to the requirements of the diaphragm material. It not only must be immune to attack by the fluid but also must have flexibility at very low temperatures. Improved synthetic-rubber materials have been developed which now make this type of accumulator entirely practicable.

A third type of accumulator, coming into wider use, is something of a combination of the two former types in that it consists of a cylinder having a piston to separate the oil and the air, but utilizes air above the piston instead of a spring. Early attempts to construct this type of accumulator were generally unsuccessful because of the difficulty of making packings which seal properly when the pressure on both sides are equal. Some of the newer type packings solve this problem satisfactorily.

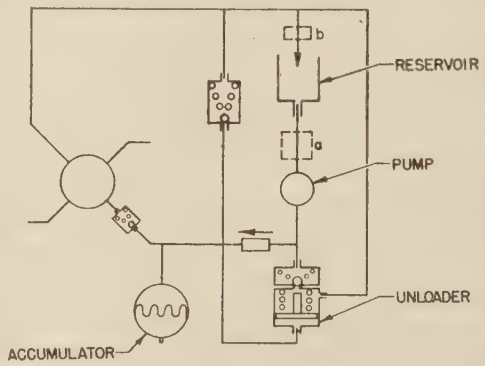

FIG. 9 DIAGRAM SHOWING UNLOADER VALVE AND ACCUMULATOR CONNECTED INTO POWER CIRCUIT

The unloader valve and the accumulator are connected into the power circuit as shown in Fig. 9. Here the pump draws the supply of oil from the reservoir and then discharges it to the upper portion of the unloader valve and also to the general system. The accumulator is shown connected to the general system at the left. It is obvious that if the selector valve, or valves, be closed, the oil delivered by the pump will first be used to charge the accumulator; and when that is accomplished, the pressure will rise to a point at which the unloader will open, thus permitting the pump to circulate the oil freely through the unloader valve and back to the reservoir. When a selector valve is opened, the pressure tends to fall, and the oil is drawn from the accumulator which starts the operation desired. As the pressure reaches the level at which the unloader valve again closes, the pump will discharge to the system and to the accumulator instead of back to the reservoir. Of course, when the pressure is again increased, the pump is again unloaded.

In Fig. 9 the dotted rectangles are shown as possible locations of filters. While location (a) is more desirable, since fluid is filtered before it goes to the pump, this location is seldom used because of the increased losses in the intake pressure at the

pump. If the filter be installed in location (*b*), this difficulty is avoided.

Special Power System

Among the advantages of the hydraulic system, as applied to aircraft, has been "greater flexibility of installation." Examination of one or two simple devices might illustrate this point. Suppose that the general hydraulic system was designed to operate at a pressure of 1000 psi, and then an additional device such as, for example, an automatic pilot which operates at a pressure of approximately 200 psi were added. By connecting a pressure supply for this device properly, it can be operated from the general system without endangering the system at all. Fig. 10 shows a method of connecting the reduced-pressure system into the general hydraulic system. Here, the unloader valve and check valve to the full-pressure system are exactly as shown previously. The check valves between the engine-driven pumps and the general system prevent by-passing of fluid through either pump in case of shaft failure. Between one pump and its check valve, what is known as a selective relief valve is inserted, and immediately above that a tap-off of fluid to the reduced-pressure system. Of course, a shutoff valve to prevent flow in this reduced-pressure system when it is not being used and an ordinary relief valve, connected to this reduced-pressure system to eliminate overpressures, are necessary.

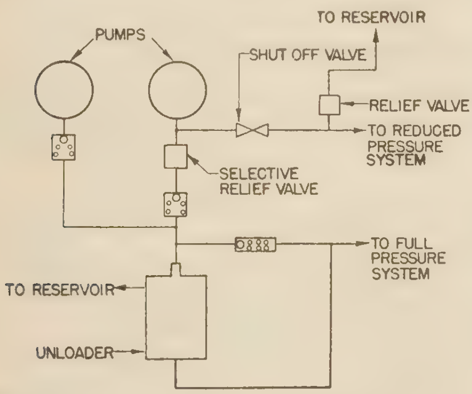

Fig. 10 Method of Connecting Reduced-Pressure System Into General Hydraulic System

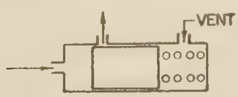

Fig. 11 Selective Relief Valve Diagram

The selective relief valve is somewhat similar to the ordinary relief valve and is shown diagrammatically in Fig. 11. Here, the fluid enters the valve from the left where it acts upon a piston. When the entering pressure is low, this piston covers up the outlet port and, consequently, there is no flow through the valve. However, as the pressure increases, the piston moves to the right a sufficient distance to make the loss through the valve equal to whatever pressure the spring was set to produce, as shown in the right-hand end of the valve.

The real difference between the ordinary relief valve and the selective relief valve is that the piston in the selective relief valve is packed and the rear of the piston is open to the atmosphere. Then the valve is unaffected by back pressures in the outgoing line, because these back pressures cannot act against the back of the piston. Hence it tends to hold the entering pressure to a definite figure or a definite amount above atmospheric rather than above return-line pressure; any higher pressure merely holds it wide open.

As this valve is installed in a system corresponding to Fig. 10, the oil from the right-hand pump enters the end port of the selective relief valve and discharges through the side port, through the check valve, the unloader, and the general hydraulic system. If the shutoff valve be opened to use the reduced-pressure system, this selective relief functions to maintain the predetermined pressure in the lines going to the reduced-pressure system, but it does not prevent the pump being used in the general or full-pressure system whenever the reduced-pressure system is turned off. It is, of course, true that all the time that the pump operates, it is operating against the pressure established by the selective relief valve. However, this is usually of so low a value that the pump can operate against it indefinitely. This results in a flexible arrangement in which both pumps can be used to operate the general system. Whenever the reduced-pressure system is being used, one pump is always available to the full-pressure system, and the other pump operates the reduced-pressure system.

This selective-relief-valve installation is shown merely as an example of what may be done with the hydraulic system in an airplane. It would be difficult even to begin to enumerate all the special conditions which are being provided for hydraulically in various airplanes at the present time. All sorts of sequence-operating devices are being hydraulically controlled, and many complicated actions have been made possible by proper valving. In many cases, it has been impossible to accomplish the desired result by any means other than hydraulic.

Special Control Systems

Some of the rather special systems which are used primarily because of the great degree of controllability or sensitivity are of interest. Among these are units in which the pilot must not only be able to direct the fluid or the power to the desired operational unit, but he must also be able to feel the load of the operational unit and to know its position at all times. Actually, four definite types of systems are used because of this great degree of controllability:

1 In the "power-amplifier" or "load-feel" system, the pilot supplies part of the energy for the operation, and the hydraulic system supplies the remainder. This system can be arranged so that the pilot supplies most of the power or, on the other hand, he may not supply more than 0.1 per cent of the power, but the part that he does supply is always in definite relation to the total amount required.

2 A second type of control is known as a "follow-up" system, which may be subdivided into mechanical and hydraulic types of positional transmission, both of which will be shown later. These are devices in which the pilot selects the position to which he wants some operational unit to move and, upon completion of this movement, the system automatically stops. Such a system is frequently used to operate landing flaps. In this case, the pilot merely moves his flap-control handle to whatever position he desires, say, for example, 30-deg flaps. The flaps move to that position and stop of their own accord. It is the fact that it is not necessary for the pilot to know what load is required to move the flap to the selected position, which differentiates the "follow-up" system from the "power-amplifier" system.

3 A type of control system which combines the two previous ones is usually known as a "follow-up plus load-feel" system. This is the type of system used for surface-control boosters. In these, the pilot must know not only in what position his control surfaces are, but he must also supply some of the power required to move the controls to this position. While it would be possible to handle control surfaces by means of a power follow-up system,

moving the controls very lightly from one position to another, he would have no conception of how hard it is to get them there. Unless the pilot has some idea of the amount of energy required, he is apt to force the ship into a more violent maneuver than it was designed to withstand, so the use of the "follow-up plus load-feel" system is indicated.

4 The fourth control type might be classified as the "full-automatic" system. Many different services are performed by the hydraulic system without any attention whatever from the pilot. A good example of these is the automatic temperature control for engines.

Application of the "Load-Feel" System

One of the most common examples of the "load-feel" system is the power brake valve. This form of valve control is shown diagrammatically in Fig. 12. The pilot does not want to know the

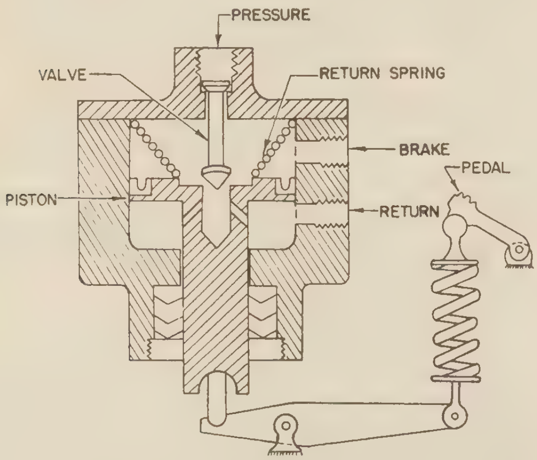

Fig. 12 Application of "Load Feel" System to Power Brake Valve

exact positions of the brake shoes or disks, the only thing which interests him is the pressure that is being applied to them. On smaller airplanes, the pilot was able to supply sufficient energy to operate the brakes without assistance from the main power supply. However, for airplanes having gross weights of about 10,000 lb or more, few pilots are physically capable of applying the brakes without assistance.

In the brake valve, shown in Fig. 12, the inlet for the pressure from the hydraulic system is at the top. Since the valve is shown in the closed, or brake-off position, this pressure is prevented from going to the brake by the cone-shaped valve closing the port. On the other hand, the brake line is open to the return port. This insures that the brakes will not be applied until the valve is moved. When the pilot applies the brake, he pushes down on the pedal as shown. This motion is carried through the brake-valve lever at the bottom of the figure and results in the upward movement of the piston in the valve. The first effect of this movement is to close off the return port by seating of the lower cone of the return valve, thus preventing any pressure admitted to the brake line from escaping through the return line. A very slight further movement of the piston in an upward direction will then open the pressure valve admitting pressure to the brake. As this pressure builds up, it will act downward against the piston in exact proportion to the pressure in the brake. This downward force on the piston is transmitted through the brake-valve lever and up to the pedal so the pilot feels the reaction on his foot. Obviously, if he pushes only lightly on the pedal, the pressure builds up in the brake valve to such an intensity that the piston reacts strongly enough to move his foot back a few thousandths of an inch, no more pressure passes into the brakes, and the result is a very light application of the brakes. As the pilot pushes harder on the pedal, the pressure in the brake valve builds up higher and greater brake action is achieved.

Notice that the actual movement of the brake pedal is very small, as a matter of fact, only a few thousandths of an inch. The valve acts perfectly. It does what is required but it feels unnatural to the pilot, since he has been accustomed to mechanically operated brake pedals in smaller airplanes, or to automobile brakes in which the pedals have a substantial movement. It does not feel natural to push on a pedal which is apparently as unyielding as a brick wall. To make this feel more natural to the pilot, a spring is introduced between the pedal and the piston. In Fig. 12, this is shown as though it were directly in connection with the pedal itself. This spring does not have any actual function in so far as the brake valve is concerned. It is merely introduced to make it feel more natural to the pilot. Here is an example of a power amplifier in which the pilot's first effort is amplified by the hydraulic system and then applied to the brake. There is apparently no particular limitation as to the amount of power which may be controlled by the brake valve, and it is to be expected that, as airplanes continue to get larger, the pilot will actually supply only about 1 or 2 per cent of the power required to apply the brakes.

Preselective Flap Control, Example of "Follow-Up" System

The mechanical version of the second class of control system, called the "follow-up" system, is illustrated in Fig. 13. This is a system which might be used as a preselective flap control. When

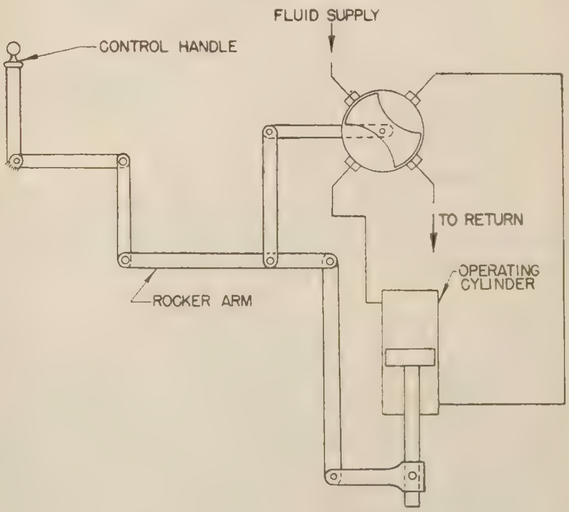

Fig. 13 Mechanical Follow-Up Type of Control System

the pilot moves his control handle to some position, the valve is opened and the resulting motion of the operational unit is transmitted mechanically to the valve through a rocker arm so that, when the desired position is reached, the valve has been brought back to neutral and the operation ceases. This mechanical position-transmitting type of "follow-up" system is seldom used, except where the power required for the operation is fairly large or where structural considerations make its application easy.

If the power requirement is low and installation difficulties prevent direct mechanical connections between the control handle

and the unit to be operated, a "follow-up" system having hydraulic positional transmitting connections may be used, Fig. 14. In this system, the operating unit, which is known as a "slave unit," is connected by lines to a follow-up cylinder and a valve whose body is fastened to the follow-up cylinder and moves with it. In the figure, the piston rod is shown as being anchored, and the valve lever is connected to a handle for pilot operation. If the handle be moved in the direction shown by the arrow, pressure will be admitted to the right-hand end of the follow-up cylinder. This pressure will push the cylinder itself toward the right, reducing the space in the left-hand end of the cylinder, thus forcing the fluid to flow through the pipe to the right-hand end of the slave unit. Since the slave unit is fixed, the piston will move to the left, and the oil returning from the left-hand end of the slave unit passes through the valve and is returned. It is evident that, as the follow-up cylinder moves to the right, it will move the valve body to the right, and, since the pilot's control handle is in a fixed position at that time, the valve will automatically

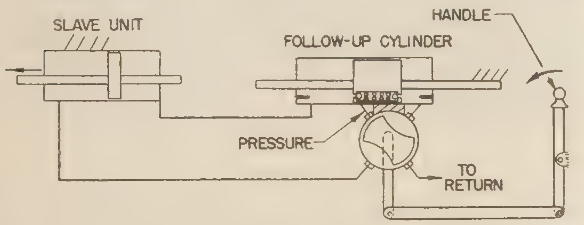

Fig. 14 Follow-Up Control System With Hydraulic Positional Transmitting Connection

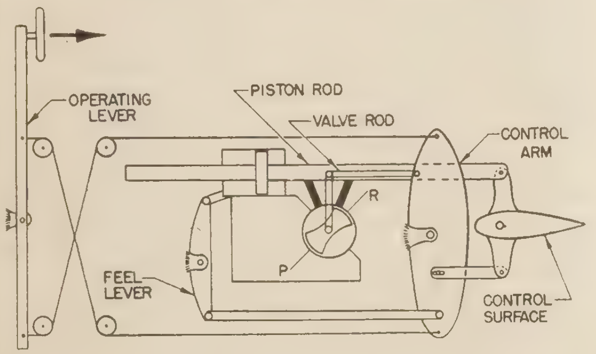

Fig. 15 "Follow-Up Plus Load-Feel" Type of Control Unit

close off when the position of the slave unit corresponds to that of the handle.

This type of circuit is, in effect, two cylinders in series and, in order to compensate for thermal expansion of the fluid or leakage, check valves are placed within the piston of the follow-up cylinder, and lifters are arranged at the ends of the cylinders so that whenever an extreme position either right or left is reached, one or the other of the check valves will be raised from its seat, permitting fluid to equalize between the follow-up cylinder and the slave unit and thus resynchronize.

"Follow-Up Plus Load-Feel" System of Control

The type of control known as the "follow-up plus load-feel" is used where the pilot must not only know the position of the controlled unit but also have some realization of the loads required to move it to that position. Fig. 15 shows such a system connected up as a surface-booster mechanism. The operating lever in the left-hand side is connected by cables to a control arm. The piston rod of the operating cylinder is connected to a horn attached to the control surface and, at the same distance from the center, a valve rod which operates the valve is connected about as shown. The valve body is fastened to the piston rod and moves with it. If the operating lever be moved in the direction of the arrow, that is, to the right, the control arm is pushed in the opposite direction. This moves the valve rod and opens the valve in a direction which will permit pressure to enter the right-hand end of the cylinder. This pressure entering will move the piston rod to the left thus moving the control surface in the desired direction. If the piston rod moves to the left, it carries the valve body with it and, since the valve rod is fixed in position, the valve will close when the piston has moved a sufficient distance to bring it into synchronism with the control arm. There is no need of having any connection between the control arm and the control surface, but a connection having lost motion is used because in the event of failure in the hydraulic system, the pilot would wish to have control of the airplane by manual means even though he might have to push very hard for the controls to accomplish the desired results.

Referring again to the operating cylinder, notice that it is not fixed to the structure but is connected through a lever and rod back to the control arm. The cylinder could have been fixed but, if it had been, an irreversible system would have resulted and the pilot would not have any "feel" of the loads required. By connecting the cylinder back through this lever and rod, the position of the pivot about which this lever moves can be so varied that the force transmitted to the control arm is in any desirable proportion to the effort which is exerted by the cylinder. If the pivot be put nearer the cylinder, the pilot will feel only a small portion of the effort.

It is, of course, obvious that the type of valve, shown in this diagram (Fig. 15) and also in the previous one, is not at all suitable for this use, since it is not sensitive in action and must be rotated through large angles. However, it was used merely for illustration since the principle is the same, and the rotary valve is much easier to understand in a diagram than the somewhat complicated but more sensitive valve which is actually used in practice.

Fully Automatic Type of Hydraulic Control

The final type of system is the fully automatic one in which the pilot has nothing to do with the operation. Such controls are used frequently as temperature controls, manifold-boost controls, or propeller controls. A fully automatic temperature control is illustrated in Fig. 16, which shows an operating cylinder, the piston rod of which is connected to a bell crank so that as the piston moves to the left, the scoop is opened wider. At the right-hand side is shown a rotary valve with its operating lever connected to a sylphon bellows which expands when heated. This bellows would be placed in the coolant circuit and, as the engine increases in temperature, the coolant temperature would increase and the sylphon bellows would expand. This would rotate the valve in a clockwise direction, admitting pressure to the right-hand end of the operating cylinder, thus moving the piston to the left and

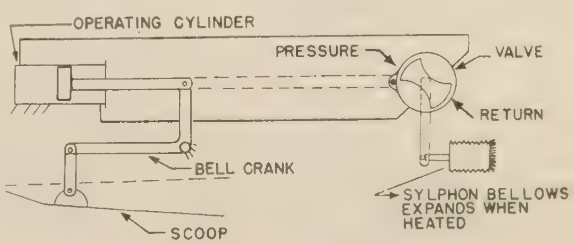

Fig. 16 Fully Automatic Temperature-Control System

opening the cooling scoop further. It is probable that the scoop would operate a great deal faster than the temperatures of the engine, and, consequently, the bellows would change so the scoop would be opened completely before the bellows had a chance to contract and close the valve. The scoop, being too far open, would overcool the engine and the sylphon bellows would contract, thus opening the valve in the opposite direction and closing the scoop. This would go on indefinitely, resulting in the scoop alternately opening and closing fully all the time.

Of course, restrictors might be placed in the lines so that the scoop could only move slowly. It is not advisable to control the hunting tendency in this manner since it limits the rapidity with which the scoop can be moved, and many times it is necessary to open a scoop quickly.

Another way of controlling this hunting would be to connect the valve body with the piston rod by a connection which is shown in dotted lines. Then, if the bellows heats up, the valve opens admitting pressure to the right-hand end of the cylinder and the piston moves to the left thus opening the scoop, but at the same time the valve body moves to the left which tends to close the valve. Now, if the engine should get warmer, the bellows would expand further, again opening the valve, and the scoop would open still further. Hence it is evident that the temperature of the sylphon bellows will have to be somewhat higher when the scoop is fully open than it would be when the scoop is nearly closed. In other words, a definite fixed temperature cannot be held regardless of the opening of the scoop. The relation between this range of temperature and the valve operation gives the system what is known as "drooping" characteristics.

Nearly all fully automatic controls require that there be some droop in the characteristics, in order to avoid hunting. The amount of droop depends upon the particular application and circumstances. It is expected that many more fully automatic controls will be used in airplanes in the future as designers become familiar with the simplicity of these automatic devices and learn that the most important feature about them is to select properly the amount of droop in their characteristics.

The discussion of various types and details of aircraft hydraulic systems might be continued indefinitely, although the basic ideas have been shown.

The detail design of hydraulic units and systems is based upon just three factors: (1) an understanding of the principles which have been illustrated, (2) a knowledge of fluid mechanics, and (3) experience. Most of the difficulties in connection with present-day hydraulic systems are due to the inexperience of the designers of the particular system or, in some cases, lack of test information, both of which are inherent in such a new and rapidly growing field of endeavor.

Controversy Over the Choice of a Medium for Aircraft Power Transmission

By RICHARD L. HAYMAN,[1] BURBANK, CALIF.

The growth in size and the increase in speed of modern aircraft has created the need for powered operation of many controls, accessories, etc., to supplement the work performed by the crew. The services performed by engine power and remotely transmitted to various parts of the airplane are too numerous to mention here but are tabulated in part in the paper. The choice of a medium for the transmission of this engine power to remote points has provoked one of the most intense controversies of aircraft engineering. In spite of the bitterness of the controversy that has at times prevailed, it has probably proved to be a dominant factor in the improvement of the two major surviving media, namely, hydraulics and electricity. This paper sets forth the cause of the controversy and presents a fair evaluation of each system of power transmission.

Introduction and General Explanation

Most powered operations on aircraft can and have been performed by either the hydraulic or the electrical method, although perhaps not as conveniently one way as another. The choice actually resolves itself down to an evaluation of the comparative advantages and disadvantages of these two media for the particular job at hand. Certain companies subscribe almost completely to all-hydraulic operations; others cling to all-electric operations, while the majority hover between the two. This would verify the fact that opinions differ widely as to the proper evaluation of the services. To understand this radical difference in opinion of sound engineering minds one must study the complexity of the problem. This paper will attempt to present some aspects of the controversy in its present stage of development.

The problem is rightfully divided into two parts: (1) the transmission of power from one point to another, and (2) the conversion of this power into useful work. While no attempt will be made to treat these parts separately, the division should nevertheless be borne in mind.

Three Basic Forms of Power Transmission

The three basic forms of power transmission (other than purely mechanical) in common use in modern aircraft are hydraulic, electric, and pneumatic (including vacuum). Of these three basic services, electricity alone is an absolute necessity. Lighting and communications are two chores that make the use of electrical power mandatory. Since it is desirable to keep the number of different types of power media to a minimum, this fact is a strong argument for the use of all-electric power.

The use of vacuum and compressed air is relatively minor at this time in this country and will be neglected in this paper. Certain instruments have utilized vacuum for their operation for some time. The vacuum pumps necessary for this work have served double duty by providing the low-pressure air used by the inflatable wing-deicer boot. The use of electrically operated gyros and heat deicers may make obsolete these services. The use of hydraulics is universal, however, in spite of the fact that most of the services performed by it could be done electrically. To understand this we must enumerate the requirements of a power medium.

A brief explanation of the three basic services follows:

Aircraft hydraulics utilizes a low-viscosity mineral oil, and operates (in this country) at pressures ranging up to 3000 psi. Aircraft electricity consists of 12 and 24-v d-c systems, 110-v a-c systems, and an experimental 208-v, 3-phase, 400-cycle a-c system. Pneumatic power ranges from vacuum air systems to high-pressure air systems.

Choosing a Medium for Power Transmission

The Requirements. If the requirements for a power-transmission medium were to be listed in the order of their importance, they might read as follows:

1 Reliability (including lack of vulnerability).
2 Functional suitability.
3 Low weight (including efficiency).
4 Simplicity (from both development and maintenance standpoint).
5 Low cost (in terms of time and material now; in terms of money after the war).
6 Compactness.

There are probably others that could be added under separate headings, but we will try to include all requirements under one of those listed. The order of importance of the requirements may vary with the size, type, and function of the airplane and is therefore subject to change. All of the requirements listed necessarily overlap to a certain extent, so that it is impossible to discuss one without referring to others. Thus reliability and simplicity will go hand in hand. Simplicity and functional suitability will combine, and simplicity and cost will be contested as being synonymous. Low weight and compactness will fall into a similar category.

Reliability. The relative reliability of electrical and hydraulic systems has been strenuously contested verbally, but very little

TABLE 1 UNITS FREQUENTLY OPERATED BY CYLINDERS OR MOTORS IN LIEU OF MANUAL OPERATION[a]

Power Plant	Flight Controls
Starter	Surface-control power boost
Cowl flaps	Dive flaps or brakes
Oil-cooler flaps	Tabs
Prestone radiator flaps	Wing flaps
Carburetor hot-air valve	Automatic pilot
Carburetor air-filter control	Slots
Two-speed supercharger contr	**General**
Propeller-feathering pump	Landing gear and landing-gear door
Propeller deicing pump	Floats
Fuel booster or assist pump	Folding wings
Armament	Brakes
Gun chargers	Steering
Ammunition boosters	Fans (heating and ventilating)
Elevation boosters	Cabin supercharger
Gun turrets	Winches (for loading, etc.)
Bomb doors	Dump valves and chutes
	Landing lights
	Deicer pumps and controls

[a] The list is not necessarily complete.

[1] Hydraulics Engineer, Lockheed Aircraft Corporation.

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conclusive proof has been brought forward to substantiate a claim for either medium. Actually it is extremely difficult to make a good comparison and arrive at any definite conclusion. In every case unreliability of one or the other is a function of poor design, installation, or maintenance. Since, however, we will always be limited by human ability to perform these functions, we can do no more than take actual cases as a parameter to compare the two services.

In an endeavor to make an evaluation of any usefulness, it was felt that actual service records should be explored. Service reports on types and makes from many different companies, covering a period of several years, were studied toward this end. In every case of failure of electrical or hydraulic equipment or systems, the seriousness of the failure naturally depended on the nature and the importance of the units being powered. Both systems have necessarily gone through a period of strenuous development. Aircraft hydraulics is newer than aircraft electricity and, although lacking the standardization that electrical-parts manufacturers have achieved, concedes no ground to electrical systems in reliability.

We find in practically every case that trouble was due to lack of experience, either on the part of the equipment manufacturer, the aircraft manufacturer, or the ultimate operator of the aircraft. Since the use of these auxiliary powers to such a great extent in aircraft is really quite new, lack of experience is natural. Both systems went through similar epidemics of trouble. In the hydraulics field at certain times, there were numerous failures of regulators, accumulators, ball-check valves, and packing. Sheared pump shafts provided another flurry of trouble and less predominant were hose failures and control-valve failures. In the electrical field, the same period was marked by frequent generator, voltage-regulator, inverter, limit-switch, and starter failures, and burned-out or broken wires or plugs. Less frequent were motor failures, fused battery solenoids, and burned-out relays. Cold-weather operation caused trouble to both systems, although perhaps more to hydraulics, while high-altitude trouble was more serious to the electrical systems. No one of the troubles developed in either system has been insurmountable. All have been susceptible of solution, the net result being a marked improvement in the equipment of all manufacturers.

For purposes of comparison, three different airplanes, representing three different manufacturers, were singled out. Service troubles were totaled for both electrical and hydraulic systems for these airplanes over a given period of time. It will be evident that these reports occurred during epidemics of trouble. They do not, however, represent all of the failures that occurred, as a large percentage are never formally reported. The reports cover a considerable number of airplanes and represent a long period of time.

The figures, as noted, list the number of unsatisfactory reports received on airplane No. 1 for a given period, but do not necessarily mean the total number of failures, as several unsatisfactory reports include more than one failure. In this airplane, the electrical-system source was capable of a maximum of approximately 3.25 hp delivery, while the hydraulics source delivered a peak of approximately 5.5 hp at any one time.

Hydraulic Failures

1 Packing failures: 27 unsatisfactory reports, comprising 28 or more[2] occurrences; of these, three reported belly landings due to the failure.

2 Hose failures: 4 unsatisfactory reports; one reported belly landing due to the failure.

3 Broken lines or fittings: 4 unsatisfactory reports; three due to interference.

4 Pump failures: 4 unsatisfactory reports; comprising four or more[2] occurrences.

5 Cylinder failures: 11 unsatisfactory reports of which 9 were structural failures.

Electrical Failures

1 Burned or fused wires or groups of wires or plugs: 24 unsatisfactory reports, comprising 32 or more[2] occurrences; of these, 9 reported complete electrical-system failure, of which 6 reported forced landings due to runaway propellers.

2 Voltage regulators: 3 unsatisfactory reports; one reported 30 per cent of all voltage regulators tested were out of adjustment.

3 Inverter failures: 4 unsatisfactory reports, comprising 21 or more[2] occurrences; 18 were reported due to high altitude.

4 Generator failure: 8 unsatisfactory reports, comprising 22 failures; average life reported to be 15 hr.

5 Starter failure: 4 unsatisfactory reports.

6 Relay failure: 5 unsatisfactory reports comprising 7 occurrences.

7 Fused battery solenoids: 4 unsatisfactory reports, comprising 4 or more[2] occurrences.

There were other miscellaneous trouble reports in both systems, but since they were not repetitive, no inclusion was made in the foregoing list.

Both electric and hydraulic motors were in use on this airplane, and only one unsatisfactory report on an electric motor was received.

Another airplane of a different type, made by a different manufacturer, indicated a total of 108 unsatisfactory reports for hydraulics, and 110 unsatisfactory reports for electrical equipment for the same period of time. Only basic electrical-system failures were included here; no heating, lighting, communications, or instrument failures were listed. Airplane No. 3 of a different type by a different manufacturer shows 63 reports of trouble with hydraulic pumps against 38 reports of trouble with generators, voltage regulators, and relays. Total maximum horsepower available from the hydraulic pumps was approximately 45, as compared to approximately 13 delivered by the electrical generators.

If these troubles were summarized, they would probably prove to be nearly equal. If, however, a comparison is made on the basis of number of troubles experienced per horsepower transmitted, the favor rests rather heavily with hydraulics. It may be disputable as to whether or not the power comparison is a fair yardstick, but for want of a better one it will have to suffice.

In passing, a plea should be made for more immediate action on service complaints. They are all eventually acted upon, but the length of time that often elapses before any action is taken is appalling. This has undoubtedly done more toward prejudicing certain persons against one or the other system than any other avoidable cause.

Vulnerability. The subject of relative vulnerability of hydraulic systems and electrical systems is a delicate one indeed. In order to be clear on all fronts, we will define vulnerability as lack of ability to survive gunfire.

The first, and natural, and perhaps correct assumption is that electrical systems are less vulnerable. Foremost and insurmountable is the fact that there is no fluid to be lost in electrical systems, and any number of minor lines may be severed without damage to the rest of the system. Although fusing of wires or plugs from short-circuits is not infrequent, it is seldom that a serious fire results from this cause. In a hydraulic system, however, we are carrying an inflammable fluid around in pipes to

[2] One or more unsatisfactory reports noted "many failures," but did not specify how many.

most parts of the airplane. Thus, while the hydraulic system itself may be incapable of starting a fire through its own operation, the fluid it utilizes may be ignited from the incendiary projectile that breaks the line or unit containing the fluid. The same projectile that would break a hydraulic line, but fail to ignite the fluid, might conceivably break electrical connections in the immediate vicinity, creating a short that would ignite the fluid. While this may or may not be a far-fetched conclusion, the blame for the fire would rest with the hydraulic fluid, since the electrical wires were there of necessity.

In contrast to these arguments it can and has been theoretically shown that hydraulic systems can be made less vulnerable than comparable electromechanical systems by virtue of the smaller exposed vulnerable area. Where gravity cannot be utilized emergency measures can usually be applied more readily to a cylinder than to an electric motor-gearbox combination. Relative to emergency means of operation, the advantage enjoyed by a gravity extendable hydraulic landing gear over the electromechanical gear disappears with the use of the ball-bearing screw jack. A functional advantage may still rest with the cylinder for the gravity extension by use of the dashpot. The same thing may be accomplished on the electric drive by means of a centrifugal-governor-operated brake, but not so simply as the dashpot on the cylinder.

Enough more could be said on this subject of vulnerability to make a separate paper. It is certain, however, that those best qualified to present the facts are too busy fighting to keep the airplanes in the air to take time out to write about it. Unfortunately, the information that does filter back is often as conflicting as the arguments which arise over the basic subject under discussion.

Functional Suitability. By functional suitability, we mean a solution to a mechanical problem that gives us the exact operation we are seeking in the simplest, most direct manner possible. It is here that hydraulics enjoys one of its greatest advantages. As stated earlier, the problem of translating transmitted power into useful work is practically a separate and distinct problem from the transmission of power itself. It is in this translation of power into useful work that the hydraulic cylinder plays a principal role.

As has been said so many times before, the straight-line motion of the cylinder has no satisfactory electrical counterpart. There is more, however, to the advantage of the cylinder than just straight-line motion. The starting, stopping, and rate of travel of the cylinder are very simply controlled. The cylinder is its own basic travel-limit device for both extremes of its stroke. By means of the dashpot, the rate of travel of the cylinder during part of the stroke may be effectively governed. No special means have to be provided to start and stop the cylinder anywhere in its travel, as we are dealing with lightweight parts, traveling at low speed, immersed in oil. The electric motor-screw jack combination does not offer a favorable comparison with this operation. Limit switches must be provided to govern the length of stroke, and magnetic clutches or brakes must be employed to start and stop the unit with any reasonable accuracy. For use in areas subject to extreme vibration, the cylinder is ideal and natural. As a vibration damper, the oil column serves a very useful purpose. Use of a screw jack, gearing, and flexible shafting in a function in which vibration defies stress analysis does not seem logical. Electromechanical units have been developed through years of experience, however, that will withstand this vibration. Similarly, electrical brakes and purely electrical power turrets have been developed that have been operated successfully. These electrical uses, however, are functionally about as natural as the use of phosphorescent hydraulic oil for lighting.

The transmission of power from one point to another cannot be done so simply or economically with hydraulics as with electricity. In this regard, the electrohydraulic gun turret offers a common-sense solution to a difficult mechanical problem. It enjoys the simplicity and lack of vulnerability of the electrical transmission system, but utilizes the compactness and natural sensitivity of control and response of the hydraulic system. It might be said that this is an ideal example of intelligent use of both media. This should not necessarily be inferred as being a recommendation for the use of an electric-motor-driven pump for each and every piston in the airplane; rather, it is a plea for open-minded intelligent use of both media, employing them to their best advantage wherever possible.

It was stated earlier that the cylinder had no electrical counterpart; this is not strictly true. The solenoid is essentially similar to the cylinder in its action. For aircraft use it is limited by size and weight to light duty. This can be put to excellent use in the hydraulic system itself. The directional-control valves for landing gear, flaps, power-plant-cooling flaps, etc. present a problem in remote control all their own. The problem can be solved by two basic methods: (1) either the control valves can be located in the cockpit and directly controlled, or (2) they can be located at the unit to be remotely controlled. To locate them at the unit to be controlled obviously provides the cleanest hydraulic system from the standpoint of plumbing complications. It is in this remote control that the solenoid finds an ideal use. The solenoid-operated valve presents another common-sense solution to a problem involving the use of both media.

It has been said that to combine both systems is to double the vulnerability and be subject to the malfunctioning of both. The following statements would prove this untrue:

1 Conceding that present electrical systems are less vulnerable to gunfire than present hydraulic systems and that the installation of double wiring is feasible for such valves, we are not doubling our vulnerability.

2 A magnificent reduction in piping and cables can be made by the use of solenoid-operated valves. Reducing the quantity of piping involved reduces the vulnerability in direct proportion. This can only too readily be demonstrated on many airplanes where the multiplicity of piping strains the imagination as to how a projectile could possibly miss all of it.

3 Whether the valve be directly controlled in the cockpit, remotely controlled by cable, or remotely controlled by electricity, a suitable emergency means of operation must be provided if the operation involved is of major magnitude. The use of solenoid valves for these operations does not change this.

4 The design of solenoid-operated valves has progressed to the point where they can no longer be regarded as being in the experimental development stage. It is only extensive use that will provide the opportunity to prove them to the point where their use is as natural as direct control.

Returning to the subject of functional suitability, we find that the smoothness of operation that can be obtained with calibrated valves, as used on power turrets, on surface-control power boost, or similarly as the metering pin on oleo struts cannot be so simply duplicated electrically. Similarly, the ability of hydraulics to transmit true "feel" through its transmission lines is an advantage readily utilized by the hydraulic brake. It might also be worth mentioning that the ability of a hydraulic cylinder or motor to absorb an overload indefinitely without damage cannot be so naturally accomplished electrically.

Electricity, on the other hand, presents many functional advantages which cannot be equaled hydraulically. For very low power requirements, hydraulic power transmission is not economical, and we naturally turn to electricity. Units utilizing high-speed rotary motion, such as centrifugal pumps, are ob-

viously better done electrically, and the synchronous electric motor will find many unparalleled uses. The ability to store energy for ground operation is readily accomplished by the battery for the direct-current electrical system. Size and weight limitations of the battery, however, make the compressed-air bottle or hydraulic accumulator more satisfactory for storing large amounts of energy for emergency operations requiring heavy work, so that no advantage can be conceded to the battery. The alternating-current electrical system, unless augmented by direct-current service, is no better than a hydraulic system for ground operation and lacks the ability of the compressed-air bottle for emergency duty.

Other advantageous properties that must be conceded to electrical-power transmissions are its cleanliness and its relative independence of normal operating temperatures. Gearbox and motor lubricants, mechanical clearances, etc., are affected by temperature, but they do not present the design problem that an expanding and contracting fluid offers.

Summarizing, it may be safe to assume that, for most purposes, hydraulics is more suitable as a solution to mechanical problems involving relatively large forces or power, while electricity is more suited to light-duty work.

Low Weight. It is believed that no voice will be raised too strongly in objection to the statement that, power for power, present-day direct-current electrical systems are heavier than equivalent hydraulic systems of the 1000 to 1500-psi range. This fact has been one of the main talking points other than functional suitability for the use of hydraulics to supplement electricity.

Power for power, hydraulic pumps are lighter by far than electrical generators, be they direct-current generators or alternating-current alternators. Likewise, the hydraulic motor is lighter for an equivalent power rating than even the presently proposed high-speed alternating-current electric motors, and by far lighter than direct-current motors. For example, a piston motor has been successfully run continuously at 4000 rpm at 1500 psi, delivering approximately 8.6 hp. This motor weighs 4.3 lb, thus giving a ratio of 0.5 pound per hp. For intermittent duty, the motor has been run at higher speeds and higher pressures. An intermittent-duty 8-hp d-c motor running at about 5500 rpm is quoted at approximately 17 lb, while a 12,000-rpm intermittent-duty 8-hp a-c motor weighs approximately 14 lb. For continuous duty at this power, no actual figures could be obtained, but an approximation of 30 to 40 lb seems to be in order. The analogy breaks down, however, for small power requirements, and the electric motor becomes lighter than the hydraulic motor. In transmission, electric wires are lighter than equivalent hydraulic lines and oil, for small power on 24-v direct current, and for all power on the higher-voltage alternating-current systems.

The weight and size of the cylinder are dependent upon the total work performed, and the efficiency of the stroke, not upon the rate of operation. Thus for a given operating pressure, it is the peak load on the stroke-load curve which determines the cylinder diameter. It is desirable, of course, to keep this peak load as small as possible. On a landing-gear cylinder, where the cylinder size and time of retraction may well dictate the size of hydraulic pump and lines required, this becomes of extreme importance. Thus in Fig. 1 we have a typical landing-gear-cylinder load curve.

Assuming that this cylinder is used in a system with a peak design pressure of 1500 psi, the cylinder diameter required would be 6 $^7/_8$ in. (neglecting rod area). The cylinder volume would then be 2.57 gal on the retraction side. If it is desired to retract this gear in 10 sec, approximately 15.4 gpm must be fed to each gear cylinder. Assuming for simplicity that the nose gear would have the same load curve as the two main gears, the total flow required from the power pumps would then be 3 × 15.4 or 46.2 gpm. By attaching the cylinder to the drag-link fulcrum horn by means of differential linkage, the peak load was reduced to 36,000 lb as shown in Fig. 2. The cylinder diameter required for this condition was then 5 $^5/_8$ in.; the volume was reduced to 1.72 gal. The rate of flow to each cylinder now became 10.3 gpm, and the total flow required from the power pumps was 30.9 gpm. The added weight of linkages was more than compensated for by the reduction in cylinder weight alone, while the weight saved in line-size reduction, oil, power pumps, and the attachment structure for the cylinder was pure gain.

Utilizing the higher pressures (1500 to 3000 psi) and taking as full advantage as possible of design features as just illustrated,

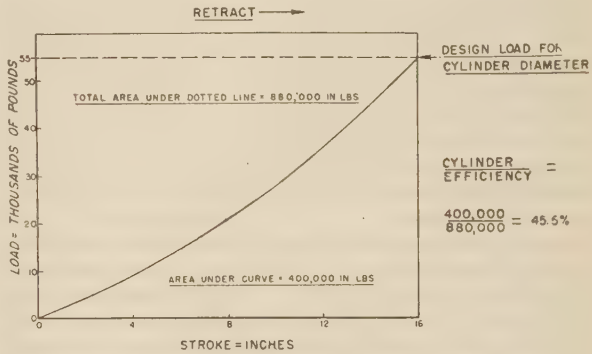

Fig. 1 Typical Landing-Gear-Cylinder Load Curve

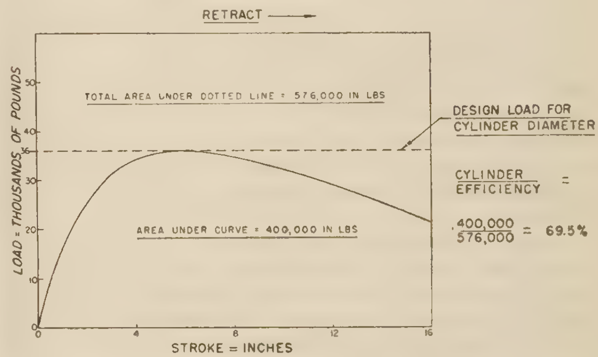

Fig. 2 Curve Showing Reduction of Peak Load in Landing-Gear Cylinder

the weight superiority of hydraulic systems over 24-v d-c systems can be maintained in most cases over a-c systems. This is true, however, only as long as we are dealing with relatively high power requirements. As allowable gear- and flap-retraction times are increased and the power requirement drops, a turning point is reached where the electrical system enjoys a weight advantage over the hydraulic system. Where this point lies must be determined for each different airplane, unless a cursory inspection of conditions makes it self-evident.

In this regard, the a-c alternator has a property that must not be overlooked, i.e., the ability to absorb approximately 50 per cent overload for 5 min without damage. Thus it is conceivable that, on a large cargo and/or passenger airplane, the alternator size, as dictated by lighting, heating, communications, equipment, air conditioning, and other ordinary duty, will be sufficient to handle landing gear and flap loads without any increase in size, merely by being overloaded for a short period of time. If there is no duty which requires hydraulic service, the

total weight of the complete hydraulic-power system must be added to the comparable weight analysis of cylinders versus motors and gearboxes, etc. This free power may well place the weight advantage in electrical operation many times. The comparative weight between high-pressure hydraulic systems and a-c electrical systems then resolves itself down to particular cases.

Simplicity. It would seem logical that hydraulics, since it is really the mechanics of fluids, would be far simpler than electricity. Practically, it may be said that this is reasonably true. The hydraulic cylinder by no means presents the design problem that the gearbox does. With the use of "O" ring packing and the establishment of standard clearances, cylinder design has become extremely simple. Similarly, the development and testing of the cylinder is much simpler than that of a high-reduction gearbox. Providing both cylinder and gearbox satisfactorily pass a life test, there should be no service required other than a periodic drain and flush for the gearbox lubricant and a periodic check of the cylinder packing.

As a balancing factor against the mechanical development time required by the gearbox is the standardization of electrical parts. In practically every case 100 per cent of the purely electrical equipment is purchased from electrical-equipment manufacturers, and the development cost of such equipment is amortized over many different companies. The aircraft electrical engineer's job, then, is one of installation, rather than of design and development.

Aircraft hydraulics, however, is relatively new. We have not settled as yet the majority of problems that will have to be resolved before we can begin to use standard items "out of stock." It may be a long time before we are ready to agree on a type of system, an operating pressure, and even upon such basic parts as a suitable leakproof coupling. Consequently, the aircraft hydraulic engineer's problem is one of design and development as well as installation. It is realized, of course, that electrical systems will undergo a period of development with the advent of a-c systems, but such development work will be a function of the electrical-equipment manufacturers rather than the ultimate users, even though they be the guinea pigs for its testing.

In the field of hydraulics, there is no parallel gigantic research organization to which we can present our problems and know they will be worked out from beginning to end. There is no one organization set up to engineer, develop, and manufacture a complete hydraulic system from reservoir, to pump, to valve, to cylinder. Each aircraft company is dependent upon its own engineers plus those of a multitude of vendors. Undoubtedly, there is room in the hydraulics field for such a complete organization. Perhaps we shall someday see one.

Simplicity of maintenance, like reliability, is very close to being a stand off between the two systems. A mechanic with only a smattering of electrical knowledge will favor the hydraulic system because it is basically easier to visualize what goes on in a pump or cylinder or valve, than in a generator, electric motor, or relay. Similarly, the trouble is ordinarily easier to trace in a hydraulic system, although on occasions this can be a drawback.

In military theaters, the necessity of stocking hydraulic oil is undoubtedly a nuisance. If, however, this hydraulic oil is the only satisfactory low-temperature lubricant for gearboxes, etc., its use might be justified. It would seem easier, also, to replace a shot-up wire or bundle of wires than a similar group of prefabricated tubing. If we consider, though, that the tubes and hydraulic cylinders are replacing gearboxes, torque tubes, flexible shafts, and cables, we are not so prone to concede the advantage. As it is necessary to have trained mechanics and trained electricians wherever the aircraft goes, any claim to advantage in one or the other system would seem purely academic.

Low Cost. Low cost is closely allied with simplicity, and the same remarks apply to it as to simplicity. The cylinder is basically cheaper than the motor-gearbox combination, but the standardization of electrical parts is much superior to hydraulics standardization, thus resulting in somewhat of a balance. If the electrical engineer is forced to use special generators, special voltage regulators, special motors, etc., his cost would probably be greater than a similar hydraulic arrangement.

It is too often believed that cost should not be considered in time of war. Cost, we must remember, however, is a measure of time and material, and time and material is a measure of quantity of production. Thus to achieve the correct balance, we must weigh cost carefully with function to make sure that the time and material expended justify the purpose. Too often recently we have seen numerical superiority prove more important than quality. To strike a happy medium, then, is the purpose of every conscientious engineer and manufacturer.

Compactness. A strong case can be made for the hydraulic cylinder when space is at a premium. In a small fast pursuit plane, where every nook and cranny contains some piece of equipment, the ability of a cylinder to change its "gear ratio" by changing its diameter and stroke, and yet do the same amount of work, is extremely useful. It is usually in such planes that rapid rates of gear retraction are demanded, thus making the job a natural for hydraulics. Small motor - gear screw combinations can be made, but they cannot compete with the cylinder on a power basis simply because the cylinder is independent of rate of operation while the motor is not. Plumbing, however, outdistances wiring by a great deal in space consumed. Thus from a compactness point of view, the choice depends upon where the compactness is wanted if it is going to be a contributing factor to the decision.

Aside from the solenoid, electrical power is limited to high-speed rotary motion. To convert this into low-speed rotary motion or linear motion requires gearing of some sort or other. Such gearing, however, as developed by electrical-equipment manufacturers is not necessarily extremely heavy or inefficient. The use of precision-built planetary gearing in steps provides very high reductions with amazing efficiency for relatively low weight. A new design of ball-bearing screw jack has opened up new fields for electric motors by its very high efficiency. Thus with spur gearing and the ball-bearing screw jack, one may conceivably go through a very high gear reduction, translate into linear motion, and emerge with a relatively high mechanical efficiency. Such gearboxes may also contain torque limiters, magnetic brakes, speed limiters of the centrifugal type, and all the necessary bearings and still be compact and relatively light in weight.

Evaluation of the Two Media

As previously stated, the evaluation of the comparison between the two media is probably the most opinionated part of the whole controversy. To those who say that everything must be sacrificed to vulnerability during the present emergency, the obvious answer seems to rest with all-electrical service. The weight, simplicity, and functional advantage usually resting with hydraulics, however, would seem to justify its use, even though necessary to employ some armor and self-sealing tanks. That this view is held by certain German designers is evident from some of their more recent designs.

Any tendency to adopt all-electrical service could be construed as being partially a fault of the hydraulics engineers in that we have not worked together perhaps as closely as we might have. That this was originally due to competition is undoubtedly true. Nevertheless, our aircraft electrical men were compelled to work together by being so dependent upon outside

sources, and in so doing have gained very definite advantages.

Through simplification of existing systems and careful design of future systems, hydraulics can be made, it is believed, no more vulnerable than a comparable electrical system. For civil aircraft where vulnerability to gunfire is of no concern, hydraulics should again be the favorite it has been in the past. Surface-control power boost will play no small part in more firmly establishing hydraulics as a power-transmission medium. With the advent of larger and faster aircraft, the power requirements will grow accordingly and with this we shall probably see general use of a-c electrical systems and 1500- to 3000-psi hydraulic systems.

In the field of naval vessels, the same trend to all-electric operation and control was made. The pendulum is now swinging back again, and the newest electrohydraulically controlled ships have proved themselves so well in actual combat that a definite movement to extensive electrohydraulic control is foreseen.

A decision to restrict all new military aircraft to all-electric control on the basis of past performance would therefore seem unwarranted and short sighted at this time. There are very definite advantages to be gained through the use of hydraulics in military aircraft as well as in civil aircraft. It is recommended, then, that the best uses of both systems continue to be utilized, even though it may mean mergers of the services in some cases to achieve ideal conditions.

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Discussion

H. F. Rempt.[3] The author's presentation is so fair that it is quite difficult for the writer to pick out exactly where the aircraft electrical system is really stepped on. However, since the problem is a controversial one, the following comments are given on parts of the paper, in the order that they were presented:

The two main parts of the problem were expressed as "transmission of power" and "conversion of power." It is believed that a third part should be included, namely, "remote control of power." In the present airplanes, it is hard to find one application of power that is not remotely controlled.

The author states, "of the three basic services, electricity alone is an absolute necessity." If this fact is conceded, then it is necessary for every hydraulic application not only to compare favorably with the same electrical application, but also to justify its share of the installation of the entire hydraulic system.

Under "failures," the number of unsatisfactory reports for hydraulic and electrical operation was approximately the same. However, the statement was made, "the favor rests rather heavily with hydraulics if the comparison is made on the basis of number of troubles experienced per horsepower transmitted." Rather than use horsepower as a comparison, it is believed the number of systems or units should be used.

The assumption that electrical systems are less vulnerable than hydraulic systems is true, particularly if the comparison is made by similar systems.

The writer differs with the statement under "Suitability," that electrical power turrets are not functionally natural. Since the guns elevate in an arc and the turrets revolve, it is believed an electric turret is a natural application for rotating power.

The mention of some type of operation "providing the cleanest hydraulic system" does not agree with the writer's experience when standing in wheel wells. The writer does not believe there can be a clean hydraulic system.

The comparison of compressed-air bottles and accumulators to the battery is in error, since the battery is in the airplane anyway and the bottles and accumulators then become added weight.

The writer cannot subscribe to the statement, "the aircraft electrical engineer's job is one of installation rather than design and development." Most of the new developments in aircraft electricity come about by a design and development program undertaken by an electrical manufacturer in conjunction with preliminary design information from the aircraft electrical engineer. The main difference between electrical and hydraulic equipment is that the electrical equipment is not usually made in an aircraft plant because of the special machinery and technique that is required.

For simplicity, the engineer's ideal of a "little black box that snaps in place" is practically answered by present-day electrical equipment. The airplane will be simple to service if the parts are simple to install. By means of bench-testing of complete electrical systems, the installation problem is also simplified. There is no comparable method of bench-testing complete hydraulic systems.

The writer considers the use of electrohydraulic control a step in the right direction. It will be only a matter of 5 or 6 years before the complexity of control requirements will make the all-electric airplane a reality.

[3] Electrical Engineer, Lockheed Aircraft Corp., Burbank, Calif.

The Evolution of the Hydraulic Pump as Applied to Aircraft

By DALE HERMAN,[1] BEVERLY HILLS, CALIF.

Having passed through a period of manual control of auxiliary power functions on earlier aircraft, and later the hand-pump and low-pressure vane- and gear-pump stages, continued demands for higher pressure eventually led to the present-day types of piston pumps which supply necessary power to hydraulic systems of control. The author outlines the basic requirements for well-designed oil-hydraulic equipment including pumps, and then in detail describes the gear pump and the axial-piston pump in both the constant- and variable-delivery designs. Considerations in selecting pumping equipment and the design of the hydraulic circuit are stated.

EXCEPT for earlier simple types of aircraft and small craft of current design, power to perform many auxiliary operations formerly handled by direct manual control has long been a necessity. Power was first introduced in the form of hand pumps, actuated by the pilot, to operate retractable landing gear and brakes.

Being capable of high pressures at reduced volumes, these pumps are still widely used for emergency service. With the advent of larger and faster aircraft, power-operated pumps became necessary, and the earlier hydraulic systems were equipped with low-pressure vane and gear pumps. Continued progress made it expedient to provide more power for the hydraulic system, necessitating higher pressures, and consequent improved designs of gear pumps, and later the introduction of piston pumps to cope with the loads. Today the engine- or electric-motor-driven pump is the heart of the aircraft hydraulic system, and the trend is definitely toward higher pressures on military, transport, and cargo planes.

Basic Pump Requirements

Well-designed oil-hydraulic equipment, including pumps, as used for aircraft, should be neither heavy nor bulky and should transmit power efficiently. Consequently, pump bearing loads lean toward the permissible maximum in order to reduce bulk, and operating clearances are held to a minimum to reduce leakage losses, commonly termed "slip" in pumps, and higher speeds are used to reduce weight per horsepower.

The weight feature of the pump and of other hydraulic equipment is carefully evaluated by every aircraft engineer when undertaking a new design. The efficiencies of the accessory items, such as the hydraulic pumps, which are being added to the engine loads, are not always given as close attention. The hydraulic horsepower generated by a pump is expressed by the following commonly accepted formula:

$$\text{Hydraulic horsepower} = \text{displacement (gpm)} \times \text{pressure (psi)} \times 0.00058$$

The engineer should study the characteristics of the pump selected, from the standpoint of power input under system operating conditions, as well as from the hydraulic output, as represented by this formula, to determine efficiency, and to evaluate the useful work transmitted.

The component parts of hydraulic pumps, designed for high-pressure service, must be manufactured with precision, and ample provision must be made to lubricate properly all working parts, which are in turn suitably housed to protect them from dust, moisture, etc. present in the atmosphere. However, the efficiency and life of the pump, like that of any other piece of precision machinery, incorporating bearings and close running fits, will depend largely upon the care with which it is installed and the attention it receives while in service.

Oil under pressure tends to leak through small running clearances, however minute. All such leakage represents a loss in the applied energy without accomplishing useful work. Such loss in energy is transformed into heat, raising the oil temperature, and if considerable leakage is present, auxiliary cooling may become necessary. Consequently, clearances between moving parts, subject to pressure, must be held to a minimum to insure high operating efficiency. These must be more closely controlled in high-pressure pumps.

If foreign matter of an abrasive nature is allowed within the hydraulic system due to inadvertent carelessness while installing or maintaining the equipment, it will impair the life of all operating parts in the pump, valves, and motors comprising the system. It is noted, therefore, with considerable satisfaction, that the aircraft manufacturers who are pioneering higher operating pressures in hydraulic systems have not overlooked the matter of improved oil filters. This subject is covered in detail in a paper by W. W. Thayer.[2]

In his connection with a manufacturer of vane, gear, and piston pumps for industrial and marine service, the author has acquired considerable experience with the gear and piston designs, which are employed exclusively on aircraft systems, and thus is qualified to discuss the merits of each design, and the class of service for which each is best adapted.

Gear Pump

A large percentage of aircraft systems to date has employed the gear-type hydraulic pump, as pressures below 1000 psi have predominated. The gear pump which is relatively simple in construction, consisting of two accurately machined and fitted gears in mesh, occupied a prominent place in the field until pressure and volume characteristics reached a point at which the requirements exceeded the efficiency limitations of this construction. Gears and the bores in which they operate can admittedly be machined to the same degree of accuracy as pistons. However, due to the larger critical dimensions of the gears, as compared with pistons, to obtain required displacements, these parts have greater linear expansion with temperature change, and must therefore be fitted initially to insure adequate running clearances at the two temperature extremes to which they are subjected in service. Fits on both the width of the gears and the diameter have a bearing on the internal slip or leakage of

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Contributed by the Aviation and Hydraulic Divisions and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

[2] "The Modern Hydraulic Reservoir," by W. W. Thayer, appearing on p. 589, of this issue of the Transactions.

the pump and therefore on its volumetric and over-all efficiencies.

In developing a line of hydraulic equipment for machinery manufacturers, and subsequently for aircraft, first consideration was given to the design of a suitable pump with constant-displacement characteristics. The gear pump, which was most commonly used, was first considered but discarded because this pump is hydraulically unbalanced over the projected area of each gear, as illustrated in Fig. 1. While running, the extent of this unbalance undoubtedly will exceed that shown on the diagram,

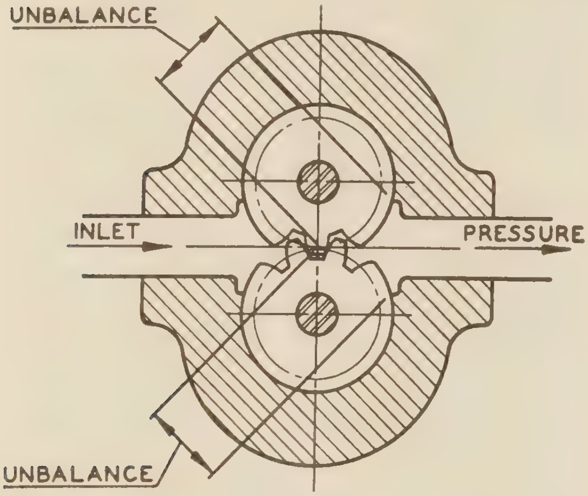


Fig. 1 Unbalanced Gear-Pump Design

because the pressure will tend to distribute itself around the periphery of each gear, being maximum at the discharge port and decreasing toward the suction port. Since the bearing diameters must necessarily be smaller than the periphery of the gears, it is necessary to increase the length of the bearings in proportion to the width of the face of the gears, in a conventional design, to keep the projected areas and the resulting bearing loads within safe operating limits. This also tends to limit the maximum pressure and displacement characteristics of such a pump.

It therefore became apparent that the necessary running clearances between the outside diameters of the gears and the pump body, as well as that of the bearings to their bores, had to be accurately maintained or the radial loading, combined with the differential expansion of the respective parts due to temperature changes, would cause the periphery of the gears to come into metal-to-metal contact with the housing. This would cause scoring and loss of efficiency. Bearing wear, resulting from this unbalanced hydraulic loading, which is accelerated by the presence of foreign materials in the oil, will therefore result in rapid deterioration of the pump.

The design was then modified to balance the gears hydraulically by arranging porting that would apply pressure on the periphery of each gear diametrically opposite to the discharge port and of like area. Gear motors, balanced in this manner, have been used on aircraft, as well as on marine and industrial applications, but their efficiency due to slip does not compare with the piston motor; and the same condition holds true when they are used as pumps.

On most gear pumps the running clearance over the side faces of the gears is fixed to conform with thermal-expansion and lubrication requirements. One manufacturer has introduced an innovation in the normal recognized construction of gear pumps, to control the running clearances over the width of the gear. In certain capacities, this design has been recommended and approved for 1500-psi service.

Gear pumps are made commercially having constant-delivery characteristics, the delivery per revolution being fixed except for slip which increases with pressure. Hence it is necessary to use such pumps with an accumulator system where variable volumes are required for operating different devices, for best efficiency.

As an alternative, on extremely low-pressure systems, where pumps of small displacement are used, such as on gyropilot systems, the delivery of the pump beyond system requirements is by-passed through a relief valve at a constant pressure, thus converting the unused energy into heat. To avoid the necessity for such a second low-pressure pumping system, the use of a pressure-reducing valve is recommended. This takes oil from the high-pressure system on the plane and reduces it to the low pressure required for the automatic pilot or similar auxiliary controls.

Axial-Piston Pump

The axial-piston pump is essentially a multipiston engine, the stroke of the pistons being determined by the angle of the rotating group, including cylinder block, pistons etc., to the drive shaft. These units were first developed in larger sizes for industrial-machinery applications, and a wealth of experience was available as a result, when the design of lightweight aircraft models was undertaken. Pumps of this same design have been in use in marine, ordnance, and machinery service for power transmission for 8 years. Sizes are available from the smallest constant-delivery aircraft unit that weighs $3^1/_4$ lb and develops 4 hp at 4200 rpm and 1000 psi, to marine gears requiring variable-displacement pumps, operating at 850 rpm and developing up to 300 hp.

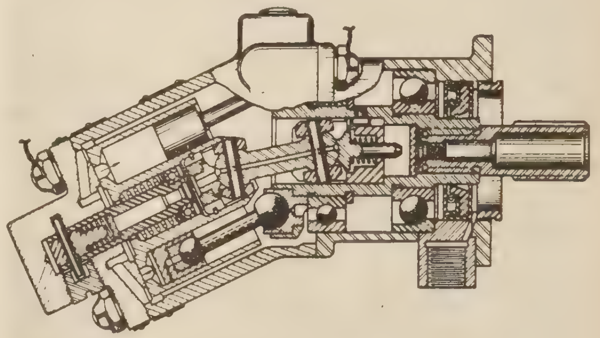


Fig. 2 Constant-Delivery Type of Axial-Piston Pump

The design is well adapted to aircraft service because the bearings can be small in diameter, permitting high-speed operation; and rotating parts are compactly spaced in a manner to minimize weight. The units operate equally well as motors or as pumps, and at the same time offer the advantages of interchangeability of the rotating group of parts as a service consideration. Of equal importance, they may be constructed for variable-delivery output.

These pumps are now available in aircraft types with constant- or variable-delivery characteristics. Basically they have seven or nine pistons that are joined to the drive shaft by connecting rods, the latter having ball-and-socket bearings on both ends. No piston rings are used, but the pistons are fitted to the bores of a bronze cylinder block to a clearance of a few ten-thousandths of an inch, depending upon their diameter. The cylinder block is in turn mounted on a stub shaft and is driven by a universal shaft, having a universal joint at either end to insure that the

speed of the cylinder block corresponds to that of the drive shaft. This universal drive merely acts as a timing member, and overcomes bearing friction within the cylinder block. It does not transmit power, as all working loads are transmitted from the drive shaft directly through the rods to the pistons.

In the constant-delivery type of pump shown in Fig. 2, the angular position of the cylinder block is fixed as determined by the housing. These angles vary from 10 to 25 deg, depending upon the delivery required, as this angle determines the stroke of the pistons and thereby the displacement.

The drive shaft is, of course, mounted on ball bearings, with the angular-contact type taking the thrust produced by the pressure on the pistons, and the radial bearing carrying the radial load imposed by the angular position of the pistons. On pumps designed for pressures over 1000 psi, two radial and one angular-contact thrust bearings are incorporated.

The valve porting in the cylinder block and valve plates of these pumps is clearly illustrated in Fig. 3 and is designed to balance hydraulically most of the hydraulic thrust equivalent to the effective piston area subjected to pressure. Therefore, the Kingsbury type bearing, on the periphery of the hardened and lapped steel valve plate, carries relatively low loads.

These constant-delivery pumps are available in various capacities up to a displacement of 0.507 cu in. per revolution, and for three pressure ranges, as follows:

1000 psi continuous duty	1250 psi intermittent duty
1500 psi continuous duty	2000 psi intermittent duty
2000 psi continuous duty	3000 psi intermittent duty

Although the principal working parts of these pumps are interchangeable, the shafts, bearings, housings, and valve plates must necessarily be proportioned for the pressure service. In other words, it was not necessary when designing for higher pressures to decrease clearances or to add any parts or stages to the lower-pressure design. The addition of stages would naturally reduce the over-all efficiency by the introduction of additional parts, creating friction and other internal-leakage points.

The largest constant-delivery piston pump available, rated for 1500-psi service, which delivers 6.4 gpm at 3000 rpm cruising and at rated pressure, weighs 5 lb. The weight per gpm at rated pressure is 0.8 lb. This is a more favorable weight-performance ratio than for any other aircraft pump design of comparable capacity known to the author. The input is 6 hp under these conditions.

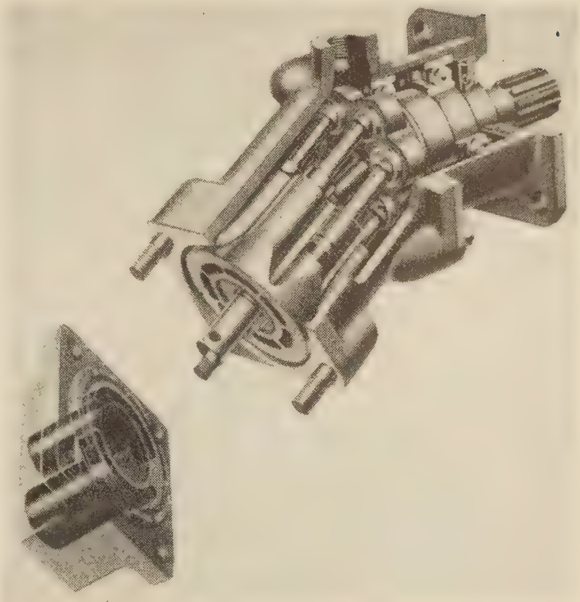


Fig. 3 Valve Porting in Cylinder Block and Valve Plates of Constant-Delivery Axial-Piston Pump

Another design of this pump, rated for 2000 psi, is being used by one manufacturer intermittently on an accumulator circuit operating at 3000 psi. At the latter pressure the delivery is 6.3 gpm at 3000 rpm cruising, and the pump weighs 6 lb. The weight per gpm under these operating conditions is 0.95 lb.

The reduced running clearances on the piston-type pump, as

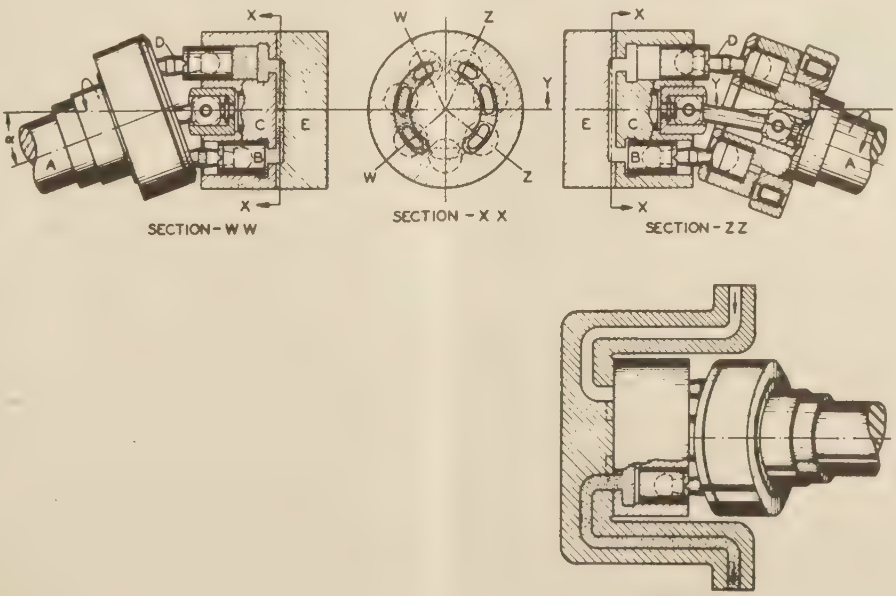


Fig. 4 Operation of Variable-Delivery Type of Axial-Piston Pump

compared with the gear pump, materially improves its self-priming characteristics, particularly when starting against a pressure head, and when there is air in the system. This condition might be encountered in an accumulator circuit.

The variable-delivery version of the axial-piston pump is essentially like the fixed-stroke type, except that the angular position of the cylinder block can be changed relative to the shaft, while in operation. The degree of angle determines the amount that the pistons may reciprocate in their bores, creating suction during one half of the revolution and discharging during the other half, thus varying the delivery. By moving the cylinder block across the center, the direction of flow through the pump may be reversed. This latter feature is used to advantage on certain rotary drives such as are employed on gun turrets, where both directions of rotation are required. The operation of this pump is illustrated in Fig. 4.

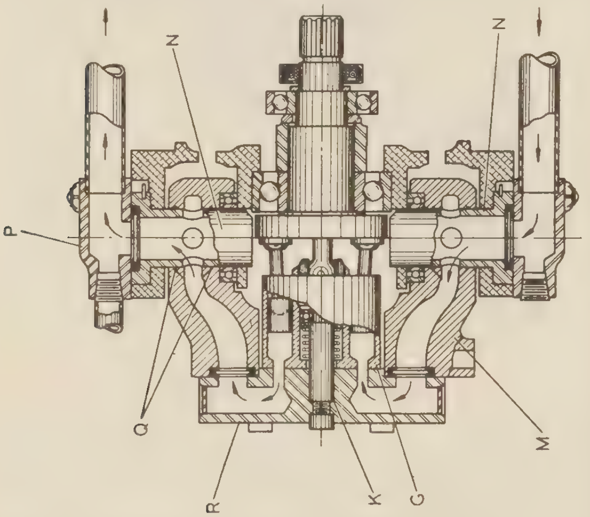


Fig. 5 Construction for Swinging Cylinder Block to Obtain Varying Volume

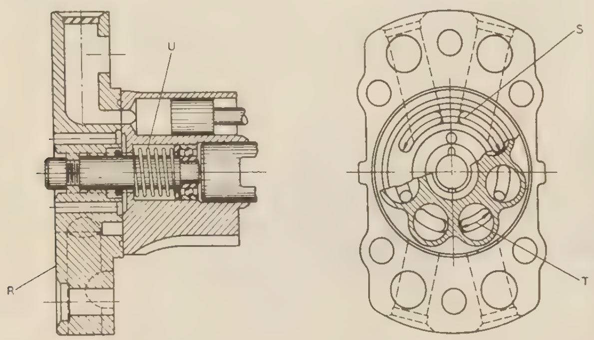


Fig. 6 Details of Porting in Cylinder Block and Valve Plates of Variable-Delivery Axial-Piston Pump

The construction used for swinging the cylinder block to obtain varying volume is shown in Fig. 5. A yoke containing the stub shaft on which the cylinder block is mounted is supported on two stationary pintles, and ball bearings are inserted in the arms of the yoke to reduce friction to a minimum. The porting of the oil to and from the cylinder block through the yoke and pintles is indicated by arrows. Flanges mounted on the pump housing, which is not shown on this schematic drawing, sealed by neoprene Vickerseals, retain the pintles and contain the connections. Circular seals are inserted into the yoke arms, and pressure is applied to their outside diameters to insure a minimum of leakage between the swinging yoke and the pintle.

Fig. 6 shows further details of the porting in the cylinder block and valve plate.

Several methods of controlling the flow from these variable-delivery pumps are used in production. The first consists of a mechanical linkage, by which the operator can vary the delivery of the pump by moving a lever or cam. A hydraulic servo or power booster is sometimes introduced between the linkage and the pump yoke, to minimize the manual effort. The latter control is also used in conjunction with syncromotors for remote electrical control and is employed on armament equipment.

The method commonly employed on engine-driven pumps is the self-actuated pressure compensator. This in its simplest form is a spring-loaded piston connected to the yoke, which normally holds it in the maximum displacement position, whereby, with pressure connected from the discharge of the pump acting on the piston, it overcomes the spring load and strokes the pump to the neutral position. Such a control causes the delivery to vary approximately inversely with the pressure. The conventional control of this type requires a heavy spring for high pressure and smooth operation and becomes quite bulky. Besides, on most installations the change from full pump volume to zero stroke must be accomplished with a minimum pressure differential, say, from 1500 to 1700 psi. This makes a practically constant pressure available to the system regardless of pump delivery. To accomplish this a compact pilot pressure-controlled device, providing convenient pressure adjustment has been designed, which insures accurate smooth regulation within the desired range. The differential rate generally employed is from 100 to 200 psi. This control is designed so that two or more pumps can be operated in parallel on the system, and each will supply its portion of the variable demand. With this type of control, and all valves in the system closed, resulting in no demand, the pump will maintain a constant preadjusted pressure, pumping only sufficient volume to compensate for slip within the pump itself, and in valves which are exposed to pressure.

Pumps of this general type for 1500-psi 15-hp service were first furnished to an aircraft manufacturer in 1939, and are now available for applications requiring maximum pressures from 800 to 2000 psi. Pumps furnished for 1700 psi have a maximum displacement of 1.518 cu in. per revolution, providing 25 gpm at 3750 rpm, during take-off, and 19 gpm at 3000 rpm, when cruising. The maximum input approximates 28 hp, and this unit weighs 25 lb.

Selection of Pumping Equipment

When selecting a hydraulic pump for a new design, the engineer must first consider the total hydraulic horsepower to be transmitted in terms of the formula previously mentioned. The two factors entering into this equation are the maximum pressure for which the system is to be designed, and the maximum displacement required for the limiting operation, which on many installations is that required to operate the landing gear. The pressure selected is generally determined by the size of the ship, taking into consideration weight and availability of the pump and all other accessories used in the system, such as unloading valves, relief valves, accumulators, hydraulic fittings, tubing, etc. The average pressure in use today would fall somewhere between 1000 and 1500 psi in production aircraft, but the tendency on new designs is to consider higher pressures. One manufacturer has designed a satisfactory system, functioning to a maximum pressure of 3000 psi. The power-driven pump must operate intermittently at this pressure, as a conventional unloading-valve and accumulator system is employed, requiring

that the pump obtain this maximum pressure only while charging the accumulator.

Rather than attempt to take sides in a controversial matter, the author prefers to point out the features which favor the use of higher-pressure systems and those substantiating a more conservative viewpoint, which would approach the matter of higher pressures more gradually. The arguments advanced for higher pressures are that they will permit the use of smaller sizes of tubing and fittings, as well as valves, cylinders, pumps, and hydraulic motors, resulting in considerable weight saving. Such savings will obviously add to pay-load capacity and also allow steel fittings at many points, thereby increasing ruggedness and decreasing bulk.

On the other hand, experience with systems of this type is at present somewhat limited and the necessary auxiliary equipment required for a complete system is not available to the same extent as for lower pressures. Closer manufacturing limits will be necessary in order to obtain the fits required in certain of these accessories, thereby in some cases increasing manufacturing costs. Control valves of the type that meter or throttle the oil flow for control purposes will present more of a problem because of the smaller sizes of the orifices necessary to obtain nicety of operation; and lint and other foreign substances in the system will become more critical in valves of this type. More consideration will have to be given to distortion of valves and other portions of the system resulting from pressure changes. The combination of vibration and high pressures will present difficulties in maintaining seals at fittings and flange joints of the respective units incorporated in the system.

The foregoing represents an unbiased viewpoint since the author's company manufactures certain equipment for systems falling within this range of pressure. It would, however, seem that any transition should be along conservative lines.

In the case of hydraulic circuits on machine tools on which we have had broad experience, the maximum pressure used approximates 1000 psi, as on this class of equipment nicety of control on fine feeds and reversal together with long life are the paramount considerations.

Another design problem which must be considered is whether or not a constant-volume pump, which is generally used in conjunction with an accumulator system, or a variable-volume pump should be used. In making this choice, the engineer should take into consideration the equipment which is hydraulically actuated by the circuit.

The majority of military and transport ships now in production employ the constant-delivery pump, with an accumulator suitable for supplying intermittent high demands and also providing a source of pressure for the brakes when the plane is on the ground and the engines are not running. Our observations are that such a system is most desirable on installations requiring intermittent hydraulic demands in flight. Most of the unloading-valve systems at this time function only during take-off and landing, and the remainder of the time the accumulator remains charged and the unloading valve cut out. During this time the pump will circulate freely to the reservoir during flight, unless there is an occasional demand, or some leakage from the accumulator portion of the system, in which case it may require intermittent recharging from the pump. The equipment operated from a system of this type generally includes: landing gear, flaps, oil coolers, wing floats, cowl flaps, power brakes, bomb doors, and wing-folding operations.

The use of a variable-volume pump would on such installations only eliminate the unloading valve and possibly the accumulator from the main system. Therefore, it would not offer any great weight saving or other economy.

The constant-delivery pump on such a system is subject only

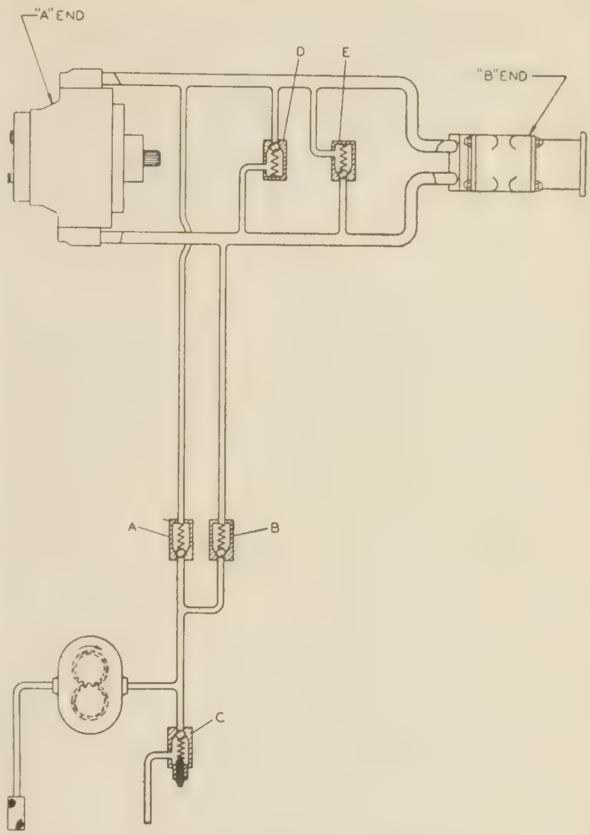


Fig. 7 Variable Pump and Fixed-Displacement Piston Motor Combined in Locked Circuit for Control of Armament Equipment

to intermittent service at system pressure, which is a desirable feature, as it tends toward long pump life.

The repeated stressing and relieving from pressure in the lines between the pump and unloading valve introduces some line shock and fatigue in this portion of the circuit. This becomes more pronounced at higher pressures. A properly designed unloading valve, however, minimizes this objectionable feature.

The variable-volume pump is gaining more acceptance on the larger-plane designs, where larger volumes of oil are required, and there is a continuous varying demand for hydraulic power during flight. Another feature of the variable-volume pump is that it will supply a system requiring a constant maximum volume in flight, irrespective of engine speed. The pump will operate at relatively short stroke during take-off, but at maximum stroke while the engine is idling, so that its delivery will remain constant throughout the normal operating speed range of the engine. Some of the controls which favor the use of the variable pump are (1) power boosters for control surfaces; (2) cabin supercharger drives, using hydraulic motors; (3) fan drives, employing hydraulic motors; (4) turret drives; (5) constant-speed accessory drives.

It is obvious on hydraulic circuits with the operations listed, that the steady usage would cause the unloading valve on a constant-delivery system to operate frequently in flight, and the pressure surges might introduce complications within the power-booster system, for example. The maintenance of steady pressure with a variable pump that would adjust itself to demands is conceivably better for such an installation.

The sensitivity of control, combined with reversible flow without a dead spot, made possible with the variable pump and fixed-displacement piston motor offers decided advantages for the operation of armament equipment. An infinite range of speed control, relatively unaffected by varying loads, due to the position of the guns in the slip stream, from the slow rates necessary for tracking the target to high slewing speeds, is possible with such a system. A typical locked circuit of this type is shown in Fig. 7. Control of this type is used on turrets in "Flying Fortresses," on various antiaircraft mounts, tank turrets, and in the largest gun turrets on ships. The low moment of inertia characteristics of the hydraulic motor and its corresponding high torque for a minimum of weight combine to make this type drive ideal for this service.

As a summary, it would be apparent that each design or type of construction has merit in one or more applications, and full consideration must be given to the circuit before selection is made.

The Modern Hydraulic Reservoir: How It Provides Micron-Range Filtration and Pump Supercharging

By W. W. Thayer,[1] Los Angeles, Calif.

At the present time, the term "high pressure," with reference to aircraft hydraulic systems, means 3000 psi. Successful design of such systems, now in quantity production by the author's company, depends upon careful consideration of details, rather than upon the adoption of any features that might be classed as unorthodox. For instance, detailed attention has been given to an improvement in service life of certain hydraulic units whose operation depends upon close-fitted sliding parts. This consideration, resolving into a study of methods for combating abrasion and cavitation, has led to the development of a filter for the removal of particles as small as 10 microns, and a simple arrangement for reducing pump cavitation by building up a supercharging pressure in the hydraulic-system reservoir.

When contemplating the design of large airplanes, almost without exception, the aircraft designer is thinking in terms of high pressures for the hydraulic system that will operate landing gears, bomb doors, wing flaps, and other services. It has become traditional in the aircraft industry to refrain from defining the word "large," as applied to airplanes. However, "high" as applied to hydraulic pressures, means, at least for the moment, 3000 psi. This is twice as high as the next highest pressure in common use.

The author's company has been the first to go into quantity production of airplanes with 3000-psi hydraulic systems. Obviously, the higher pressures yield major savings in weight and space. Experience has been that design practices for the higher pressures are consistent with the same fundamental principles that have been well established at low pressure. It has not been necessary to resort to any engineering trickery to make high-pressure systems work, but painstaking attention to design detail has been essential.

Improving Service Life of Hydraulic Units

As an example of design detail, this paper will describe the work that has been done toward improving the service life of hydraulic units, which, because of some specialized function, depend for leakage control upon close-fitting sliding contact between metallic parts, rather than the well-known V-ring or O-ring packings. Common examples of such units are pressure regulators, surface control booster valves, wing-flap synchronizers, constant-flow valves, and piston pumps.

Such hydraulic units are subject to wear because abrasive particles find their way into the hydraulic fluid, despite all practical precautions. Contamination may result from the natural wear of metal parts, from the flaking-off of protective plate, from dust sucked in as the reservoir breathes, from machining chips or free abrasives remaining in parts after washing, from particles broken loose by screwing in fittings, flaring lines, and tightening connections, from attaching the system to portable test stands that may have accumulated dirt, from dirt that enters past piston-rod packings or when filling the reservoir with oil, from blowing out tubing with an air hose, etc.

In addition to the effects of abrasive wear, deterioration of pumps may be accelerated by prolonged operation under conditions of cavitation. Sizable pits frequently occur where least desired, even in steel parts that have been hardened and ground. Furthermore, torque pulsations in a pump, running under cavitating conditions, may lead to the serious consequence of drive-shaft fatigue failure.

A typical aircraft hydraulic-pump installation is illustrated in Fig. 1. (A fixed-displacement pump is shown, rather than a variable-displacement type, due to the rather limited application of the latter in aircraft up to the present time.) The pump runs whenever the engines are running. As long as there is sufficient pressure in the "accumulator" to hold the "pressure regulator" in the by-pass position, the discharge from the pump is returned directly to the reservoir, at essentially zero pressure. Whenever the accumulator pressure falls, as it does whenever a hydraulic-operating cylinder begins to move, the pressure regulator acts to divert the pump discharge into the power system.

In the unloaded condition, the torque pulsations in a cavitating pump are probably less severe than if it were loaded, but the forces tending to cause pitting are undiminished.

Various changes were necessary to free the ordinary hydraulic system from the effects of abrasion and cavitation. It was the development of these protective measures which brought the commonplace hydraulic reservoir out of obscurity and into the center of interest.

There are few units in the hydraulic system which give the service mechanic, the design engineer, or the test pilot the same feeling of self-confidence as that engendered by the hydraulic-system reservoir. Everyone, with the possible exception of the weight engineer, casts a friendly eye on its bland, rotund exterior. They know that it is nothing but a hollow sheet-metal shell, conceived according to easily applied formulas set forth in Army-Navy specifications; so much foam space, so much basic volume, so much hand-pump reserve, etc. The reservoir has consistently been the one item in the hydraulic system which could be identified, and thoroughly understood, without the aid of a schematic diagram.

During the last year, even reservoirs have "gone modern;" incorporating a built-in filter working in the micron range, and air injectors for automatic self-pressurizing. The modern reservoir has acquired, as service mechanics found if they unscrewed the filler cap without heeding placarded instructions, a literal, as well as a figurative, "air of mystery."

By the time the first of these super reservoirs was actually installed in an airplane, the filter, originally designed solely for dirt removal, had become so important that the pumps could

[1] Mechanical Test Engineer, Douglas Aircraft Co., Inc.

Contributed by the Aviation and Hydraulic Divisions and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

scarcely operate without it, and the automatic self-pressurizing device was serving an auxiliary function that alone would have justified its inclusion in the hydraulic system.

Analysis of Filter-Performance Requirements

About 3 years ago, a search was begun to find a filter suitable to protect hydraulic units for 3000-psi systems. In most cases, these units already employed the hardest practical materials, to minimize the effect of abrasive particles. After more than a dozen filter vendors had been contacted, it was obvious that no one had made a thorough analysis of the requirements of such a filter. The problem resolved itself into the following four highly controversial questions:

1 Where should the filter be located in the circuit?

2 What size are the finest particles it must remove?

3 What flow resistance or how large a filter could be tolerated?

4 How long a life could be reasonably demanded, or what service or maintenance responsibilities must be accepted?

There were several possible answers to the question as to where the filter should be installed, as follows:

1 A single filter, through which all the oil must pass before reaching the pumps, would provide the best protection for the entire hydraulic system, provided that it removed small enough particles. The original prospects of obtaining a suitable filter for this location seemed remote indeed, since the location combined the maximum flow rate found anywhere in the system with the minimum amount of pressure available to force oil through the filter. Actual flow rates reached as high as 30 gpm, in the particular airplane for which a filter was immediately required, while even the most optimistic estimates allowed no more than 2 psi to force this amount of oil through the filter, at 70 F, if the hydraulic pump had to provide the necessary suction. Actually, as will be shown later, when the need for pump supercharging is discussed, even this amount of pressure was not available. Since the maximum flow resistance of the filter was to be no more than 2 psi, and since flow resistance increases as the filters accumulate dirt, the resistance of a new clean filter was limited to something on the order of 1 psi. Furthermore, in order to conserve space and reduce weight to a minimum by the elimination of a separate housing, the size of such a filter was arbitrarily limited by the largest object that could be conveniently taken in and out of the hydraulic reservoir under consideration, this being a cylinder about 6 in. diam and 9 in. long.

2 The finest practical filter, with an acceptable flow resistance, could be located ahead of the pumps, and as fine a filter as necessary could be located ahead of each one of the critical units, where flows were relatively low. This arrangement sacrifices a certain amount of protection for the pumps, where it is vitally needed, requires more maintenance, and is contrary to the present emphasis on standardization, because of the wide range of flows and operating pressures which must be accommodated.

3 Another alternative consists of installing a by-pass-type filter at any convenient point in the system, such as the regulator return line. This system obviously offers less protection against abrasive particles than either of the other two systems.

Evaluation of Critical Particle Size

Before making a choice of one of these three installations, it was necessary to determine the smallest dirt particle that it would be advisable to remove. A large number of simple slide valves (as shown in Fig. 2), with 0.0003 to 0.0005 in. initial diametral clearance were built and wear-tested. The wear tests consisted essentially of cycling the valves, while oil containing a known percentage of a known size abrasive passed through them.

A simple hydraulic system was proposed for the cycling test. The slide valve was located between two reservoirs, and oil forced back and forth between them by applying air pressure alternately to one reservoir and then to the other. The need for a filter in an airplane hydraulic system, with hundreds of feet of tubing and dozens of valves and cylinders, was strikingly demonstrated by the troubles experienced when this simple arrangement was used in an initial run to establish a reference datum for operation under supposedly ideal conditions, with clean parts and new oil. It was found to be impossible to operate the valves even a few thousand cycles without scratching.

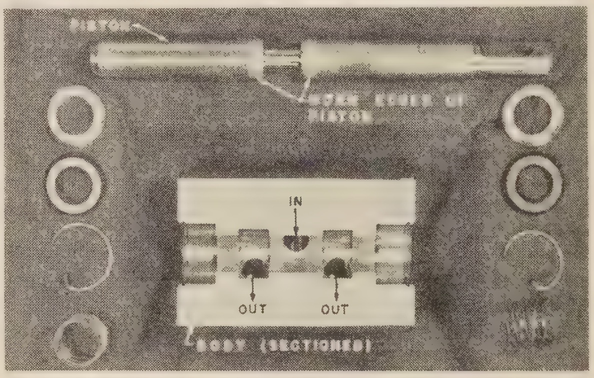


Fig. 2 Construction of Typical Valve Used to Evaluate Critical Particle Size

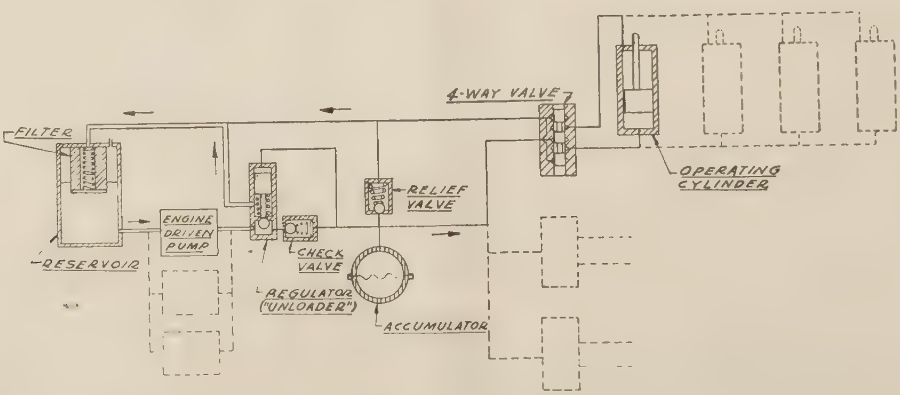


Fig. 1 Schematic Diagram Showing the Basic Parts of an Aircraft Hydraulic System

Finally, after a number of unsuccessful attempts to clean the system thoroughly, a slide valve was operated with only standpipes full of oil at each port. When this valve did not scratch, it was evident that scratching was not inherent in the valve itself, as it was almost beginning to seem, but must be due to dirt.

In slightly revised form, this simplified system was used to test the relative rates of wear of seven grades of abrasives, having nominal sizes ranging from 3 to 58 microns. A micron (μ) is 0.000001 meter, and 1 μ-in. is 0.000001 in., so that 1 μ is 39 μ-in. It is an interesting side light that the very character of the word "micron" may have been responsible, in no small measure, for the success of the entire development program. The filter manufacturers were quick to realize that something about the word attracted popular attention; its applicability to future advertising may well have mitigated the usual difficulty in obtaining research funds necessary for this type of work. Now, after three years, the word is widely used throughout the aircraft hydraulic industry, symbolizing a distinct change in the concept of satisfactory filter performance.

All the abrasives used in the test program caused some change in the valves, but there was a tremendous difference depending upon the particle size. From a study of the leakage measurements, wear measurements, and appearance of the pistons, some of which are shown in Fig. 3, it was decided that all 10-μ-size particles should be removed, and as many as possible of the 5-μ size. Indications were that particles as small as 1 μ would not be very serious.

In order to obtain some idea of the amount of dirt that a suitable filter must be capable of absorbing, the dirt content of several samples of hydraulic fluid that had been in service for 1000 hr was measured. From this it was estimated that the filter should be capable of absorbing 2 or 3 cu in. of dirt before requiring service or replacement, on the basis of service periods not more often than every 2000 hr flying time.

Thus the requirements for the type of installation in which all the oil is filtered each time it flows through the reservoir may be summarized by the following:

1 Remove all particles larger than 10 μ.

2 Pass 30 gpm at 70 F, with 2 psi pressure drop, after collecting 3 cu in. of dirt.

3 Occupy a space no larger than 6 in. diam $\times$ 9 in. length.

These requirements are so difficult to meet that for a long time it was necessary to maintain active interest in the other types of filter installations that have been mentioned, so remote did it seem that any satisfactory filter of the type 1 would be found. The first thing that was brought out by detail tests on various filter media, particularly felt and paper, was that a tremendous surface area would be required for such a filter, and that the thicker the filter medium, the greater would be the required area. This narrowed the field to the use of paper, or some other very thin filter medium, for in no other way could so much area be built into such a small space.

The problem of the filter manufacturer then became one of arranging the paper in the best possible manner. If too much paper is used, the internal passageways will become too small and tortuous, and may perhaps clog up, shutting off large areas of the filter. If too little paper is used, the flow rate per unit area will be high, and so will be the pressure drop. As the result of considerable development work, several successful designs are now available.

Approval of the retentivity of the filter papers used in these designs was granted after tests using the same graded abrasives that had been used for the wear tests.

Measurement of Filter Flow Resistance

The measurement of flow resistance through such a filter proved to be a more interesting matter than had been anticipated, for it was here that the filtering and pumping problems became interwoven. In the first place, the flow rate into the particular reservoir for which this filter was originally designed is quite high, amounting nearly to 5 changes per min. The reason for the high flow is that airplane operators feel it is well worth while to be able to retract the landing gear in a matter of seconds, and, in this case, 45 hp is available for the purpose. At 3000 psi, this requires a flow of about 25 gpm. (When this figure is compared to the 75 gpm that it would take to do the same work at 1000 psi,

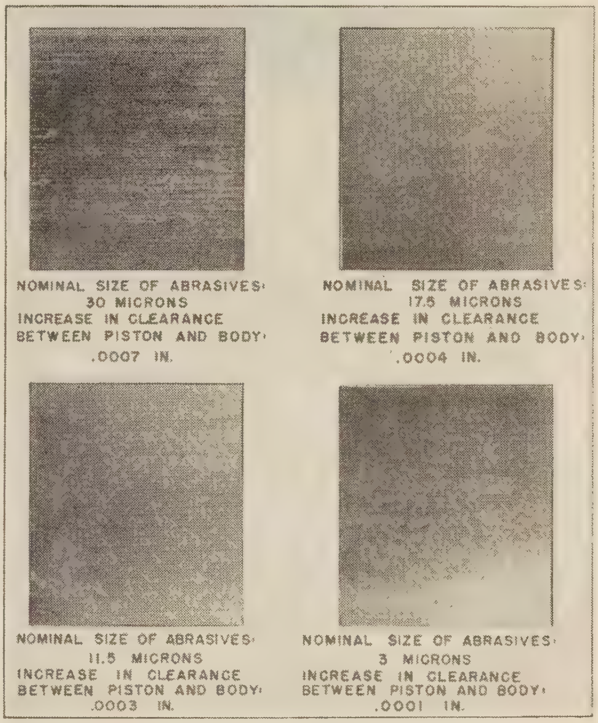
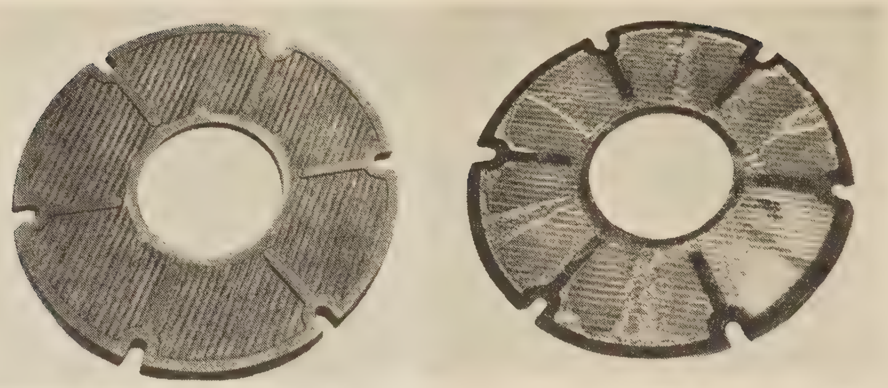


Fig. 3 Partial Summary of Wear-Test Results

the reason for widespread interest in high-pressure hydraulic systems is obvious.)

It had been the original intention to discharge the return oil into the reservoir tangentially, and suck it out to the engine-driven pumps through the filter. When an attempt was made to measure the flow resistance of the filter using this arrangement, it was not only impossible to obtain accurate readings, but the energy in the return oil was sufficient to set the entire content of the reservoir in rapid rotation, promoting so much air entrainment and foaming that it was virtually impossible for the pumps to operate. A number of modifications were tried without success. Finally, the filter was utilized as a diffuser, through which the oil was discharged into the reservoir. In this case, instead of entering tangentially at 30 fps the oil entered radially with a velocity of less than $^1/_2$ fps. This arrangement effectively eliminated all troubles due to foaming.

In addition, this revised installation yielded a particularly accurate means for measuring flow resistance of the filter. A hole was simply tapped into the unfiltered oil region, inside the filter, and a standpipe extended up from it. Oil was circulated through the filter at the required rate, and the pressure drop was determined by measuring the height of the oil in the standpipe, above the oil level in the reservoir. By injecting measured



(a) (b)

Fig. 4 Sample Filter Disks; Filter Assembly Consists of Many Such Disks
(a, Shows amount of dirt collected during 1000 hr of service in an airplane. b, Shows amount of dirt collected during test without causing objectionable flow resistance.)

amounts of dirt into the oil just ahead of the filter, the dirt capacity of the elements was investigated, and found to exceed the requirements originally estimated. This completed the final tests necessary to fulfill the predetermined standards of performance.

In Fig. 4, a sample filter disk from a laboratory-tested filter is compared to a disk, obtained at a much later date, from a filter assembly with 1000 hr of aircraft service. The ample dirt-storage capacity of the design is obvious. Unfortunately, the illustration does not emphasize the fact that, despite its relatively clean appearance, the service-tested disk has actually caught a large number of dangerous dirt particles.

While the completion of these filter tests provided considerable reassurance with respect to slide valves, where abrasion was the primary cause of concern, additional problems were anticipated with respect to the piston pumps, due to cavitation.

Analysis of Pump Operating Conditions

Probably there should be some explanation as to why pumping problems still exist, after airplane hydraulic systems have been built for so many years. For example, operators are demanding satisfactory operation at higher altitudes, and more hydraulic operations are being required at altitude conditions. Whereas, the commonest employment of the hydraulic system has hitherto been landing gear and wing-flap operation near airport levels, now surface-control boosters, turrets, bomb doors, etc., find frequent use at service ceiling. Even if the hydraulic pumps are running unloaded, injurious cavitation may occur at high altitude, at low temperature, or at high pump speed. Furthermore, the trend to larger airplanes moves the pumps farther and farther away from the hydraulic reservoir, perhaps 30 to 50 ft. It is frequently impossible to locate the reservoir higher than the pumps, and the line connecting the reservoir and pump must be kept as small as possible to reduce weight and facilitate installation.

Operation of pumps under conditions of cavitation can be dangerous in at least four different ways. If carried to the extreme, the hydraulic system will be rendered useless because of lack of oil supply. Before this condition is reached, pump efficiency may fall so low that overheating and seizure may result. As an even earlier effect, the extreme vibration attendant upon partial cavitation may cause fatigue failure of the pump shaft. Still another effect is the formation of small pits frequently in parts which must remain smooth for sealing purposes.

Sufficient study has been made of the pressure losses between reservoir and pump to conclude that some form of supercharging is justified from the weight standpoint, and is the only answer for high-altitude operation.

The minimum allowable absolute pressure in the hydraulic-system reservoir is essentially the sum of the following major pressure losses:

1 Line loss, plus the difference in head between pumps and reservoir.
2 Valve-port loss, getting the oil into the pump cylinder.
3 "Unavailable pressure," due to vaporization of oil and to dissolved or entrained air.

If atmospheric pressure at the altitude of flight does not exceed the sum of these losses, cavitation will exist unless supercharging is provided.

It is not necessary to determine the magnitude of these losses with scientific accuracy in order to determine which factors are most important from the standpoint of making a general improvement in pump operating conditions.

Survey of Pressure Losses on Inlet Side of a Pump

Line loss may be readily calculated from well-known and reliable formulas, which are further simplified because the calculations of interest usually fall in the laminar-flow range. Such calculations show that the use of larger lines to reduce pressure loss is extremely limited and costly. The use of the next size larger lines in one particular airplane would involve a 23-lb increase in weight, and save only slightly more than 1 psi line loss at 70 F, or, if viewed from the temperature standpoint, would permit operation with oil about 20 deg F lower in temperature. Temperature has such a tremendous effect on line loss that it is impossible to take care of very cold oil by increasing line sizes. Fortunately, the flows are usually low when the oil is cold, because under these conditions the main engines, which drive the pumps, are likewise cold and must be run slowly. As the engines warm up, the oil may be expected to warm up also, due to the friction of circulating it through the lines. Of course it is always possible to provide additional resistance in the circuit, if necessary to accelerate warm-up, at the expense of adding a thermally controlled relief valve.

Losses through the various expansions and contractions at the inlet to a pump cylinder may be approximated using formulas and coefficients available in any hydraulics textbook. It requires a bit of optimism to trust such calculations to any great

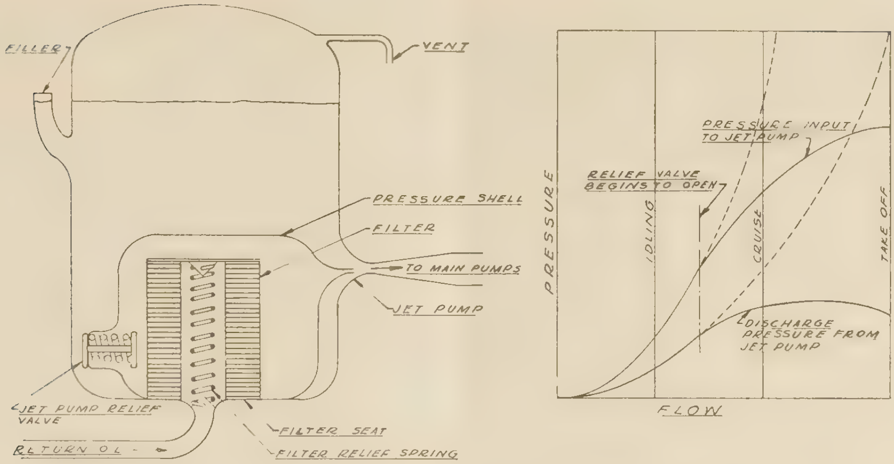


FIG. 5 JET-PUMP METHOD OF PUMP SUPERCHARGING; RESERVOIR UNPRESSURIZED

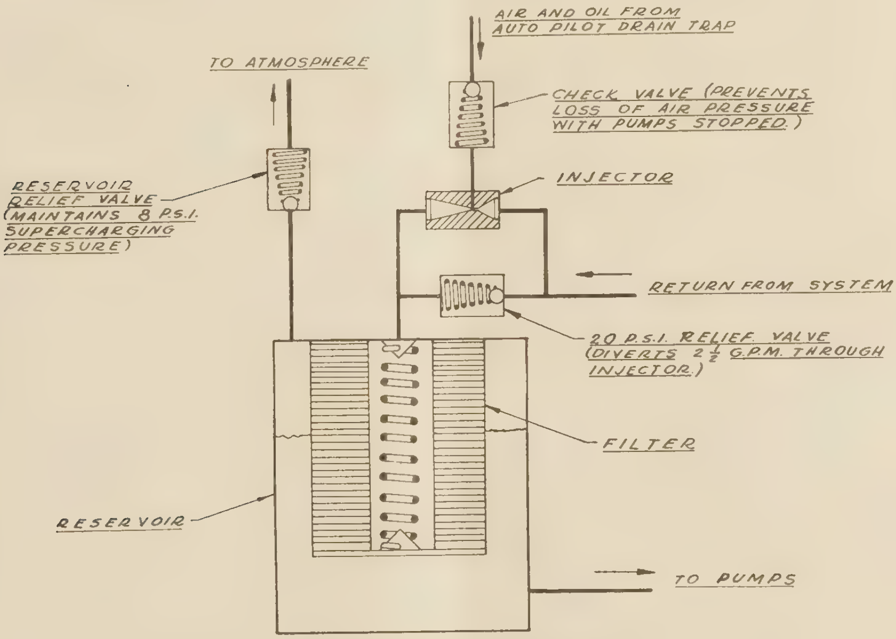


FIG. 6 AIR-INJECTION METHOD OF PUMP SUPERCHARGING; PRESSURIZED RESERVOIR

extent, because the resemblance between the passageways in the pump and those in the textbooks is rather remote. Furthermore, there is a surprising lack of agreement among texts as to the coefficient for sudden contractions under ideal conditions. The calculated losses for one particular pump amounted to 2.7 psi, which, when added to the other calculated operating losses, yielded a total that was quite closely confirmed by test. This restored a measure of confidence in these methods, which, if correct, bring out the point that even if the port losses could be cut in half the increase in safe operating altitude would be small.

The loss due to oil vaporization is in a similar category. It is probably impractical to effect any improvement, but, before making such a decision, it is necessary to have some idea as to what the magnitude of the loss might be. The fact that some oil actually turns into vapor in a pump chamber is substantiated by an observed decrease in volumetric efficiency, above a critical speed, for a positive-displacement pump, operating at constant discharge pressure. Since the pump chambers are not running full of oil, the remaining space must be occupied by vapor. This would seem to indicate that an extremely low absolute pressure is reached in the pump, since one vendor of hydraulic fluid quotes a vapor pressure as low as 5 mm of mercury at 175 F. However, this vapor-pressure figure may be very misleading, because oil, being a mixture of hydrocarbons, has no fixed boiling point.

Vapor pressure is frequently measured by allowing the liquid to vaporize until it fills a space several times as large as the liquid volume. In pumps, the vapor volume becomes important while yet a small fraction of the liquid volume. For this reason, the vapor pressure of interest to aircraft hydraulic engineers is not the vapor pressure that is likely to be quoted in good faith by a chemistry laboratory.

When this point was investigated further, measurements of the gas volumes developed from hydraulic fluid, maintained at

150 F, and a vacuum of 25 in. of mercury, until gassing stopped, yielded a value nearly equal to one half the liquid volume. In a hydraulic pump, the liquid has only an instant to flash into vapor, so that there is an additional factor, time, of unknown importance. Furthermore, no line has yet been drawn to define the critical ratio of gas to liquid volume. The one item which has been established is that the pressure rendered unavailable to the pumps, due to oil vaporization, is of the order of 5 in. of mercury, rather than 5 mm.

If this figure of 5 in. of mercury is accepted as the minimum absolute pressure normally attainable in the cylinder of a pump handling hydraulic fluid, then the importance of entrained or dissolved air becomes greatly minimized. For instance, the fluid would have to contain 14 per cent air, by volume, at sea-level pressure, in order to amount to half air and half liquid at 5 in. of mercury abs. However, if the pressure could fall to 5 mm of mercury abs, it would take less than 1 per cent of air at sea-level pressure to reach this same ratio.

When all of the foregoing losses are considered, it is apparent that the volumetric efficiency of the pumps on an airplane would begin to decline, well below the service ceiling demanded by current airplanes, even if the pumps were mounted in the hydraulic-system reservoir.

No one is prepared to say how low a volumetric efficiency is safe for continuous operation, but there is obviously no reason to take chances, if a supercharging pressure can be readily obtained.

Methods of Pump Supercharging

Various systems for boosting the inlet pressure at the engine-driven pump have been suggested, but most of the obvious ones can be discarded because they involve moving parts, or make the reliability of the hydraulic system depend upon the reliability of some other apparatus that is relatively complicated. The author's company has developed a satisfactory arrangement using a jet pump, Fig. 5, but this was not selected, because another method is available which pressurizes the reservoir with equipment that is already required to perform another service.

The system finally adopted, Fig. 6, employs a simple Venturi, which, acting as an air injector, builds up a pressure of about 8 psi in the sealed hydraulic-system reservoir. In addition, it serves to inject into the reservoir the hydraulic leakage from several unpacked lap-fitted slide valves that are component parts of the autopilot system. The latter function requires the Venturi to provide a subatmospheric pressure at the autopilot drain connection, at all times when the engines are running; that is, the Venturi must never stall, either because the reservoir pressure becomes too high for it to pump against, or because the fluid flow through it is reduced.

In order to insure that stalling can never take place, it is necessary to provide the Venturi with a flow that is never less than a predetermined minimum, and set the reservoir relief valve, which governs the pressure against which the Venturi is pumping, at some pressure less than that which the Venturi is capable of producing, when supplied with minimum oil flow. The flow provision is a simple matter. Whenever the engines are running, oil is circulated through the reservoir. It is only necessary to add a spring-loaded valve in the reservoir return line, and locate the Venturi in parallel with it. Since too much air added to the oil tends to defeat the basic purpose of the system, which is, of course, to force solid oil into the pumps, it is wise to design this spring-loaded valve so that its resistance is essentially independent of flow, and thus introduce no more air into the oil than is absolutely necessary.

The rate of introducing air was chosen so that the supercharging system would maintain sea-level pressure in the reservoir, if the airplane took off with an initially unpressurized reservoir and climbed directly to its service ceiling. Ordinarily, the reservoir will never be unpressurized, because a check valve in the air-inlet line prevents the escape of pressure back through the Venturi when the engines are stopped.

Every pound per square inch of pressure in the reservoir is purchased at the expense of about 3 times that much "back pressure" on the oil returning to the reservoir; that is, the nominal setting of the spring-loaded valve which controls flow rate through the venturi is roughly 3 times the minimum amount of supercharging pressure that is provided. This additional pressure is expended in overcoming the resistance of the check valve in the air-inlet line, in overcoming pipe friction, venturi inefficiency, and pressure loss through the filter, and in providing a margin for tolerance on the pressure settings of the reservoir relief valve and the valve controlling venturi flow rate.

Each major unit in the hydraulic system has had its share of notoriety in past years; the power brake valves, pressure regulators, accumulators, surface-control boosters, brake deboosters, emergency air valves, 3000-psi pumps. It is truly a fitting reward for loyal service that the lowly and obscure hydraulic reservoir, now with built-in pump supercharging and micron-range filtration, at least achieves in some measure, the attention that has been accorded its associated units.

Aircraft-Engine Temperature Control

By WILLIAM A. RAY,[1] GLENDALE, CALIF.

Exposed to variations of as much as 200 deg F in ambient temperatures over short periods of time, the aircraft-engine temperature-control problem is a vital consideration of design and operation both of military and commercial aircraft. Weight, compression ratios, combustion rates, factors of safety, efficiency, ambient temperatures, and general operating conditions are all factors contributing to the necessity of controlling engine temperatures. The author undertakes an objective analysis of the problem and describes a system of hydraulic control which has been developed and which has proved satisfactory in service.

A PROPER appraisal of aircraft-engine temperature control should be based upon an objective view of the engine itself, its work load, and its surroundings, as compared with typical internal-combustion engines with which most engineers are familiar, either in stationary or automotive work.

The most obvious difference between internal-combustion engines and aircraft engines is that of weight; the former seldom being lighter than 5 or 6 lb per hp, whereas aircraft engines will average just over 1 lb per hp. Combustion rates and compression ratios are high on aircraft engines, whereas the factor of safety on mechanical stresses is generally low by comparison. On the heavier internal-combustion engines, high factors of safety on mechanical stresses are used to overcome those normal defects which occur in all mechanical structures but which cannot be economically sought out, at least up to the present time, on automotive or stationary engines. Aircraft engines necessarily light and safe from an economic standpoint justify extreme care in manufacture. There are no marginal factors of safety to cover these typical manufacturing defects. All aircraft engines must be normal or better.

On the operating side, aircraft engines also have their distinct differences. Over-all operating efficiency must be a maximum. Inefficiency cannot be tolerated since extra fuel would be required, thus seriously reducing pay load and cruising range, particularly on ocean flying where landing fields may not exist for 2000 or 3000 miles.

With the average internal-combustion engine employing customary cooling methods, it is usually impossible to operate at too hot or too cold a temperature from a safety standpoint. However, this can easily occur on an airplane, since any substantially fixed cooling arrangement, which may be used on most other engines, can never meet all conditions that occur on an airplane, with the result that an engine may get too hot, resulting in fire; or too cold, resulting in misfiring. The latter often occurs, particularly on military airplanes, when the need for maximum power exists, as, for example, when pulling out of a glide dive.

The aircraft engine is also subjected to wider ambient temperatures on the cooling medium than almost any other type of engine, it being quite possible to have changes in the temperature of the air medium through which the plane is flying approaching 200 deg. The effectiveness of this cooling medium is further complicated by the variation in speed of the airplane, most variations of necessity having no relation to the cooling needs of the engine.

Furthermore, from an operating standpoint, for the sizes of the engine involved or horsepower output, all airplane engines are warmed up faster than any other type, particularly on military planes which are subjected to widely variable and sudden load changes.

These questions of weight, compression ratios, combustion rates, factors of safety, efficiency, ambient temperatures, and general operating conditions have all contributed toward the necessity for control of engine temperatures. All these factors likewise accentuate the time function, requiring both responsive and watchful temperature control.

Analyzing the Engine-Cooling Problem

An objective analysis should include a scrutiny of the cooling job itself. There are normally three ways that heat may be dissipated from aircraft engines, two of which are always used depending upon the type of engine, that is, whether it is air-cooled or liquid-cooled. Both types dissipate a great deal of heat to the lubricating oil, in contrast with stationary-engine practice where little or no effort is made to dissipate heat through this medium. However, the design of aircraft engines for higher work rates makes it imperative to dissipate substantial amounts of heat through the lubricating oil. Thus the lubricating oil, distinctly a cooling agent and subjected to extremes of temperature may catch fire at one extreme, yet freeze to a solid at the other. This heat in the oil is usually dissipated through a radiator, the effectiveness of which is dependent upon the atmospheric conditions through which the airplane is flying. The air-cooled engine dissipates additional heat from cylinder walls or finned heads. Here again extreme temperatures may at one end result in loss of strength of the engine parts, and at the other end misfiring, because cylinder and head temperatures affect ignition. The liquid-cooled engine, originally using water as a coolant, now uses a chemical (ethylene glycol, commonly known as prestone) as a coolant, because of the more desirable properties of this chemical as compared with water. On a liquid-cooled engine this coolant collects that heat occurring at the cylinder walls and cylinder in the same manner as in the air-cooled engine. The coolant heat is then dissipated by a radiator in the air stream. Here again extremes of temperatures will result in engine failure or misfiring.

Hence it is important to control the air flow over these cooling elements, whether in the form of the lubricating-oil radiator, the prestone radiator, or the air-cooled engine cooling fins. This control is usually obtained by suitable flaps that permit or restrict the air flow. These flaps are desirably located around the engine, a point more often than not remote from the pilot's control, making it necessary to obtain some kind of a remote operation of the flap. Mechanical operation is not always satisfactory because of the work loads involved, and the difficulty of working the operating gear through the airplane structure. The result is that on most airplanes, a more flexible, outside, work-doing medium is used to power the flaps and secure remote control. Until quite recently, this remote control has been mainly manual. The pilot or flight engineer periodically checked engine temperatures and corrected the flap position as required.

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Thus on the typical airplane, there has existed a means for operating the temperature flaps, so in order to secure automatic control, it has only been necessary to provide some automatic piloting method to control the operating medium. The sensitive thermal element of the automatic control is usually more accurate, minimizes the human element, and stays on the job at all times.

The ease with which automatic temperature control can be adapted to the usual flap-operating gear, also makes it possible, in most cases, readily to switch back to manual control for such occasions when only human perception can decide the best condition, such as for take-off, landing, or fast warm-up. However, if the ship has automatic controls sensing such functions, they can be made automatic as well. Thus if desired to minimize head resistance, as when taking off heavily loaded ships, flaps can be retracted more than usual; or when landing, when head resistance may be desirable, flaps can be opened more than usual.

Hydraulic Controls Developed

Development work, as done by the author's group, has hinged around hydraulic operation, as it was felt that this actuating medium was more reliable and more easily repaired, because of its mechanical simplicity and the ease with which faults in the system may be detected. Electric operation, lubricating oil under pressure, air under pressure or vacuum, and even direct thermal operation are possible alternatives. Direct operation, consisting of a thermal unit powerful enough to position flaps, is not practical for larger engine installations.

However, hydraulic operation as mentioned, is believed to be admirably adaptable to most temperature applications and readily lends itself to constant operation without destruction either to control valves or operating cylinders, has very few moving parts, and the weight is well in line. Hydraulic operation has further advantages in that in most cases of system failure, the flaps will fair or trail. Although maximum engine power is not always obtainable with faired flaps, and retirement from combat may be necessary, sufficient engine power is available for safe operation of the airplane. In fact, on most aircraft, the design is such that the flaps will fair or trail at cruising speed.

The selection of hydraulic oil as the working medium resulted in the initial development of a temperature control that is adaptable either to lubricating oil or engine coolant. This particular control is operated by a sensing element or thermal element of the liquid-expansion type. This element was selected because of its great work-doing possibilities, compared to other types of sensing elements, such as bimetal expanding or bending members, thermocouples, vapor-tension or gas expansion. A liquid element is particularly suited to aircraft work in that it is not responsive to altitude conditions; hence it holds its temperature calibration regardless of atmospheric pressure, provides a hard working force, and in general, does not require compensating means or glands. Development work has also overcome the slowness of response of this type of unit to a point at which it is comparable, for example, with bimetal immersed directly in the fluid.

This liquid-charged element operates a four-way selector valve which, in a sense, is an amplifier, since a small work load of the order of 0.01 to 0.02 in-lb will cause or prohibit the operation of 1 hp of net work at the flap, the smaller work being provided by the thermal element. The four-way selector-valve operation controls the flow of hydraulic oil usually at 1500 psi pressure to the hydraulic cylinder or strut, which in turn operates the air-controlling flap. This arrangement is additionally effective in that hydraulic oil or pressure is readily convertible to force in its most usable form in so far as flap operation is concerned, hence the efficiency of this working medium is very high.

The most desirable type of control is one that will modulate the flap, that is, not open or close only, but assume some position best suited to obtain the correct temperature control. Operating efficiency of the engine installation virtually requires modulating control, not only to maintain actual engine efficiency by holding correct temperatures, but maximum aerodynamic efficiency by lessening unnecessary flap opening. However, modulating controls have an inherent weakness, that is, a tendency toward instability, to "hunt," or to oscillate about the desired position. This is overcome in nearly all modulating mechanisms by what is known as the reset or follow-up mechanism. This mechanism readjusts the calibration point of the temperature control for the particular flap position. Thus when the engine is cold, and the flaps are closed, the temperature control will respond at a temperature 20 deg, 30 deg, or 40 deg F, as the case may be, lower than it would if the flaps were open.

Modulated Temperature Control

A typical modulating control does not have a fixed temperature-responsive point in the sense that we understand the typical thermostat to have. For example, a typical on-and-off thermostat would turn on at a certain temperature, say 210 F, and turn off at 220 F. The 10 deg F difference is known as differential, and the operated unit would open or close, only in response to such a control unit. The modulating control may start at 210 F and initiate opening of the flap, the resultant operation of which would shift the range of the control, resulting in a small movement suited to the cooling needs. The flap would not open fully, but would open, for example, at 20 per cent, and additionally, the control point or response point of the control would be shifted from 210 to 216 F. Another rise in temperature would cause a further opening of the flaps, an increment suited to the rise of temperature. The flaps would be fully open, for example, at 240 F, and the new response point of the control, because of the reset mechanism, would be at 240 F. The temperature dropping some amount below 240 F would result in a closing of the flap again an incremental amount as needed. Thus with a reset mechanism, an actual closing condition is obtained which may be higher in temperature response than another point at which the flaps would open. The reset mechanism is solely responsible for this condition, and ideally prevents "hunting," or oscillating about the control point, or unnecessary flap operation.

The range, in this case, from 210 to 240 F is known as the reset range, and is here 30 deg F. As compared to the on-and-off control, first mentioned, it appears to be wide, and would result in a wide controlling temperature. However, the on-and-off control is subject to swing in control. For example, a control which shuts off at 220 F, as just outlined, does not result in holding the control medium at 220 F. A swing occurs which will carry it greatly beyond this point, depending upon the characteristics of the medium and system on which the control is being used. A properly designed reset range, and it should always be wide enough to be so designed, will result in holding the control temperature entirely within that range, particularly if the control is rapidly responsive. Hence although the reset range of the modulating control is wider than the differential of typical on-and-off control, the actual control temperature is more often closer or narrower than that obtained with an on-and-off type. Therefore, a reset mechanism was included in the control, not only to prevent "hunting" but for closeness of control.

Pilot supervisory control can be exercised taking the flap control completely away from the automatic control. This can be done mechanically, electrically, or hydraulically as the work

involved is slight, and hence reasonable, regardless of the medium used. The medium to be used depends upon the remoteness of the location as well as upon the reliability of the types available. The use of certain mediums is precluded because of inaccessibility. The particular control in question may be overridden either mechanically, electrically, or hydraulically. Each will work equally well. The mechanical method of overriding has the obvious advantage of providing a direct operation in the event of either electric- or hydraulic-system failure, particularly in view of the fact that the hydraulic temperature control will fair the flaps upon proper operation of the selector valve. However, electrical or hydraulic operation is used where it may be physically impossible to reach the control mechanically. The overriding control thus provides four positions, i.e., automatic, open flap, closed flap, locked flap. Predetermined flap positions are possible or manual modulation may be used, according to needs.

An engine temperature control using a hydraulic work medium is successfully operable from either lubricating-oil or engine-coolant temperatures. In addition, operation is fast enough to cover all flight conditions, even those existing in pursuit ships. In fact, actual test curves on one of the fast pursuits shows that the control is never lost—never has to overshoot to catch up with its position. Also, overriding or pilot supervision is just as effective as if the temperature control never existed. The weight is low. Field experience and operating life have been satisfactory.

The development and subsequent operation of this contro have demonstrated the excellent possibilities of hydraulic operation of engine temperature control. Further advancement gives indication that air-cooled-engine cowl-flap control may be solved hydraulically. Advances in the art, accelerated by the substantial production now existing, will most certainly result in even greater reliability, lower weight, lower cost and accuracy for any aircraft temperature-control problem.

High- and Low-Pressure Airplane Hydraulics in Europe

By JEAN MERCIER,[1] NEW YORK, N. Y.

Development of hydraulic control systems for aircraft began to make progress in Europe about 1936, the trend being toward pressures of 5000 psi and higher. Details of the equipment comprising the hydraulic systems, applications made, and lines of development when the war interrupted the program are treated in this paper.

It may be recalled that during the heroic phase of the airplane in the first world war, the kinetic energy of any abnormal landing shock was absorbed, as is said in French, by "breaking wood." The wood referred to was, of course, the wooden undercarriage, even when it ceased to be made of wood as originally.

So many of the Allied planes were thus immobilized that the author's father, Henri Mercier, who at the time was Director of the Mechanical Section of Inventions of the War Department in France, suggested what he and his colleagues termed "a pneumatic shock absorber with a very long stroke!" Compared with the shock absorber of today, the very long stroke, which was 30 in., will surely bring a smile. To avoid any drag which might disturb the flight, a telescopic retracting system was developed.

The Armistice of 1918 interrupted this research before any test flights were made. Incidentally, the fluid retraction of the landing gear was not hydraulic but pneumatic; a small Letombe compressor actuated by a propeller was the prime mover.

Twelve years passed before another French engineer, the late Georges Messier, took up the problem. His was a hybrid development. By combining Col. Deport's oleo-pneumatic gun recoil with samples of the American Lockheed automobile hydraulic-actuating cylinders, he built probably the first modern hydraulically retractable landing gear. The Olaer Company, which the author directed, was working on similar problems for automobiles at that time and decided to enter the airplane field. To have the best chance against competition by electric controls using high voltage, it seemed necessary that an equivalent high "voltage" hydraulic system, foolproof and unaffected by wear, tear, cold, and age, be first scientifically developed.

Advantages of High Pressures

The following illustrations may help to clarify the analogy between high voltage and high pressure in hydraulics:

A few volts permit the handling of a good-sized telephone bell; a few pounds' pressure are just good enough to actuate a windshield wiper; 1000 lb will hold its own against a 20-v motor. The future will certainly bring into competition systems of several hundred volts, such as are already being applied in ships with installations of several thousand pounds per square inch. In other words, high pressure is comparable to high voltage, thus affording an efficient way of producing and transporting energy without the corresponding trouble of handling great volumes of oil or conducting many amperes of electricity.

The enormous advantages of high voltage are thus balanced, and hydraulics will be free to develop its potentialities of easy maintenance, light weight, and flexibility. The value of these qualities is particularly potent in the high-pressure hydraulic accumulator which is able, in a fraction of a second, to start off the heaviest engines.

Having adopted this point of view and before determining the optimum operative pressure, it was essential to learn the reaction of European aeronautical engineers and test pilots compelled to fly with apparatus under such apparently eruptive pressures. It is hard to believe the great difficulties encountered by the author in promoting the idea that the damage caused by the explosion of an accumulator is related to the amount of energy thus liberated, e.g., the energy stored in a 50-cu-in. accumulator at 5000 psi is not much greater than 1 gal at 1000 psi.

The best illustration of the present status of the art in the United States is Harold Adams' very interesting work,[2] in which the optimum operating pressure is calculated, and which proves that above 3000 psi there is practically no saving of weight, either in tubing or in actuating cylinder walls, if the same load factor is maintained for both high- and low-pressure hydraulic systems.

While the author is in full agreement with Mr. Adams' calculations concerning the parts involved, he has found that above 3000 psi there is still a marked saving in cylinder heads and tube fittings, as well as in the actual weight of oil involved. In addition to this, higher pressures should be of great interest for the following reasons: The smaller the apparatus, the less will be the modifications necessary in the structural design of the plane, thus saving an incredible amount of weight, space, and last but not least, headaches. One is led to the conclusion that too low a pressure, even 3000 psi, limits the free development of optimum structural design.

Furthermore, a constant pressure drop in the line of a few hundred pounds is necessary to transmit the energy from the pump to the actuating cylinder. Those hundred pounds become a relatively smaller percentage in a 9000-psi system than in one of 3000 psi. The corresponding tubing, being necessarily thicker, resists external stress and vibrations so well that the factor of safety can be reduced to the one adopted for cylinders, thus giving nearly 50 per cent weight saving. Once again, there is a point of comparison with the high voltage in electrical-power transmission.

TABLE 1 WEIGHT SAVING COMPARISON BETWEEN 5000-PSI AND 1000-PSI HYDRAULIC SYSTEMS

	Weight saved, lb	Per cent saved
Tubings, fittings, and supports	5	50
Oil	30	75
Pump	0	0
Selector valve	1	50
Landing-gear cylinders	5	60
Dive-flaps cylinder booster of pressure	11	70

As an illustration of weights saving, Table 1 summarizes the case of an airplane weighing 7000 to 8000 lb, with a hydraulic

[1] Consulting Engineer, Simmonds Aerocessories, Inc.; President, Ol-Aer Patent Company.

Contributed by the Aviation and Hydraulic Divisions and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

[2] "Design and Shape Problems in High-Pressure Hydraulic Systems," by Harold Adams, *S.A.E. Journal*, September, 1940.

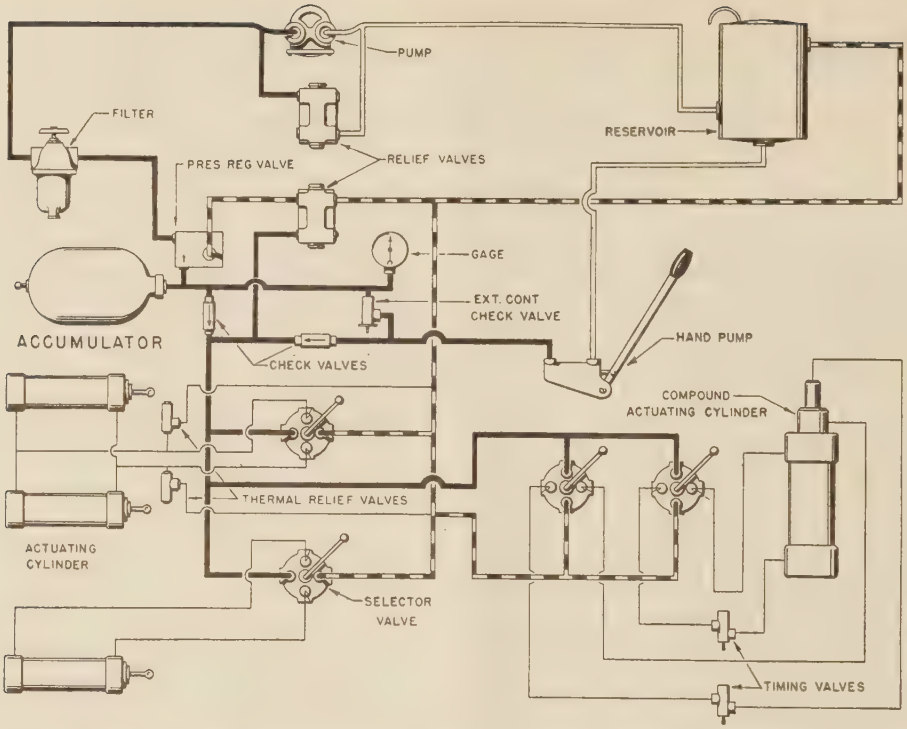

FIG. 1 SCHEMATIC DRAWING OF TYPICAL OLAER MERCIER HYDRAULIC SYSTEM SHOWING INSTALLATION OF ACCUMULATOR AS IN REGULAR USE IN FRANCE IN 1936, AND AS IN USE ON AMERICAN AIRCRAFT IN 1943

(When a variable-delivery pump, Dowty, England; Olaer Mercier, France, is used, the pressure-regulator valve is suppressed.)

system utilizing 5000 psi, compared to one at 1000 psi (Fig. 1). In this instance, the total weight saving appears to be over 60 per cent.

In Europe, prior to 1935, to the best of the author's knowledge no planes had flown with a hydraulic system operated at a pressure appreciably over 1000 psi.

It was difficult to get the French Bureau of Aeronautics to have little more than faint sympathy for such evident madness as the proposal of the Olaer Company. However, after 2 months of completely successful trials at 8000 psi, it was agreed, much to the relief of the bureau, to drop the pressure to 4000 psi. In 1935, the Morane Saulnier fighter 405–406, equipped with Olaer-Mercier hydraulic system at 4000 psi, took off from the Villacoublay Airport, piloted by an amazingly confident man, Michel Detroyat.

HIGH-PRESSURE SYSTEMS ADOPTED FOR MILITARY AIRCRAFT IN 1936

Interesting items at that time were the automatic thermostatic actuation of the selector valve, controlling the position of the radiator-actuating cylinder and the follow-up flap control, permitting their immobilization at any degree of their travel. There were, also, separate piston-type accumulators with metallic glands, permitting independent emergency movements of flaps and undercarriages.

In 1936, the Fokker Gl light bomber adopted the same system at an operative pressure of 4500 psi. Both types of planes have since been manufactured in regular line production. The basic scheme of the hydraulic system of both of these planes is the same. In the Fokker, the following functions were controlled by the Olaer-Mercier system: Retractable landing gear with mechanical lock hydraulically operated by special cylinder, landing flaps with a safety valve allowing for automatic adjustment at high speed, dive flaps, bomb doors, cockpit opening, and cooling flaps.

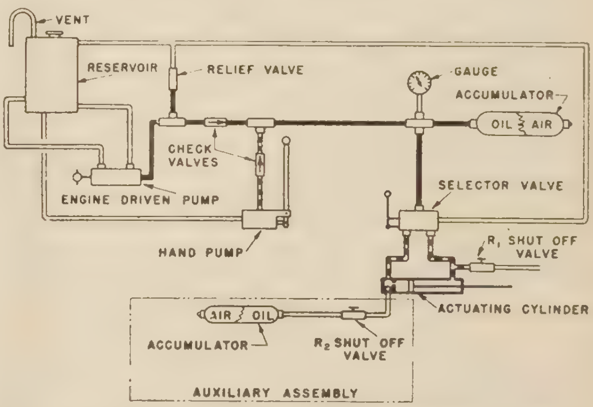

FIG. 2 SCHEMATIC DIAGRAM SHOWING ELEMENTS OF TYPICAL OLAER MERCIER INSTALLATION WITH SELF-REGULATING PUMP

(Note independent circuit and emergency outlet. In case of pressure failure in line 1, pilot opens valve R_2 and, if necessary, R_1 to lower landing gear. In use in Holland, Switzerland, Italy, France, 1936.)

Special mention must be made of the emergency hydraulic pressure conducted to the hydraulic-flap and landing cylinders, utilizing an automatic transfer valve on the cylinder head, isolating the normal hydraulic circuit in case of its failure (see Fig. 2). The energy for this was provided by an air bottle or an accumulator preloaded at 5000 psi, thus doing away with the hand pump. Bristol, in England, and others use a similar transfer valve to connect a gas-charge-producing cartridge device. At the same time, Caproni in Italy acquired the Olaer license in order to equip its series of light bombers 310. The chief char-

acteristic of this modified installation was a complete dual hydraulic system up to the jack, with two independent pumps, one on each engine, the pilot thus having the choice of shifting from one to the other, provision having been made for automatic transfer valves. Since then, the high-pressure hydraulic system has been adopted in Switzerland, and in Czechoslovakia by Skoda for its Avia 35. In 1938, the French fighter Dewoittine 520 also adopted Olaer. The different components of this system were on view, in operative condition, at the hydraulics meeting in Los Angeles, November, 1941, under the chairmanship of Commander Harry Marx of the Bureau of Aeronautics, and of Mr. Bashark of Wright Field.

The parts shown consisted of a complete retractable landing gear, composed of a magnesium wheel, specially coated with metallic salts, resisting 1000 hr of exposure to salt spray (which coating process is now in use in Germany and was recently found satisfactory by Northrop laboratories), and of brakes of the inflated-bag type having an extremely large brake drum located inside of the wheel contour, thus enabling the dispersion of the heat generated when landing at very high speed, i.e., 100 mph. This last-mentioned characteristic was found necessary, if not always for actual landing, at least for taxiing rapidly along the ground to shelter in war operations.

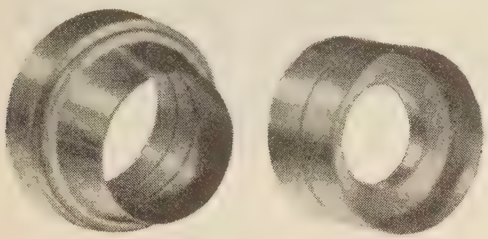

Fig. 3 Olaer Mercier Metallic Gland for Pressure Up to 30,000 Psi and Temperature Range From —100 F to +400 F
(In use in France, Holland, and Switzerland since 1936.)

Shock struts were provided with the Olaer metallic glands (Fig. 3), adequate to resist excessively low temperatures such as found in Russia and Alaska. A complete retracting system, with the nut-cracker link actuated by a cylinder having no fixed point on the airplane structure, was simultaneously developed by Bristol in England, and by Olaer in France. Here the mechanical lock is included in the hydraulic cylinder (Fig. 4), and operated by a subsidiary piston actuated by the pressure existing in the cylinder itself, thus suppressing external connections. This feature has since been adopted by the Messerschmitt fighter.

As to the simultaneous problems of fittings and flexible hose, they were very easily solved by the Ermeto fittings (Weatherhead) as long as the wall thickness of the tube is over 0.04 in., and the ordinary Alemite flexible hose, provided the inside diameters remained under $^3/_{16}$ in. For operations at very low temperatures, a carefully designed horn-shaped metallic tubing gives a foolproof solution, so long as the displacement of the units to be connected is not too great.

Other Notable Developments in Europe

It may be worth while to mention some other characteristic devices developed in Europe. To replace the mechanical push-pull control, the Exactor system, initiated in England and since licensed in the United States, has taken a leading place. Messier developed a similar system, characterized by keeping the oil under pressure in both circuits, forward and backward; the total of the pressure in both lines is kept constant due to a single spring acting on two independent compensating pistons.

Except for the Letombe Viet air compressor, which is after all

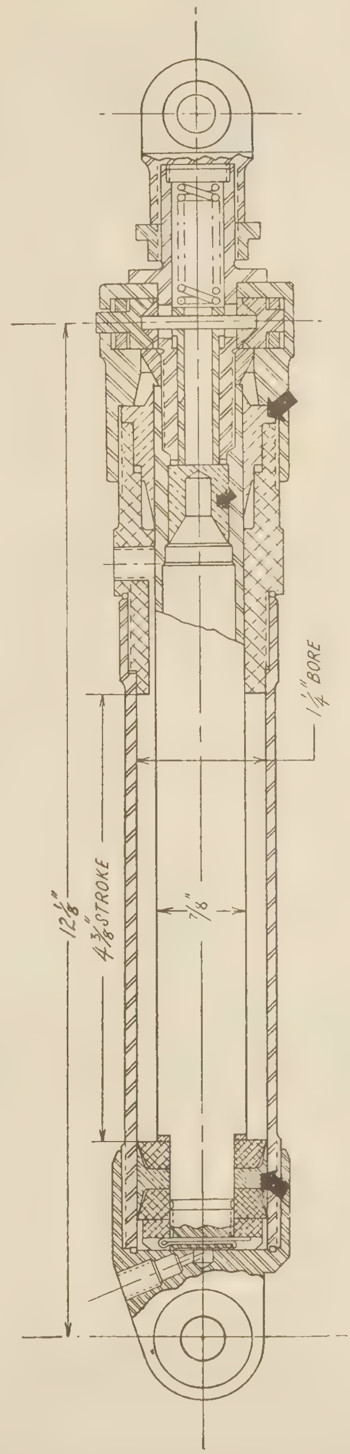

Fig. 4 Typical Landing-Gear-Actuating Cylinder for Current Type Pursuit Airplanes, Weighing About 6000 Lb
(The Olaer Mercier metallic gland, indicated by arrows, replaces the old-fashioned packing subject to aging, leakage, and rapid deterioration unless periodically renewed. The gland acts as a bearing, occupies less space, and weighs no more than the cap required to fix the old packing and the retainers. It flexes sufficiently to adapt to the cylindrical form of the piston itself, demands but slight refinement of tolerance.)

not a pump, the first application of reciprocating pumps was made in England under the name of the "Roto-Plunge" pump, manufactured by T. H. and J. Daniels Company (Fig. 5). Later on Vickers Aviation Company, Ltd., in England (which has no connection with the Vickers Company of Detroit) produced a two-cylinder pump for use in aircraft (Fig. 6).

In 1935, Bristol developed a three-stage gear pump with a normal delivery of $2^3/_4$ gpm at 1500 psi, at a speed of 2400 rpm. Between 1930 and 1933, de Boysson in France developed a very fine radial-piston pump for turret control. This pump, adopted by Messier, is shown in Fig. 7. Some 5 years ago, an automatic clutch device, controlled by hydraulic pressure,

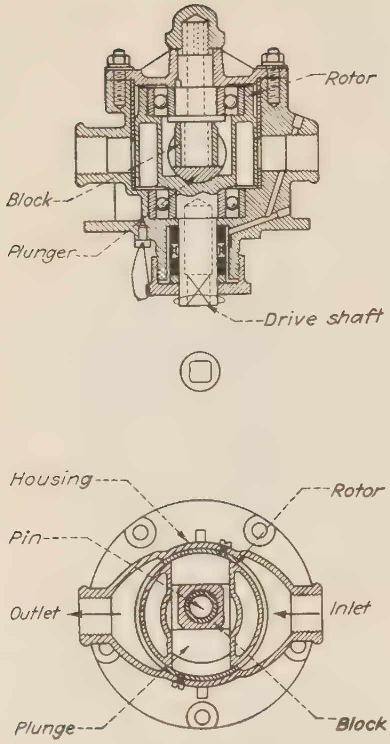

Fig. 5 Roto Plunge Reciprocating Pump, England

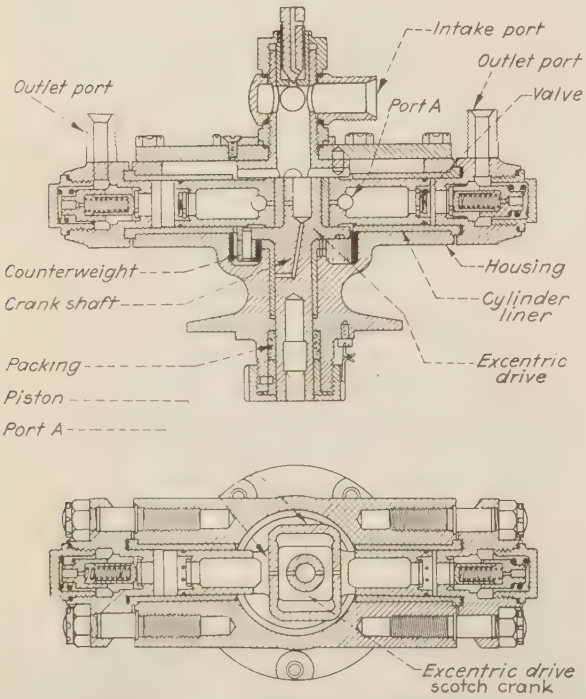

Fig. 6 Vickers, Ltd., Horizontally Opposed Two-Cylinder Reciprocating Pump

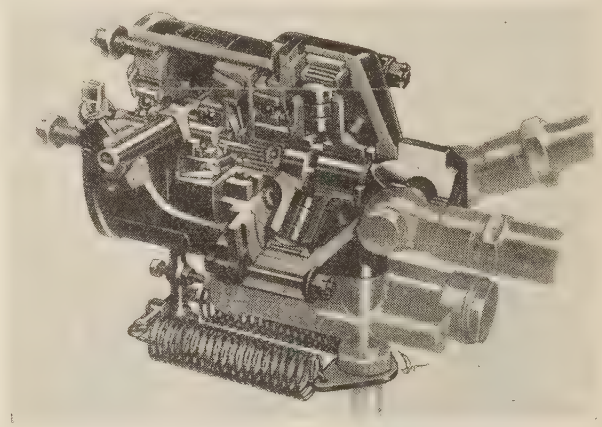

Fig. 7 De Boysson Pump, in Use in France on Messier Hydraulic Controls

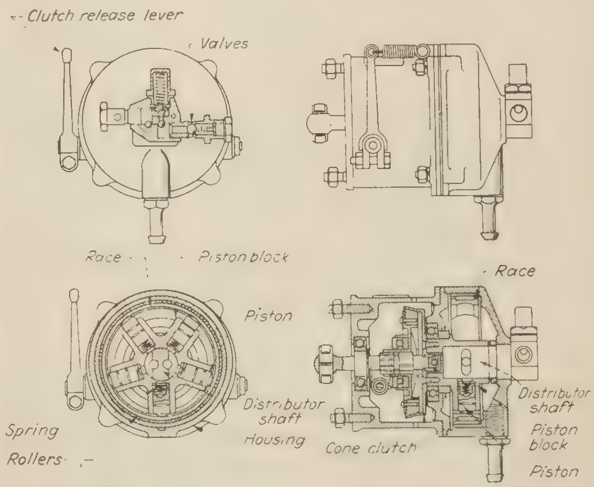

Fig. 8 De Boysson Messier Pump, France

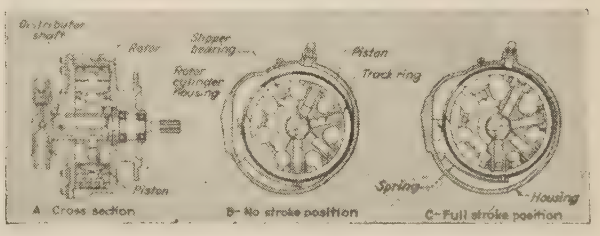

Fig. 9 Cross Section Dowty, England, Live-Line Pump

was evolved jointly by de Boysson and Messier (Fig. 8). On the same general principle, Dowty adopted a pump which a few years ago was transformed into a self-regulating pump by the addition of a spring limiting the torque (Fig. 9). This is known today as the "Live-Line" system in which the only, but very interesting, improvement over the Olaer self-regulating system consists in the elimination of the accumulator, a bigger pump taking care of the instantaneous flow requirements. This carries the risk that, in case of pump or engine failure, no energy is available.

The Olaer pump (Fig. 10), developed in the period 1935–1938, and since adopted in several European countries, is somewhat similar to the well-known Vickers wobble-plate type, described fully in a paper by Dale Herman.[3] Its characteristics are (1)

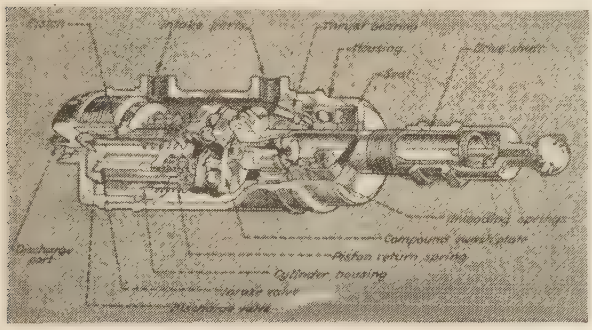

Fig. 10 Olaer Mercier Pump, 5000 Psi, Weight, 2.2 Lb
(In use in France, Holland, Italy, and Switzerland.)

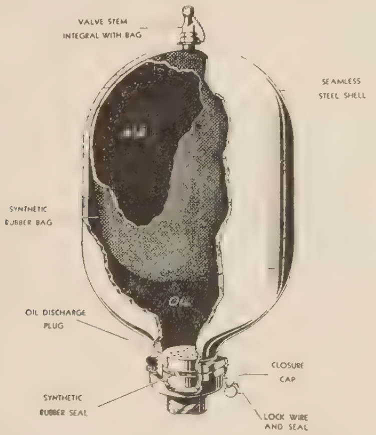

Fig. 11 Olaer Mercier Accumulator as Built in the United States

supercharging, thus avoiding cavitation at high speed; (2) self-regulation by a spring concentric to the axis; and (3) lightness of weight, 2.2 lb for a $^{1}/_{4}$-gpm, 5000-psi pump. The basic design for all these pumps, as also for many of the ultramodern hydraulic devices, can be found in the hydraulic controls used on battleships for the last 40 years. Hele-Shaw (England) and Waterbury (England) have contributed largely to their development. Happily for us aviation hydraulics engineers, the prohibitive weight of these known devices makes their use in aviation impossible. The hydraulic cylinders, utilizing rubber or leather glands, are more or less similar to the American practice.

Fig. 11 shows the Olaer-Mercier bag-type accumulator, and Fig. 4 the Olaer jack, also developed in 1936, the latter having an internal self-locking device and metallic glands as shown in Figs. 3 and 4.

The similarity holds good for European and American globe, check, regulating, pressure-reducing, safety, and selector valves.

In Europe, faith in hydraulics led to the development of hydraulic valve locks to prevent eccentric movements. Bristol and Olaer use a valve operated by a connecting rod which, by shutting off the line when the weight of the airplane telescopes the oleo strut, makes it impossible for the undercarriage to retract on the ground. A variant feature of this device prevents the outflow of oil from the jack when external force tends to displace the piston. Bristol applies his double hydraulic lock on bomb doors. This lock consists of spring-loaded poppet valves independent of the jack. The Olaer system is based on a sliding head of the jack itself, which closes the hydraulic circuit when the head tends to telescope into the cylinder due to external forces. When the internal pressure is again applied it pushes out the head thus reopening the oil circuit.

War Interrupts Work on Extensive Hydraulics Applications

There were three more interesting developments in process at the outbreak of the war in Europe. One, in collaboration with the French Navy, for watertight compartments, was later proposed for very large airplanes. If the weight increase, necessitated by the increasing distance between the hydraulic units and the increased energy to be transmitted on the "bigger and better" planes now on their way, were not provided for, the planes would soon carry more tubing than freight. A patented solution to this was found in providing, close to each hydraulic cylinder, two accumulators, one of high pressure to store the local energy, and the other of low pressure to absorb locally the outlet flow required during rapid actuation, as well as a local selector valve, controlled electrically at distance by the pilot or, in case of emergency, by hand. An enclosed hydraulic circuit, independent of the atmospheric pressure and particularly noteworthy for stratospheric flying, is thus obtained automatically by the elimination of the oil reservoir rendered unnecessary by the presence of the low-pressure accumulators. The combination of an electrical remote-control system, with hydraulic-actuating cylinders, may be compared with the nervous system of the human body in controlling the flow of blood to the muscles for actuation. This organic union might well encourage the electrical engineer to accept hydraulics "for better or for worse," or at any rate until rotary high-speed servomotors take its place.

Another development of note was a hydraulic steering device for the front wheel of the Fokker D23, a tricycle landing gear similar to the follow-up rudder control of ships, which sooner or later will be applied to airplanes.

A still further interesting point was the hydraulic control of two or more landing flaps, operating simultaneously without any mechanical connection. Instead of controlling the amount of fluid to be admitted to or exhausted from each of the operative jacks, the problem was reversed. If a disequilibrium should occur, the pilot would correct it in the normal way by means of the stick, thus automatically bringing into action a special distributing valve, which in turn corrects the amount of fluid needed for an exact symmetrical displacement of the flaps.

Of course, some additional features are required to permit the correction being made while lowering or raising the flaps. A leak in any part of the line would be taken care of automatically, because the control is not affected by the distribution of the oil, nor by the movement of the flaps themselves when not absolutely symmetrical, but by the stability of the plane itself.

[3] "The Evolution of the Hydraulic Pump as Applied to Aircraft," by Dale Herman, appearing on pages 583–588 of this issue of the Transactions.

A two-position actuating cylinder was used to permit different landing and taking-off flap positions.

ACKNOWLEDGMENT

The author wishes to express his appreciation for the courtesy and help of Commander Harry Marx, U.S.N.R., who is in charge of hydraulics at the Bureau of Aeronautics of the Navy; of Mr. Edward Greer of Greer Hydraulics Co.; and of Simmonds Aerocessories, Inc., for the use of the illustrations in this paper.

Maintenance of Aircraft Hydraulic Systems in the Field

By R. E. MIDDLETON,[1] BURBANK, CALIF.

This paper deals with certain aspects of the aircraft-accessories maintenance problem, under wartime conditions, and sets forth some suggestions as to how manufacturers of aircraft hydraulic equipment might assist in its solution. The comparative merits of hydraulic systems and electrical-mechanical systems for operating airplane accessories are discussed from the standpoint of maintenance and repair.

In contrast to peacetime when Army aircraft could be serviced under the best possible shop conditions by a thoroughly trained and experienced staff, highly complex and delicate aircraft accessories are today overhauled, repaired, and adjusted oftentimes in the open, exposed to extremes of weather and with only the most limited tools and equipment. It is true that all large well-equipped air bases have proper maintenance facilities but under combat conditions, so long as an airplane will fly, it is usually operated from outposts far removed from such shops.

Through the courtesy of the Army Air Forces, the author has had an opportunity to become quite thoroughly acquainted with the conditions under which airplanes being used in the Alaskan theater of operations are being serviced and kept in proper flying condition. This study involved observation of the maintenance and service operations at various bases, including actual combat squadrons.

Improvement Necessary in Servicing Hydraulic Equipment

This study of conditions in the field demonstrated the eagerness of the Army Air Forces' personnel to avail themselves of all constructive advice and help forthcoming from civilian technicians and representatives of equipment manufacturers. In the case of hydraulic equipment, which is doing a fine job in the field, the results are being accomplished with wholly inadequate field service provided by manufacturers of this type of equipment, and in spite of certain quite serious handicaps. In general, manufacturers have not furnished the caliber of field service essential to the most advantageous operation of the equipment and the least interference with operation of the airplane, at all times. Of course, one important reason for this is the tremendous expansion in the Army Air Forces without corresponding increase in available personnel, with a thorough knowledge of hydraulics.

It is believed that the greatest field for improvement in the operation of hydraulic systems under service conditions lies in the expansion by hydraulic-equipment manufacturers of the facilities for educating service personnel; and in furnishing the proper tools, equipment, and service manuals pertinent to the maintenance, adjustment, and repair of these accessories.

Of course, it would not be accurate to assume that similar service difficulties are not encountered in electrical-mechanical systems for operating accessories. During the tour in Alaska it was possible to observe many instances where electrical equipment of various kinds was being removed, repaired, or replaced. Actually, the percentage of such replacements appeared to be higher than the percentage of replacements of hydraulic equipment. In connection with such comparisons, it should be borne in mind that the total quantity of hydraulically operated accessories now in use on standard military airplanes exceeds considerably the total of electrically operated accessories. Therefore, even though the percentage of replacements of hydraulic equipment is low, the actual number of such replacements may still be greater than the actual number of replacements of electrically operated equipment.

More Rugged and Serviceable Designs Will Help

The following suggestions are made to manufacturers of hydraulic equipment in the interest of remedying at least partially some of the failures of hydraulic equipment to give the best possible service. By improving field conditions, one of the greatest obstacles to present and future progress in the use of hydraulics in aircraft will be removed.

It is the author's belief that hydraulic-accessories manufacturers have not in the past assumed their full responsibility for the successful and trouble-free operation of their equipment in the field. In many cases, the design of such equipment does not provide the reliable and rugged construction which is necessary for it to stand up under varying and adverse atmospheric, weather, and ground conditions which prevail in many theaters throughout the world where American-built aircraft are called upon to operate. In partial justification, it is true that sufficient time has not been available for developing and testing design features of the equipment which may later cause trouble. Obviously in many cases the airplane itself has had to be the proving ground for the equipment. Consideration must also be given to the extreme urgency under which manufacturers have been operating. They have been called upon to produce equipment within the most stringent delivery schedules. Nevertheless, it is believed to be entirely feasible for the accessories manufacturer to work out a comprehensive program which will allow sufficient time to find and correct the majority of the weak points of his equipment through careful test and improvement.

The facts demonstrate that in the few cases where such procedure has been followed, even to the point of apparent overcaution in proving equipment before it is installed in airplanes, the greatest serviceability in the field has been achieved.

Another important design consideration relates to the tools and methods required for maintenance and service. Many instances are on record of units which required special wrenches, holding fixtures, and adjusting tools, for their proper adjustment and repair. These may be quite acceptable where the units can be serviced in well-equipped base repair depots, which can maintain supplies of such tools, fixtures, and gages. Under war conditions, in practically no case is sufficient equipment available at the points where airplanes are actually being operated in service, and where it is necessary for mechanics to keep them flying.

Whether it is hydraulic, electrical, or mechanical, the ac-

[1] Chief Engineer, Hydraulics Division, Aircraft Accessories Corp. Mem. A.S.M.E.

Contributed by the Aviation and Hydraulic Divisions and presented at the Semi-Annual Meeting, Los Angeles, Calif., June 14–17, 1943, of The American Society of Mechanical Engineers.

cessory equipment which literally can be taken apart and put together again on the upturned bottom of a packing case, with the ordinary mechanic's tool kit, will in general be successful. Certainly it will have to be rugged equipment. Experience would appear to prove that hydraulic equipment is much more susceptible to design which is capable of this type of servicing than are electrical-mechanical devices, which are more delicate, more needful of special tools, and more difficult to repair in the field.

Another major point in a program of service improvement is the assumption on the part of the accessories manufacturer of a greater share of the responsibility for actual servicing of the equipment in the field. The first step is for each manufacturer, within his ability to do so, to assure that full information and instructions are made available to all squadrons flying airplanes in which his equipment is installed, and also to make a greater effort to furnish the necessary tools and spares at points where they are needed. It is believed that this can best be accomplished through the efforts of qualified service representatives of the accessories manufacturers who, under the auspices of the Air Service Command, can work with the Army and Navy personnel. They can serve as advisers, guides, helpers, and liaison between the service organizations, which must use and maintain the equipment, and the factories which, because they are so far removed from the battle fronts, usually tend to lose sight of their great responsibility and the tremendous job that must be done in keeping the equipment working and the airplanes flying.

As has been amply demonstrated by the aircraft manufacturers, a sufficient number of qualified service representatives can be invaluable in aiding the personnel of the armed services to become thoroughly familiar with the highly specialized, intricate and, at best, rather delicate accessory equipment They can go far toward attaining the degree of co-operation between the accessories manufacturers and the actual users of the equipment, which will not only insure long life but also reliable operation.

Centrifugal Casting of Steel

By S. D. MOXLEY,[1] BIRMINGHAM, ALA.

This relatively new method of casting is being applied to great advantage in the economic production of high-quality engineering parts. The three methods generally used are true centrifugal casting, semicentrifugal casting, and centrifuging. The author gives a comprehensive description of each method, the work for which it is best adapted, the machines used, details of molds, and physical properties of the resulting products.

THE casting of metals centrifugally is basically "pressure casting." In static molding, feeding is done by atmospheric pressure and gravity, and the pressure under which the metal is cast is dependent upon the height of the gates and risers. In centrifugal casting, the metal is forced against the mold wall and into the cavities under much higher pressures resulting from centrifugal force imparted by the revolving mold.

Generally, the methods of centrifugal casting are (1) true centrifugal casting, (2) semicentrifugal casting, (3) centrifuging.

The "true centrifugal" method consists of spinning the casting about its own axis and using centrifugal force to hold the metal on the wall of the mold, thus forming the inside without the use of a center core. When a mold is spun about a horizontal axis, the interior cavity formed by the molten metal is a true cylinder, regardless of the shape of the outside of the casting. Its inside diameter is determined by the volume of metal poured.

Fig. 1 shows a cylindrical flask lined with sand and provided with suitable end fixtures. If the mold is spun at sufficient speed and metal is poured into it, a cylindrical tube, as indicated, is cast.

If more than one inside diameter is required in a casting, the larger inside diameters may be formed by cores extended from the ends of the mold, leaving the smaller diameter to be formed by centrifugal force. Fig. 2 shows a conventional mold for making cast-iron pressure pipe. Note that the inside diameter of the socket is larger than the inside diameter of the pipe and is made with a core.

This method has been used in the mass production of cast-iron pipe for more than 20 years. During that period both sand-lined mold and metal-mold processes have been in successful commercial use, and their economic soundness has long since been proved. Higher physical properties have been obtained, and this has resulted in a reduction in the required thickness and weight of the pipe for a given service. Alloy-steel gun barrels are centrifugally cast in horizontal metal molds by a process developed over a long period of years at Watertown Arsenal.

True horizontal centrifugal casting is well adapted to the manufacture of short and long tubes in various sizes where the inside cavities are cylindrical. Steel tubes, ranging from several inches in diameter up to 50 in. diam, can now be made by the author's company in lengths of 16 ft. Typical products of this method are pipe, various kinds of tubing, radial-engine cylinder barrels, hollow ship shafts, weld-neck castings, bearings, sleeves, etc.

Vertical True Centrifugal Casting

True centrifugal casting is also done by spinning the mold about a vertical or an inclined axis. When this is done the resulting interior cavity is a paraboloid. The shape of this paraboloid is dependent upon (1) the speed of rotation of the mold, (2) the diameter of the closing fixture at the top of the mold, and (3) the angle of inclination of the axis of rotation. Assuming a fixed diameter at the top of the inside cavity, and a fixed angle of inclination, the depth of the paraboloid is increased by an in-

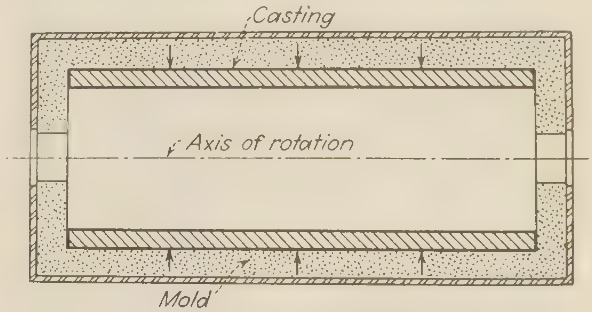

FIG. 1 CYLINDRICAL FLASK LINED WITH SAND

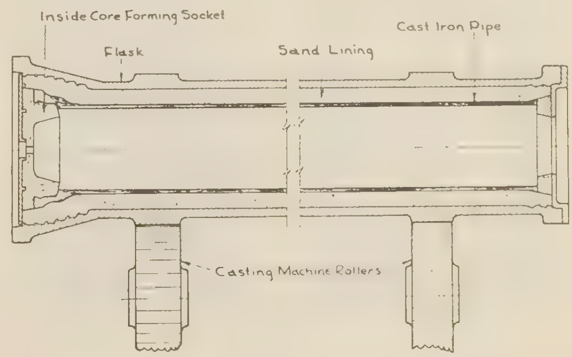

FIG. 2 TRUE CENTRIFUGAL MOLD FOR MAKING CAST-IRON PIPE

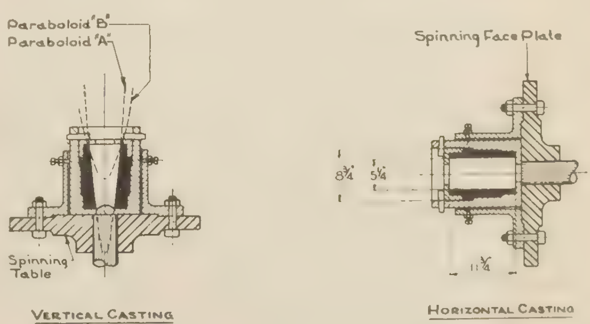

FIG. 3 TRUE CENTRIFUGAL METHOD OF CASTING RADIAL-ENGINE CYLINDER BARRELS

[1] Chief Engineer, American Cast Iron Pipe Company. Mem. A.S.M.E.

Contributed by the Metals Engineering Division and presented at the Spring Meeting, Birmingham, Ala., April 3–5, 1944, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

crease of speed. Likewise, the paraboloid will be shortened with slower speed.

Fig. 3 shows radial-engine cylinder barrels being cast by the horizontal and the vertical methods. At the right the mold is spun about a horizontal axis, and the inside cavity is cylindrical. At the left the mold is spun vertically and the inside cavity is parabolic in shape. At a certain speed of rotation the inside cavity will take the shape of paraboloid A as shown, while at some lower speed the inside cavity will be formed on the line of paraboloid B. In either case, the top diameter of the inside cavity will be larger than the bottom diameter. These castings ($8^3/_4$ in. OD $\times$ $11^3/_4$ in. long) are now being made horizontally at speeds of 975 rpm and lower, and vertically at speeds approaching 1200 rpm. The mathematics for calculating the shape of the inside paraboloid is relatively simple and is in the record.[2] Fig. 4 shows a comparison of an actual paraboloid as cast with the shape of cavity as calculated.

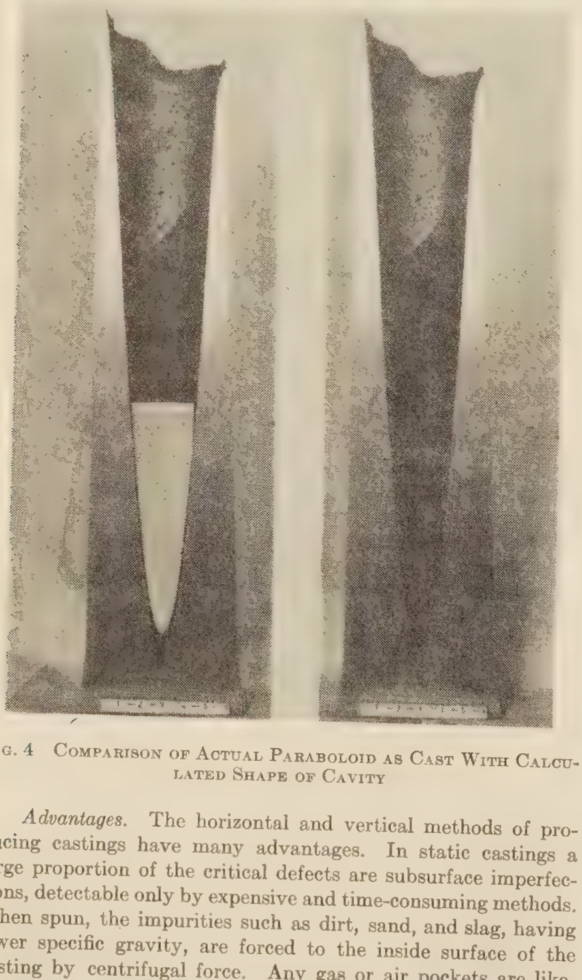

Fig. 4 Comparison of Actual Paraboloid as Cast With Calculated Shape of Cavity

Advantages. The horizontal and vertical methods of producing castings have many advantages. In static castings a large proportion of the critical defects are subsurface imperfections, detectable only by expensive and time-consuming methods. When spun, the impurities such as dirt, sand, and slag, having lower specific gravity, are forced to the inside surface of the casting by centrifugal force. Any gas or air pockets are likewise eliminated. The elimination of these foreign inclusions results in a sounder casting, and the product is more uniform. A centrifugal casting is more dense than a static casting. It is doubtful that the method of casting steel would change the specific gravity of the metal, but being free of these foreign inclusions or voids would naturally make a casting more dense, or rather, heavier per unit volume.

[2] R. F. Wood, U. S. Patent No. 1,533,780.

Directional solidification is very important in any method of molding. In true centrifugal casting, this occurs naturally, since the metal is cooled from the outside toward the center. Inspection of the castings is greatly simplified, since such defects as are found are in nearly all cases on the inside or outside surface of the castings, thus making them more readily detectable.

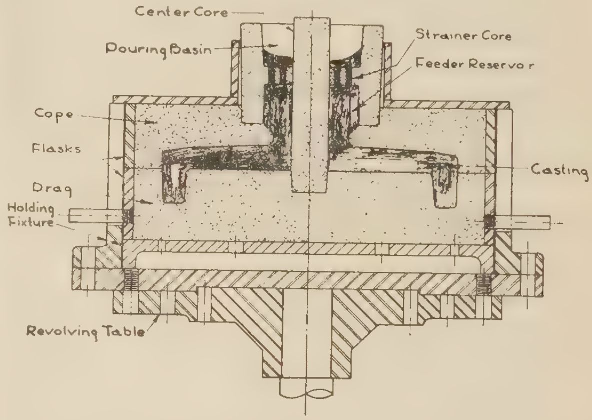

Fig. 5 Semicentrifugal Casting of Flywheel With Center Cavity

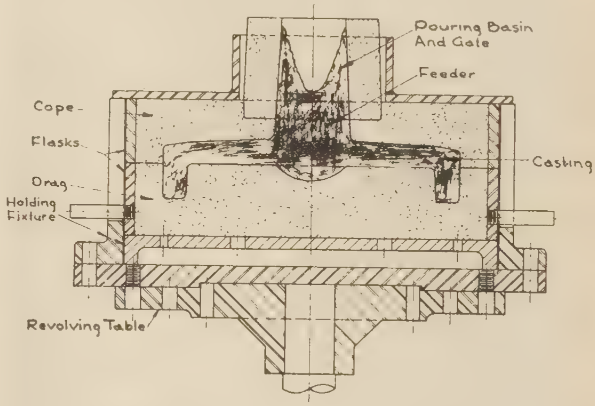

Fig. 6 Semicentrifugal Casting of Flywheel With Solid Center

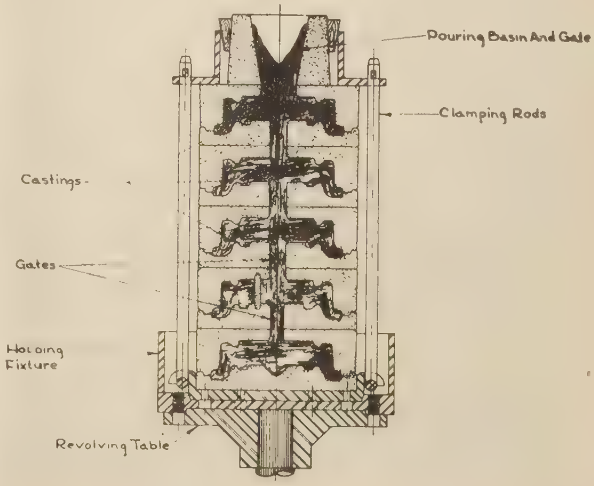

Fig. 7 Semicentrifugal Stack Molding of Track Wheels

The elimination of inside cores greatly reduces the costs in the foundry as well as in the cleaning shed.

The elimination of gates and risers greatly increases the yield and in some cases 100 per cent yield is obtained. Cases in which 100 per cent yield is obtained include cast-steel hollow ship shafts, various tubing, cast-iron pipe, etc., where inside machining is not required. In castings such as radial-engine cylinder barrels where the inside machined surface is highly important, the wall is poured thicker than required. This extra thickness on the inside serves very much as an inside gate or feeder and is later removed by machining. Thus in such cases the actual yield is lower. The centrifugal method is highly adaptable for mass production of identical castings.

Semicentrifugal Method

Semicentrifugal casting consists of spinning a casting about its own axis—usually about the vertical axis. If the casting has a center cavity, this cavity is often formed with a center core, and the casting is fed by a center gate passing down around this core. Fig. 5, demonstrating this method, shows a cross section through a sand mold mounted on the revolving table of a vertical casting machine. This is a flywheel casting and is typical of the semicentrifugal method.

Fig. 8 Semicentrifugal Casting; Stack Molding of Tractor Wheels
(Mold made up ready for placing on casting machine.)

Fig. 9 Semicentrifugal Casting, Illustrating Stack Molding of Tractor Wheels

In many cases, the center is cast solid and the center cavity, if any, is machined out, or removed by an oxyacetylene torch. This is shown in Fig. 6 where the same flywheel is cast with a solid center.

Metal molds and a combination of metal and sand molds are also used. Stack molding, where a number of castings are made in one pour, is well adapted to this method. This is illustrated in Fig. 7. Fig. 8 shows a built-up stack mold ready to place on the casting machine. Fig. 9 serves to illustrate the possibilities of stack molding. On the right is a wheel casting made one in a mold, while at the extreme left it will be noted that eight castings are made on the stick. The only limitation on the number of layers used in stack molding is the convenience and economics of the handling operations. Usually a greater number of castings made on a single stick increases the foundry yield, and this factor must be considered against the problems of handling, pouring, shakeout, and other foundry problems.

Very often it is found advisable to change somewhat the design of the casting to favor gating, feeding, and directional solidification in order to obtain more uniformly sound castings. Ordinarily, this can be done without altering the usefulness of the casting, if the engineer and foundryman will cooperate.

In good practice, the gating, feeding, and provision for directional solidification are different in centrifugal molding. The gates and feeders must be designed to suit this type of molding. The foundry technique also must be changed to suit the method. Most successful foundries have found it necessary to engineer carefully each job that is to be spun, and many jobs are found better suited for static molding.

The speed of rotation in semicentrifugal casting is lower than that used in true centrifugal casting, and hence the centrifugal pressure is less. However, this rotation keeps the liquid metal in motion. This action aids in flowing the metal into all of the cavities of the mold, and in more completely feeding the casting. It further insures against blows and voids caused by contraction during solidification. This produces sounder and more uniform castings.

This method is particularly adaptable for casting wheels, gear blanks, and other circular shapes.

Centrifuging

In the "centrifuging method" the molds forming the useful castings are positioned near the periphery of the revolving table. The metal is poured into a gate located on the axis of rotation and is fed into the molds by radial sprues. Centrifugal force is used merely to provide liquid pressure. Usually, two or more castings are made in each mold.

One of the earliest applications of this method is the old-time inlay-casting machine used in dentistry, as shown in Fig. 10. A plaster-of-Paris mold was attached to one end of an arm. The other end of the arm was pivoted to and supported by a pedestal; the arm being free to revolve around the pivot. Molten gold was poured into the mold and the mold was rapidly spun around the pivot by hand. The centrifugal force of spinning forced the molten gold into the intricate cavities of the mold resulting in a very sound casting accurately shaped.

A typical example of centrifuging is shown in Fig. 11, where five bogie-wheel hub castings are made in one mold. The center gate with the five radial sprues to feed the castings is shown.

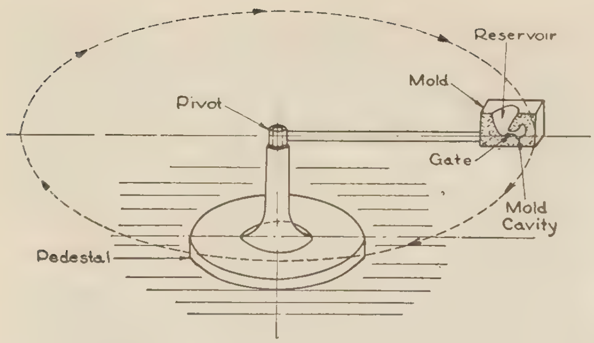

Fig. 10 Dentists' Inlay-Casting Machine, Illustrating Centrifuge Casting

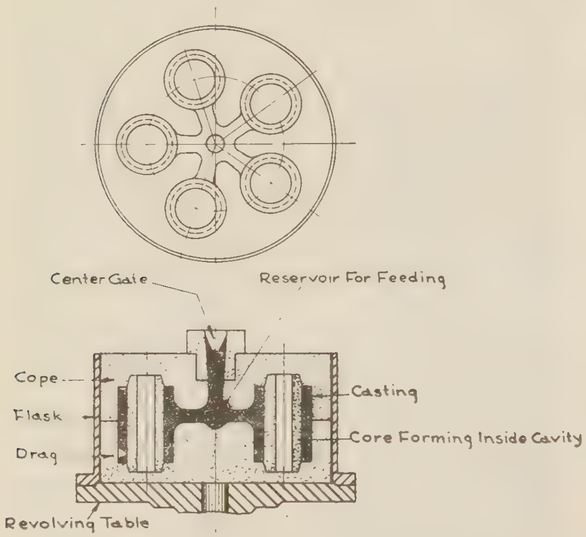

Fig. 11 Centrifuge Casting of Bogie-Wheel Hubs

In this case a core is used to form the inside cavity of each casting. This method is also well adapted to stack molding, and many odd-shaped castings are made by this process. This is illustrated in Fig. 12 which shows bracket castings made by this method in one layer, and the same castings made by stack molding in eight layers.

The advantages of centrifuged casting are substantially the same as obtained in semicentrifugal casting. Yields above 85 per cent have been reported on jobs where formerly in static casting they were less than 50 per cent. On single-mold casting by the semicentrifugal method, yields have been increased from approximately 30 per cent to more than 70 per cent. The casting machines can be of simple construction, and therefore not too expensive, either in first cost or in maintenance. Floor-type units may be used which can be moved from one location in the foundry to another at little cost or inconvenience.

Machine Construction

Many different designs of centrifugal-casting machines are used. Those used in "true centrifugal" castings are of two types, i.e. (1) indirect-driven, and (2) direct-driven. In indirect-driven machines, the mold usually rests on four rollers. Two or more of these rollers are connected to an electric motor or other driving unit. Sometimes top rollers are used. A suitable pouring basin with spout, or orifice, is provided on the pouring end of the machine. This orifice, or trough, projects slightly inside the pouring end of the revolving mold. The principle of design is the same with relatively long casting machines as with short ones. Fig. 13 illustrates the usual design of this type of machine.

In making short-length castings, the mold is sometimes bolted to a face plate which is mounted on a shaft, and thus the mold is directly driven. This type design is shown in Fig. 14. There are many variations, but these two general types serve the purpose here. With both types of machines sand and metal molds are used.

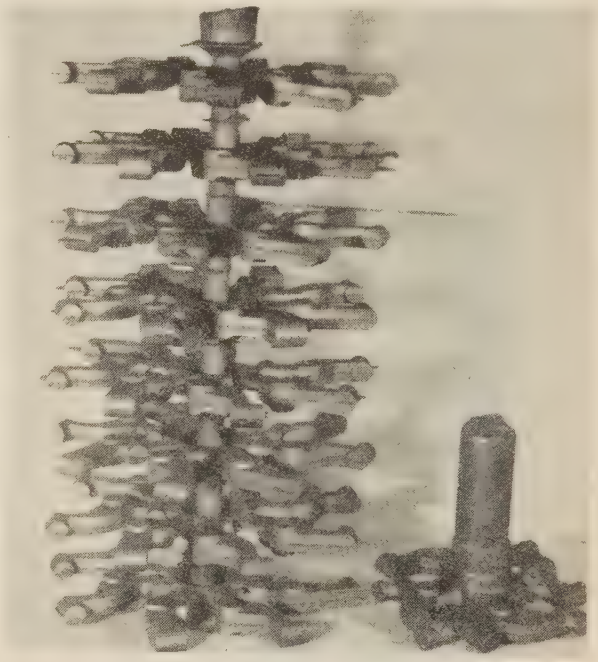

Fig. 12 Bracket Castings Made by Centrifuging in One Layer (Right); and by Stack Molding in Eight Layers

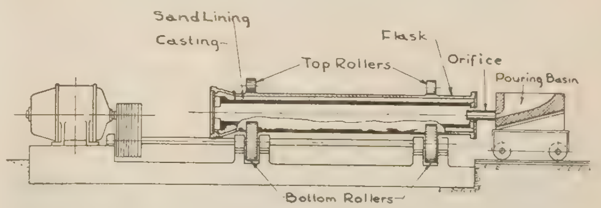

Fig. 13 True Centrifugal Machine for Long Castings

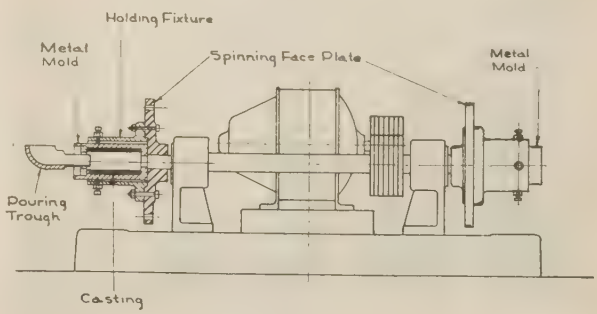

Fig. 14 True Centrifugal Casting Machine With Mold Direct-Driven

The machine used in the semicentrifugal method is identical with the one used in the centrifuge method. It consists usually of a vertical spindle on which a table is mounted, the flask being centered and securely clamped to the table, as shown in Fig. 15. Sometimes floor-mounted units with bevel-geared drives are used. This type of machine is illustrated in Fig. 16.

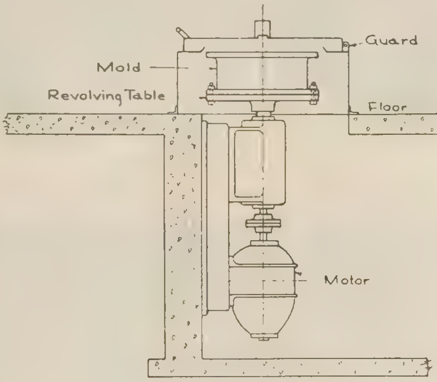

Fig. 15 Pig-Type Centrifugal Casting Machine

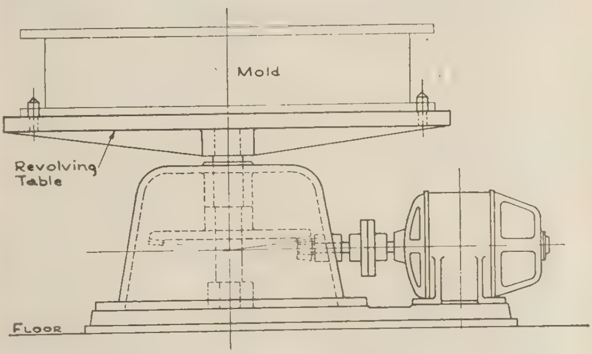

Fig. 16 Floor-Type Centrifugal Casting Machine

Drives. Many types of motors have been used to drive centrifugal casting machines. Usually a speed range of 3 or 4 to 1 will meet all the requirements, and any driving unit with this range will be adequate.

Types of Molds Used

Both metal and sand molds are being used extensively in producing castings by this method. Carbon or graphite molds have also been used for casting nonferrous metals. Each kind of mold has its place in centrifugal casting, and the type of casting to be produced usually determines the kind of mold best suited.

Permanent molds are best adapted (1) where large quantities of identical castings are to be made, (2) where the outside contour of the casting is such that it can be readily withdrawn from the mold after casting, and (3) where a fast cooling rate from the outside inward with attendant fine grain is desirable. They are also better suited for castings which have relatively uniform wall thicknesses, or without abrupt changes in wall thickness.

Permanent molds are usually made of cast iron or steel. When cast-iron molds are used, it has been found desirable to have a cross-sectional area of the mold 4 to 5 times that of the casting produced. The life of a permanent mold is dependent upon many factors and foundry techniques which have not yet been fully explored. Usually metal molds are kept at temperatures between 400 F and 800 F, during the casting operation. It is the usual practice to coat the mold with some form of facing between casts to protect it.

Sand molds are generally preferred (1) in the casting of long tubes where the metal must flow relatively long distances over the mold surface, (2) where the shape of the casting is such that the mold must be broken down to remove it, (3) where it is essential to be able to vary the dimensions and shapes of similar castings, (4) where relatively few identical castings are to be made, and (5) where a relatively uniform cooling rate through a thick section is desirable. Sand molds more commonly used are dry-sand, skin-dried, and oil-sand, and they are usually faced to prevent cutting. The cooling time of castings made in sand molds is much longer than in metal molds, thus the casting is free of chilled surfaces and is likely to have fewer casting strains. The sand used for centrifugal molding must be much stronger than in static molding to resist the added pressure and vibration caused by spinning. Usually it is found advisable to have a sand with higher permeability than that used in static molding.

Quite often a combination of metal molds and sand molds is used. Where it is desired to core holes of various shapes in castings, cores are used with the metal molds for this purpose. Often it is desirable to relieve the chill in certain portions of the casting, and this is done by using dry-sand cores. In some cases, there is a serious cutting action on the mold where the metal enters during the casting operation. A dry-sand splash core inserted in the metal mold to prevent this cutting action will materially lengthen the life of the metal mold.

Many Speeds Used in Centrifugal Casting

In the manufacture of centrifugal castings a wide variety of mold speeds is used. The terms usually referred to are "peripheral velocity," and number of "times gravity" centrifugal force, or pressure, against the mold wall. The casting speed is linked with a number of other factors, such as pouring time, size and shape of casting, wall thickness, method of centrifugal casting being used, and others.

A general rule in the manufacture of relatively thin-wall tubes by the horizontal true centrifugal method in sand-lined molds is a speed of rotation which will produce a centrifugal force of 75 times gravity. Although this rule is generally accepted, it might be said that a favorable speed would be the lowest which would prevent "raining" during the pouring operation and yet provide enough force to squeeze out the foreign inclusions in the metal.

In practice the rotational speed of metal molds usually will be found lower than with sand molds, approximately 60 times gravity. This is because the chilling effect of the mold causes the molten metal to "pick up" more readily than in the refractory sand mold.

In the manufacture of cast-iron pipe in sand molds, the general rule is 75 times gravity. The general rule in the manufacture of cast-iron pipe in water-cooled steel molds is from 900 to 1250 fpm peripheral velocity, depending upon the size and thickness of the pipe to be cast.

An examination of the rotational speeds used by the various successful steel foundries leads to the conclusion that speed is not critical. Certainly it can be said that the ideal rotational speed cannot be determined at present, in the light of the many variables involved in true centrifugal casting of metals.

Likewise, there is a wide variation of rotational speeds used in semicentrifugal casting and in centrifuging. By both of these methods castings are being successfully made with peripheral speeds on the outermost part of the casting of from 100 to 1000 fpm. Here again, factors such as gating, pouring time, size and shape of casting, and type of mold used are to be considered. In

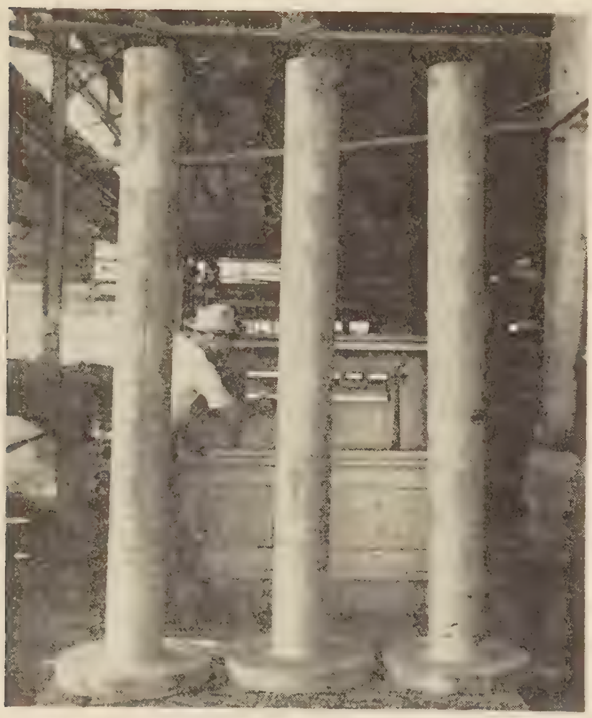

Fig. 17 Rudder Stock, Fabricated From Centrifugally Cast Tubing, Welded to Statically Cast Flange Section

Fig. 18 Hawse Pipe for Ships
(Main section centrifugally cast as a tube. Welded-on deck and shell bolsters were statically cast.)

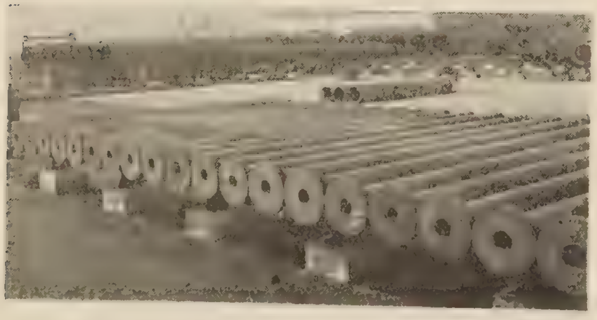

Fig. 19 Shafting for Ships
(Centrifugally cast in 16-ft lengths.)

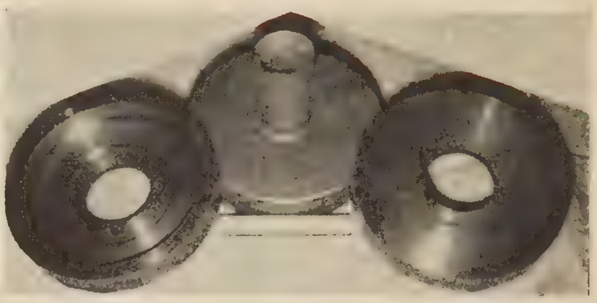

Fig. 20 Engine Flywheels
(Centrifugally cast in vertical-axis machines in sand molds.)

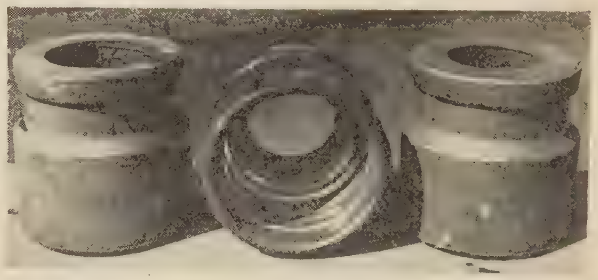

Fig. 21 Idler Wheel Hubs
(Centrifugally cast in horizontal-axis machines, using sand-lined molds and cores.)

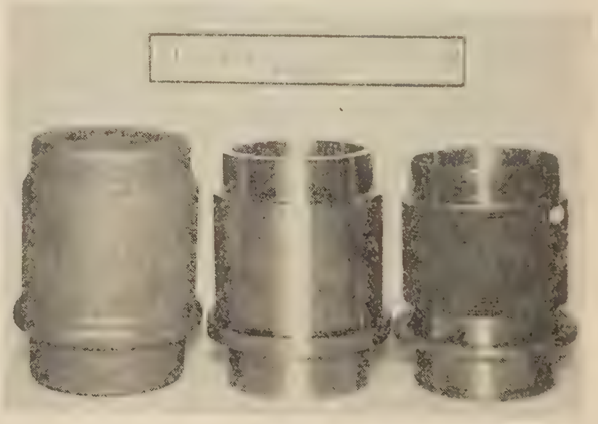

Fig. 22 Cylinder Barrels for Radial Engines
(*Left:* As cast. *Center:* Rough-bored and turned. *Right:* As finished by the engine builder.)

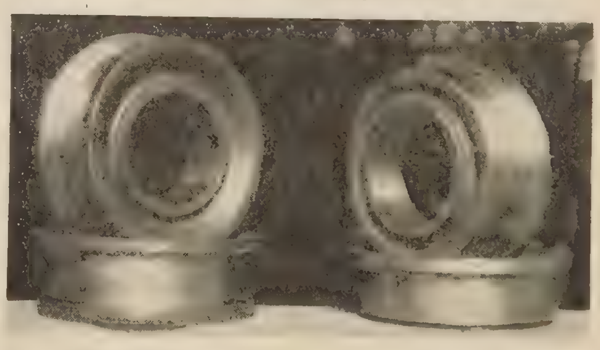

Fig. 23 Track-Wheel Rims for Crawler-Type Tractors
(Single-flange and double-flange track roller rims, cast centrifugally in vertical-axis machines. High-carbon steel for special quench heat-treatment to provide shrink fit on hubs.)

some cases the mold is filled with molten metal before revolving. In other cases the mold is poured at slow speed and then brought up to full casting speed.

Typical examples of centrifugal castings made by the author's company are shown in Figs. 17 to 23.

Physical Properties

The present emergency has given sharp impetus to the development and the production of centrifugal steel castings, not only as an improvement over static casting methods, but to produce parts formerly thought possible only by forging. Most of the physical properties of these castings have been found to compare favorably with those of forgings and, in many cases, to be better, depending upon how the casting is used.

The physical properties of centrifugal castings are practically the same as sound static castings. However, the ability to consistently produce sound homogeneous castings has made it possible to cast engineering parts that were not thought possible a few years ago. Once the foundry routine is adjusted, greater uniformity of product is usually obtained.

Compared with forgings it may be said that some centrifugal castings can be produced nearer to finished dimensions with consequent saving in machining cost. To illustrate, Fig. 24 shows

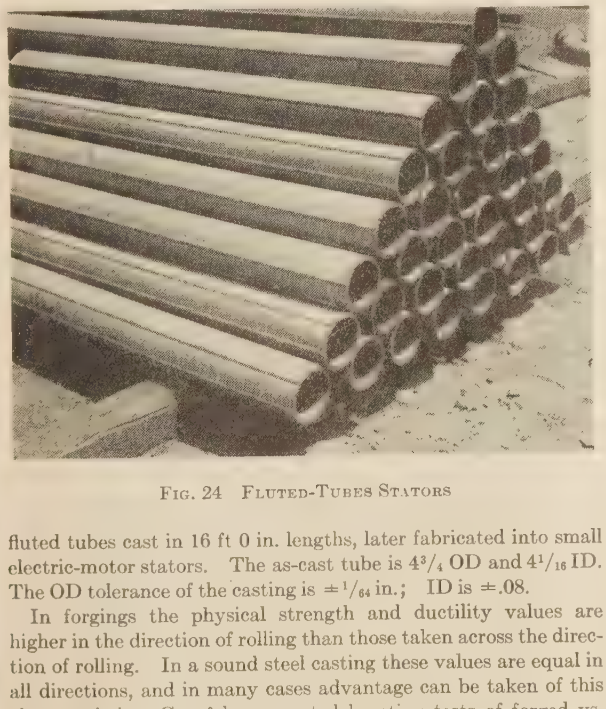

Fig. 24 Fluted-Tubes Stators

fluted tubes cast in 16 ft 0 in. lengths, later fabricated into small electric-motor stators. The as-cast tube is $4^3/_4$ OD and $4^1/_{16}$ ID. The OD tolerance of the casting is $\pm ^1/_{64}$ in.; ID is $\pm .08$.

In forgings the physical strength and ductility values are higher in the direction of rolling than those taken across the direction of rolling. In a sound steel casting these values are equal in all directions, and in many cases advantage can be taken of this characteristic. Cone[3] has reported bursting tests of forged vs. centrifugally cast airplane cylinder barrels. The forgings split longitudinally at lower pressures than the bursting pressures of cast barrels. The cast barrels also showed greater "bulge" deformation. McCarroll[4] has also reported centrifugally cast gears to have better over-all properties than forged gears.

It should be pointed out that the chief advantage of this process is its ability to produce *high-quality* castings rather than low-cost castings. Adequate equipment and competent personnel for making chemical and physical tests on the product are essential. Rigid routine control on metallurgy, melting, sand conditioning, and the other factors of the process must be maintained to insure high-quality castings.

Conclusions

No one experienced in the art would venture to say that these methods of molding will revolutionize the steel-foundry industry or replace the great bulk of work done by forging. The size and shape of the casting place definite limitations on the process. However, the centrifugal method of casting is ideally suited for producing a vast variety of work. It is a relatively new method that is rapidly finding its place in the economic production of high-quality engineering parts.

Discussion

J. B. Caine.[5] This paper describes a noteworthy contribution to this increasingly important method of forming metals. To the writer, at least, the economic importance of this new process should not be overlooked. It is incontestable from the theoretical standpoint that the most efficient method of forming a shape slightly more complicated than a round or flat is by casting. The reason that steel castings have not been able to compete in price with other methods of forming relatively simple shapes in quantity has been the necessity of making an individual mold for each casting and that steel shrinks during solidification. This solidification shrinkage necessitates the use of such large risers to supply liquid steel to the solidifying casting that the steel foundry must pour twice as much metal in the average mold as that required for the casting.

If the design of the part permits the use of permanent molds and the true centrifugal process, an individual mold for each casting is not required, and, as the author states, yields approach 100 per cent. Casting is then the most efficient method known of forming these parts. If the part is more complicated, the semicentrifugal and centrifuging processes, although requiring an individual mold for each casting or group of castings, increase the all-important yield and therefore improve the competitive position of castings.

There is at least one other advantage that the author has not stressed, namely, the use of centrifugal force to impel molten metal into thinner sections than can be cast statically. With increasing emphasis on strength-weight ratios, it will be necessary to cast thinner and thinner sections to take full advantage of high-strength cast steels. A corollary of this is, of course, the use of centrifugal force to cast fine detail and within tolerances measured in thousandths of an inch.

Nathan Janco.[6] This paper is an excellent summation of the centrifugal process of casting steel, and its progress up to the present time. The various methods of centrifugal casting are clearly outlined, as well as some of the details of the actual production methods. The paper certainly should be of considerable interest to practical foundrymen, as well as to mechanical engineers concerned with the design of steel parts.

In the production of semicentrifugal castings, or castings produced by the centrifuging method, it is essential that conditions for directional solidification be set up, in order to obtain perfect castings. In a great many cases this can be accomplished only by pouring the metal into the mold while the mold is revolving. However, if the metal is poured into the mold at the usual speeds

[3] "Centrifugally Cast Cylinder Barrels for Airplane Engines," by E. F. Cone, *Metals and Alloys*, vol. 16, no. 6, December, 1942, pp. 1062–1066.

[4] "Progress in Automotive Steel Castings," by R. H. McCarroll, *The Foundry*, vol. 66, no. 10, October, 1938, pp. 30, 31, 72, 74, and 77.

[5] Metallurgist, Sawbrook Steel Castings Company, Lockland, Ohio.

[6] Chief Engineer, The Centrifugal Casting Machine Company, Tulsa, Okla. Mem. A.S.M.E.

used for spinning, the erosive action of the metal upon the sand-mold surface will be excessive and dirty castings will result. This is extremely important as it is to no avail if metallurgically perfect castings are produced and then have to be scrapped because they are dirty. During the past few years, this difficulty has been overcome by pouring the metal into the mold while the mold is rotating at speeds between 10 and 25 rpm. After the mold is full, or partially full, it is accelerated to the proper spinning speed. This has been found to be very successful in the elimination of dirty castings. A greater speed range than 3 or 4 to 1 is necessary in a centrifugal casting machine in order to accomplish the desired result.

The writer's experience concerning rotative speeds agrees with that of the author. A general average of the speeds used for the various types of centrifugal casting work, and which covers a very large percentage of all types of castings is as follows:

True Centrifugal Castings:

1 Sand-lined molds; speed, 60 to 75 times gravity.

2 Metal molds; speed, 40 to 60 times gravity.

3 Sand or metal molds, vertically spun; (generally castings whose length does not exceed 2 times the casting inside diameter); speed, 75 to 125 times gravity.

Semicentrifugal or Centrifuging Methods: Speed 600 to 1000 fpm; average, 800 fpm.

Centrifugal castings produced by the precision casting method, or "lost-wax" process, are castings which generally have very thin metal sections and usually are spun at rotative speeds below 600 fpm. In this case no advantage is being taken of the pressure of metal resulting from centrifugal force. The centrifugal casting machines are used only to provide a method of keeping the metal in motion, and thus permit the metal to run into thin sections and to obtain sharp definition.

The density and other physical properties of a perfect centrifugal casting do not differ from those obtained from a statically cast test bar which is a perfect casting. The principal advantage of centrifugal casting is that it is practical to produce castings in quantity which have physical properties approaching the best obtained in a test bar and yet achieve a high metal yield. To obtain such results by static casting methods is likely to be difficult and expensive.

E. C. Jeter.[7] There is a definite opportunity for the expansion of the already wide field of centrifugal castings. Sometimes, however, certain groups are prone to become overenthusiastic and to apply this method to a wide variety of parts, many of which would better have remained as forgings or as static castings.

Some of these are of such a design that centrifugal casting methods could not guarantee freedom from defects, and thus centrifugal castings would not be good enough from a quality standpoint. Physical-property requirements might be above those which can be met by centrifugal castings, although there probably would not be many such cases.

Other parts might be of such design that static castings would be of high enough quality and more economical. The simplicity of design might indicate that a forging would be more economical than either a centrifugal or a static casting. Unless castings are of such design as to be centrifugal "naturals," such as flanged cylinder barrels and symmetrical castings of that type, a static casting might be good enough or be more economical, unless for some reasons of design a very great saving in gates and risers is obtained.

In the final analysis the centrifugal method of casting should be applied intelligently to the parts for which it is best suited. Promiscuously making a wide variety of castings, many of which are not adapted to centrifugal casting methods, will do more harm to this field than any other one thing.

Anton Johnson.[8] The author has described comprehensively the various methods of centrifugal casting practice and the applications best suited to each. Details are also given in the paper of types of centrifugal casting machines, and the jobs for which each is adapted.

Confirming the author's concluding remarks, the writer wishes to state that centrifugal casting methods will not revolutionize the steel-foundry practice of static castings, owing principally to the fact that large-tonnage castings can now be produced statically, with resultant high yields. However, the centrifugal casting method can be applied a great deal more widely than is commonly believed possible, even in a jobbing shop. The writer's company operates a medium-sized electric steel jobbing foundry, which produces a large variety of castings with an average weight of 20 lb per casting. As high as 25 per cent of the monthly output has been produced by the semicentrifugal and the centrifuging methods?

Will the author please advise what is the dry permeability of the sand in centrifugal practice, and why a higher permeability is used than in static practice? Also, what method is used in pouring stack molds, i.e., are they stationary when poured, and then spun? If spun, what has been the author's experience, if any, with misruns or cutting of molds?

[7] Ford Motor Company, Dearborn, Mich.

[8] Works Manager, Oklahoma Steel Castings Company, Tulsa, Okla.

Some Characteristics of Rotary Pumps in Aviation Service

By R. J. S. Pigott,[1] Pittsburgh, Pa.

Only within recent years have rotary pumps been accorded a semblance of the technical consideration which has been expended upon the piston or plunger reciprocating type, and the centrifugal and turbine types, between which two groups occurs its sphere of greatest usefulness. The rotary can handle all pumpable viscosities at any pressure up to 1000 psi. It has other qualities such as the ability to handle liquids with considerable gas or vapor present. The present paper is devoted to a discussion of characteristics of gear- and vane-type rotary pumps which make them widely adaptable to lubrication-oil, fuel-oil supply, and other aircraft uses in which they predominate.

The group of positive-displacement pumps generally known as rotary pumps lies between the piston or plunger reciprocating pumps, which are also positive displacement, and the centrifugal and turbine pumps, which are not. Rotary pumps have never received the amount of technical attention accorded the two other large groups, chiefly because, like the bat, they occupied a position between the "birds and the beasts." However, this stepchild of the pump family has come in for much more attention during the last 10 years than formerly, since it has been finally discovered that, given the same degree of design skill, the rotary pump is quite as good as, and in many cases better, than the other two general types, when applied in its proper sphere. The following generalizations can be made which apply to the whole field of applications:

Centrifugal pumps are best on low-viscosity liquids, such as water or light oil, and relatively large quantities at low or moderate pressure. The efficiency falls off badly at 2500 sec S.U. viscosity, and centrifugals are of no use whatever above 5000 sec S.U. The efficiency also falls off rapidly with reduction of size below 100 gpm, unless particularly fine design and smooth finish of impellers and passages are employed. The reciprocating pump is good on either high- or low-viscosity liquids, for all pressures, especially for high pressure, and for any size above 1 gpm. The centrifugal is always cheapest and smallest, in the field where it is best fitted. The reciprocating pump is always largest and most expensive, but has the high-pressure field (1000 psi up, at moderate to large size, all viscosities) almost to itself. There lies in between a field which the rotary may often fill better than either; it can handle all pumpable viscosities in any size, at any pressure up to 1000 psi. Two particular designs are satisfactory up to 3000 psi in small sizes, the radial-plunger and the axial-plunger designs. Durability is, in general, less than that of reciprocating pumps.

The rotary can also handle liquids with considerable gas or vapor present; the reciprocating pump is apt to develop difficulties under these circumstances, and the centrifugal gives up completely.

The rotary has now taken a prominent position in machine-tool operations, including presses for all sorts of oil-pumping duty, and may handle a variety of queer liquids like glue and tomato sauce, that are not very manageable with the two older and more considered types.

But in airplane service, practically all the pumps used are rotaries, mostly gear and vane types for lubrication and fuel supply, with some axial plunger types for hydraulic operations. We therefore become immediately interested in a study of inlet-side losses, as well as in general performance with and without air or vapor present in quantity. There have been serious failures of such pumps at high altitudes, particularly when air or vapor is present.

It is high time some of the hydraulics of such pumps were given the attention needed to provide good designs and installations for the purpose, i.e., successful high-altitude operation.

So little seems to be clearly perceived about capacity and efficiency that it even appears necessary to restate some principles which should be obvious.

The principal types of rotary pumps are as follows:

1 Gear: (*a*) Spur or helical external gear (any lube pump, aeronautic). (*b*) Straight or helical internal gear (Feuerheerd, Hill, Pigott).

2 Vane (Pesco, Chandler Evans, Vickers).

3 Oscillating piston (Kinney).

4 Screw: (*a*) Two screw (Quimby). (*b*) Three screw (Imo)

5 Plunger: (*a*) Radial plunger (American Engineering, Oil Gear, Northern). (*b*) Axial plunger (Waterbury, Vickers).

6 Lobular (Roots, Northern).

The following discussion applies to all types, but since we are for the moment most interested in the aviation applications, the examples and test data will be largely from gear and vane types.

Displacement of Gear Pumps

Many formulas for the displacement of spur and helical external-gear pumps have been used and published. All of those with which the author is familiar are based upon incorrect theory of what constitutes clearance volume, and some in rather general use are as much as 15 to 20 per cent in error.

The proof of this statement is that for years pump manufacturers have published test curves showing, for example, volumetric efficiency of 85 per cent at zero pressure difference. How can there be any leakage, or "slip," if there is no pressure difference? A pump always delivers substantially its full displacement at zero pressure difference, unless it is so badly designed as to be cavitating, or there is gas or vapor present.

Referring to Fig. 1, in a spur-gear pump, the full tooth-space volume of both gears is delivered around the outer side of the gears from suction to discharge side. The volume that is returned to suction from the discharge side is that which is sealed off from both sides, at the instant when two pairs of teeth first contact (seal point, see Fig. 1*a*). It is quite apparent that this volume has no relation to the volume under one tooth when fully meshed with the mating teeth on the center line between gears;

[1] Chief Engineer, Gulf Research & Development Company. Fellow, A.S.M.E.

Contributed by the Hydraulic Division and presented at the Semi-Annual Meeting, Pittsburgh, Pa., June 19–22, 1944, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

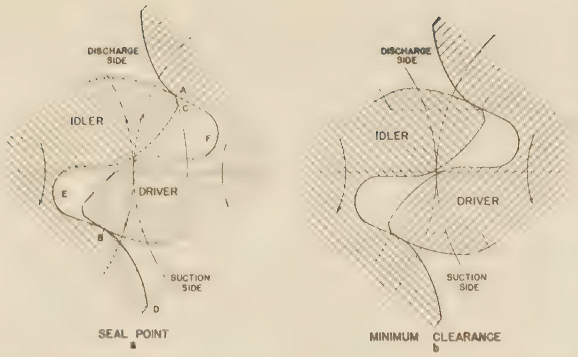

Fig. 1 Spur-Gear Pump, Seal-Point, and Minimum-Clearance Diagrams

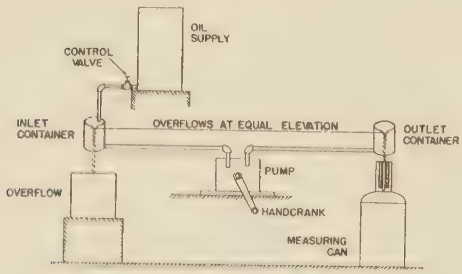

Fig. 2 Hand-Crank Displacement Test

the latter is the usual clearance volume used for most displacement formulas.

Referring to Fig. 1*b*, the two pairs of teeth are now symmetrically placed on either side of the center line; this is the position of minimum volume and is 1 to $1^1/_2$ per cent less than that at the seal point, depending upon the number of teeth and the angle of attack. This seal-point area, and that of the tooth space, can be accurately planimetered, say on 10-times-scale drawing. The proof that this operation gives displacement and clearance volume correctly is as follows:

Fig. 2 shows a pump set up for the "hand-crank test." The two reservoirs are arranged with weirs for holding the static head the same on suction and discharge; the intake side is slightly oversupplied from another tank, so that there is always a slight overflow at the weir. The pump is then slowly hand-cranked, say 100 revolutions, and the overflow from the discharge-tank weir is measured in a suitable container, i.e., a glass graduate, or a standard test can. Since there is no difference in static head, and no measurable flow loss in the pump, at very slow rotation, there can be no slip.

The results of many hand-crank tests check planimeter values within 0.1 to 0.2 per cent, much better than usual pump-test deviations. It is important to have the displacement value as nearly exact as possible, since all systematic analyses of slip and mechanical losses depend upon it, and the effect of air or other compressibles can be calculated correctly only when the displacement and clearance volume are accurately known.

A convenient proof of no slip is to perform the hand-crank test on two liquids of widely different viscosity, for example, kerosene and an S.A.E. 30 lube oil. Since the slip is inversely proportional to viscosity and directly as pressure, the two values for displacement will always differ, if there is any pressure difference in the pump due to internal losses dependent upon speed of rotation. This can be shown by cranking too fast. When the two values agree, there is no slip and no pressure difference; the value is then exactly the displacement.

Slip and Mechanical Losses

The losses in any rotary pump can be divided into two classes, which can easily be segregated in an ordinary performance test. We can, from the test and the displacement, obtain delivery in gallons per minute, displacement gallons per minute, water horsepower, displacement horsepower, and brake horsepower.

If we divide total losses into slip and mechanical losses, we can get two general values of considerable usefulness. These two values aid in detecting what details of a pump should be changed to obtain improved results.

Slip, obtained from (displacement gallons per minute—delivery gallons per minute) can be correlated with pressure difference across the pump, viscosity, and speed. Mechanical losses comprise bearing and shaft-packing friction, viscous shear in running fits at teeth or vanes, and sides of rotor, and for convenience, the hydraulic inlet and outlet losses, such as port skin friction, losses due to turns in ports, or diffusion streams, and centrifugal

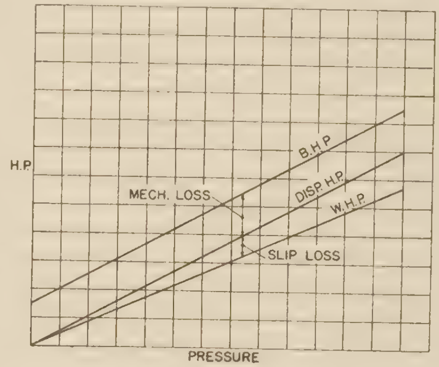

Fig. 3 Constant Speed and Viscosity, Varying Pressure

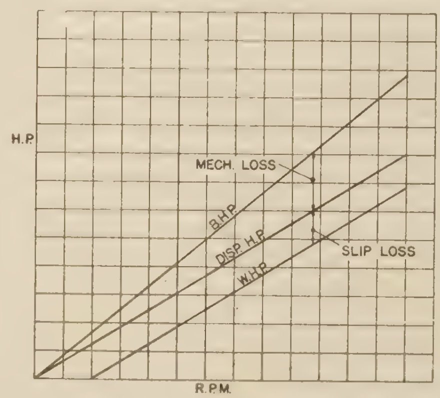

Fig. 4 Constant Pressure and Viscosity, Varying Speed

effects. Every positive-displacement pump pumps the full displacement volume, but some leaks back to suction, after the work has been done on it. Slip horsepower loss is therefore merely the difference between displacement horsepower and water horsepower as delivered. Friction horsepower, or mechanical loss, is the difference between brake-horsepower input and displacement horsepower. Fig. 3 shows the analysis for a pump running at constant speed and viscosity of liquid, varying pressure. Fig. 4 shows the same for varying speed, pressure difference, and viscosity constant.

After some struggles, we succeeded in calculating the slip and friction of rotary pumps closely enough to predict what the per-

ormance should be on test, within a few per cent. It is a somewhat tedious job, but mathematically simple. Some general orms indicate the nature of the losses.

Since all rotary pumps are capillary-sealed, i.e., close fits, the low for most liquids above water in viscosity and at pressures ess than 500 psi will be in the viscous region and follow the eneral formula

$$\Delta p = \frac{0.00173\ \mu l v}{b^2}$$

where

Δp = pressure drop, psi
μ = viscosity, Reyns (= centipoises $\times$ 0.000672)
l = length of capillary path, ft
v = velocity, fps
b = clearance, ft

If the value of $\dfrac{2b\rho v}{\mu}$ is greater than 1200, the flow is turbulent and follows the law

$$\Delta p = \frac{0.000054\, f l \rho v^2}{b}$$

where f is the friction factor and ρ is the density in lb per cu ft.

If the clearances are not in excess of 0.010 in., and ordinary ground finishes are used, the passage may be considered rough, f becomes constant at 0.054, and viscosity has no effect. For finer inish and larger clearances, roughness is low enough to make f vary with Reynolds number, and the author's curves[2] for this purpose may be used. The maximum effect of viscosity with perfectly smooth passages would be 0.25 power of μ.

The teeth of gears not in pressure contact, such as may be ound in the internal-gear pumps at the out-of-mesh position, and he rounded tips of vanes have substantially a line seal, consequently, they may be treated as rounded-approach orifices; they will be in the turbulent region at considerably higher viscosity or ower pressure than the relatively long path seals. Sides of gears, vanes or rotors, and square tops of external-gear-pump teeth will nearly always be in viscous flow.

In pumps operating at fairly high rotative speed, low or moderate pressure and very viscous liquids, there is a noticeable temperature effect, owing to the high rate of viscous shear in the capillary-seal areas, and to heat added by the energy loss from slip.

This effect may be great enough to reverse the expected and usual reduction of slip, with increase of viscosity, by heating the iquid in the clearance spaces and thus reducing the viscosity herein. For example, a pump running at 1150 rpm, peripheral speed of gear 34 fps, 100 psi pressure, the slip reaches a minimum at about 1500 sec Saybolt viscosity, and thereafter increases with viscosity of the pumped liquid.

From the foregoing, external-gear-pump slip will be largely in he viscous region for aviation lubrication pumps and slip will herefore vary chiefly as p/μ. In any case, the majority of the pumps in aviation service operate at pressure ranges from 10 to 100 psi and on viscosities of not less than 30 cp (hot), so that the slip is not very large anyway, usually 3 to 6 per cent.

The mechanical losses are more troublesome to calculate, and since there is not much interest in the matter for airplane pumps except perhaps the hydraulic pumps, it will not be detailed here.

[2] "The Flow of Fluids in Closed Conduits," by R. J. S. Pigott. *Mechanical Engineering*, vol. 55, 1933, pp. 497–501, 515; also Kent's 'Mechanical Engineer's Handbook," John Wiley & Sons, Inc., New York, N. Y., 1936.

Inlet-Side Losses

The inlet-side losses become extremely important for both oil and fuel pumps in high-altitude flying, and hitherto this interesting and valuable field of investigation has been almost completely neglected. A vacuum gage at the inlet connection of a pump gives slight indication of the minimum absolute pressure within the running parts of the pump.

Referring to Fig. 5, it is seen that one of the forces opposing

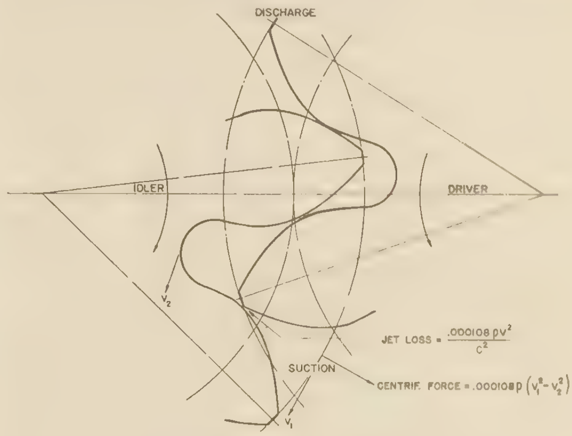

Fig. 5 Spur-Gear-Tooth Loss

flow is the centrifugal force of the ring of oil in the teeth; the second loss is the jet head required to flow the oil between the tip of one tooth and the side of the mating tooth, so as to supply the increasing volume of the space enclosed by these teeth. A third loss is the suction chamber, or port loss, and consists of the skin friction, and the diffusion of the stream from a round pipe to a much larger square section in a very short distance.

While the author has calculated these losses with quite reasonable success, fortunately it is unnecessary to depend upon any estimates, as the inlet-side loss is very easily measured.

The theoretical analysis shows, however, that the suction-side loss does vary at different positions of the gear teeth or vanes, and the measured loss by test is an intermediate value between the maximum and minimum calculated loss.

Two test methods will yield the total inlet-side losses directly. It is necessary that the liquid used have a very low vapor pressure, and be substantially free of dissolved gases. Ordinary lubricating oil, S.A.E. 10 to 50, has a very low vapor pressure, but may contain dissolved air up to 8 or 9 per cent by volume, at 1 atm and 68 F. A convenient stripping method is to restrict the inlet of the pump by means of a valve, and operate the weigh system with the pump cavitating 50 or more per cent of its normal capacity, circulating the whole body of oil in the system 5 to 10 times. Then make the cavitation tests immediately, before any appreciable air is reabsorbed. This method does quite well for most tests, but since air is slowly reabsorbed, and moreover does not strip completely, the method is not rigorous but represents rather closely the best that may be obtained in flight. A better method for air-free determinations is to use a vacuum tank weigher and operate the whole system under vacuum, sealing at possible leak points such as pump-shaft packing and valve stems with suitable oil supply. By this method, all air is positively stripped out. One method of test is to run the pump at constant differential pressure, increasing speed until the pump begins to cavitate. Fig. 6 shows such a test. Up to the cavitation point, since slip varies with $\Delta p/\mu$, and both Δp and μ are constant, the slip is constant, and the delivery line parallels the displacement.

At the point where the delivery line first departs from parallelism to the displacement line, the remaining pressure difference between the absolute pressure, measured at the inlet flange, and the barometer, is equal to the required pressure to deliver full flow to the pump.

By restricting the inlet with a valve, the pumps can be made to

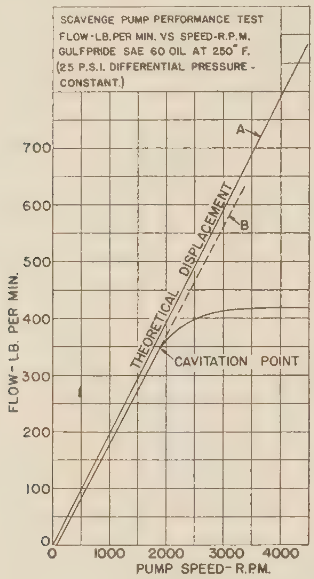

Fig. 6 Scavenge-Pump Performance Test

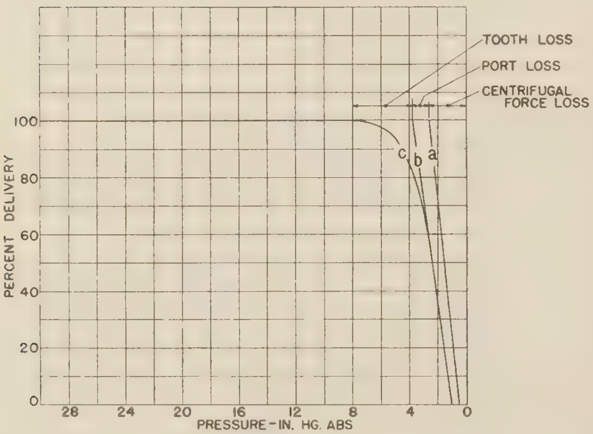

Fig. 7 Cavitation Test, Constant Speed

cavitate at a lower speed, and a plot made of pump loss versus speed.

The other method is to operate at one or more fixed speeds, constant pressure differential, and restrict the inlet until the pump cavitates, and this can be continued down to the point of pump instability. This method yields the same results and also gives performance curves exactly as the pump behaves in high-altitude flying. Moreover, it is a more convenient plot for performance testing with entrained air present, as also occurs in flight. This method of test is shown in Fig. 7. The centrifugal force is shown at a. Its value decreases from the full amount, $\Delta p = 0.000108\rho(v_1^2 - v_2^2)$ with "solid" oil, to 20 to 30 per cent of this amount at zero delivery. The reason is plain—as the pump goes into cavitation the amount of oil does not fill the tooth space. The remainder is oil vapor (no air is present). The effective density is therefore reduced and with it the centrifugal force. Centrifugal force does not decrease to zero at zero delivery, because if the differential pressure across the pump is maintained, the slip remains the same and returns to suction, and the clearance volume is also returned to suction, so that the pump at zero delivery must actually still be pumping slip and clearance volume in order to maintain pressure.

For example, in Fig. 7, the full centrifugal force is 2.77 in. Hg with the tooth spaces full of oil. The slip is 5.69 per cent of displacement, clearance volume 25.8 per cent of displacement, a total of 31.49 per cent. This value must be reduced to tooth volume as follows, remembering tooth space is displacement plus clearance: $\frac{31.49}{1.258} = 25.0$ per cent.

That is, the tooth space is 25 per cent oil, 75 per cent air or oil vapor. Neglecting vapor weight, the effective density is then 25 per cent of that full of oil, and the zero delivery centrifugal force will therefore be $2.77 \times 0.25 = 0.69$ in.

The calculated port loss added to the centrifugal force (line a) gives line b which coincides with the test line c from zero delivery up to about two thirds of full delivery, when the test line c shows a further loss, i.e., the tooth loss (see also Fig. 5). This curve shows an effect that is logically to be expected. Calculation of this loss from the simultaneous values of increasing volume and the area of the opening between the teeth, including side pockets (if any) shows that the initial tooth loss is high at the seal point, decreases rapidly as the teeth come out of mesh, and vanishes when the teeth are fully out of mesh; about 45 deg with seven teeth, for example. When inlet absolute pressure exceeds the full tooth loss, this region of high loss has a pronounced effect, but as the inlet pressure is reduced, cavitation starts in the full-mesh region in each pair of teeth and extends farther and farther as the inlet pressure is reduced an additional amount. The result is

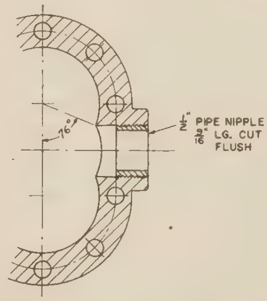

Fig. 8 Spur-Gear Pump, Condition No. 1

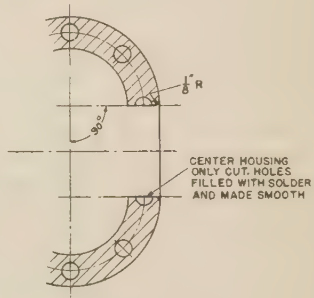

Fig. 9 Spur-Gear Pump, Condition No. 1-A

that the high-loss region is less and less well filled, consequently, the average tooth resistance drops off and finally vanishes altogether. At this time most of filling is done in the open region after the teeth are out of mesh, where there can be no tooth resistance.

These calculations are made mainly to disclose the nature and variation of the three losses; quantitative results are easily obtained by test. But such calculations show the path for improvement.

For example, one immediate indication is that the arc of gear exposed to entry of fluid on the suction side has importance. Many gear pumps are built, as shown in Fig. 8, with very little arc exposed to suction side. When opened up to 90-deg exposure, the reduction of break-point was from 7 in. to 5 in. Hg. The optimum arc exposed is 90 deg to not over 135 deg, from the center line of the gears, shown in Fig. 9.

Effect of Clearance Volume and Air

Clearance volume in a rotary pump corresponds, in a piston pump, to the space between piston and cylinder head at the end of the stroke, together with the volume of the ports from the cylinder to the valves. There is, however, a difference in operation; in the piston or plunger pump, the clearance volume is never delivered to the discharge chamber, whereas in a gear pump, the whole tooth space (which includes the clearance volume) delivers to the discharge chamber, and the clearance volume then takes a fraction of this volume back again to the suction.

If a liquid, no gas present, is handled, the clearance has no effect on the delivery; it is simply displacement minus slip. But if air or other compressible gas is present, the clearance has a pronounced effect upon performance, much the same as in a compressor.

The tooth space supplies all the oil that will reach the discharge chamber, but a volume of oil leaks back as slip, mixed of course with the compressed air, and a volume of oil and air is also returned to the suction in the clearance volume. All quantities as fractions of tooth space:

Q_t = tooth-space volume per unit of time delivered to discharge side (= 1.000)
Q_c = clearance volume oil returned to suction
Q = delivery of pump
a_t = air/mixture at tooth inlet side pressure p_t
a_2 = air/mixture at discharge pressure p_2
a_1 = air/mixture at inlet pressure p_1
p_1 = inlet pressure, in. Hg abs
p_t = tooth pressure inlet side, in. Hg abs
p_2 = discharge pressure, in. Hg abs
s_t = slip, a function of $\dfrac{\text{pressure difference}}{\text{viscosity}} \left(= \dfrac{s_d}{(1 + c_d)} \right)$
c_t = clearance volume $\left(= \dfrac{c_d}{1 + c_d} \right)$
s_d = slip, c_d = clearance $\Big\}$ in the usual term, fraction of displacement

The air delivered to the discharge side is compressed to a reduced volume.

$$a_2 = \frac{\left(\dfrac{a_t p_t}{p_2}\right)}{(1 - a_t) + \left(\dfrac{a_t + p_t}{p_2}\right)}$$

It is evident, then, that the clearance volume fills with a mixture of very much higher oil concentration than that which entered the teeth. The value for oil returned in the clearance is

$$Q_c = c_t - \frac{c_t \left(\dfrac{a_t p_t}{p_2}\right)}{(1 - a_t) + \left(\dfrac{a_t p_t}{p_2}\right)}$$

The delivery, in fraction of tooth space, is

$$Q = Q_t - s_t - Q_c - a_t$$
$$= (1.000 - c_t - s_t) - a_t + \frac{c_t \left(\dfrac{a_t p_t}{p_2}\right)}{(1 - a_t) + \dfrac{a_t p_t}{p_2}}$$

The first parenthesis is the normal full liquid delivery of the pump, no air present. The second portion is the effect of air.

To avoid confusion, it should be made clear that slip is still a function of p/μ, and it is not affected by the air present, provided the flow is viscous. In turbulent flow, the pressure drop for a given liquid flow goes up in proportion to the volume of air present, but rationalizing from the usual formulas and tests shows that it has no effect in viscous flow. We are concerned almost exclusively with viscous flow in the capillary seals of aviation pumps, on account of the small hydraulic depth and relatively high viscosity. Under these circumstances, the slip can be taken as constant, regardless of air present in the mixture, provided pressure difference and viscosity are constant.

Calculating for a series of values of a_t from 0.05 to 0.40, we get delivery, Q. Having deliveries, we can now take the pump inlet loss from a curve such as Fig. 7 for the particular pump involved, which added to p_t will yield p_1.

Having found p_1, we can convert the selected values of a_t to a_1 the air percentage at the inlet (where it will ordinarily be possible to measure it by test). Then by plotting p_1 versus a_1 and Q versus a_1, we can take simultaneous values of Q and p_1 for selected values of a_1, such as 5, 10, 15 per cent and cross plot in the form we want as shown in Fig. 10. This form of plot is the most convenient for test results likewise. In this figure, the vacuum test of the actual pump for cavitation air-free is for a pressure pump operating at 2700 rpm, 90 psig discharge pressure, aviation 120 oil at 185 F. For any other speed or discharge pressure, the figures of course change, since slip or displacement change; a change of oil temperature or viscosity alters the plots very little, since the slip is small in any case.

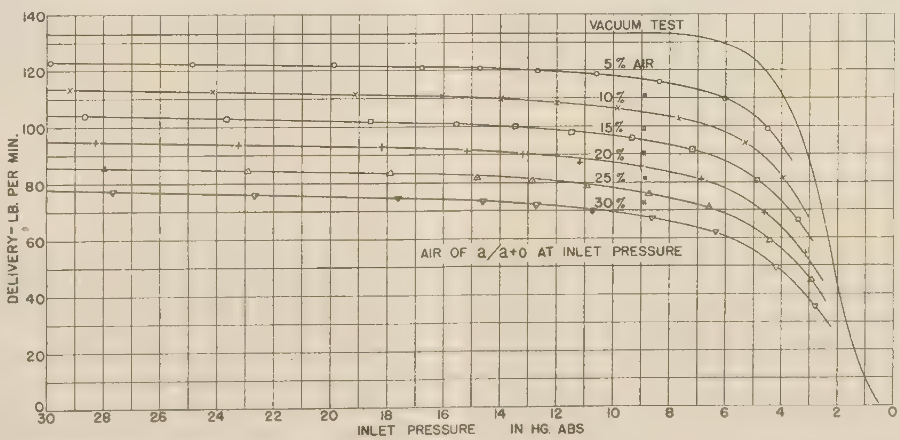

Fig. 10 Calculated Performance, Standard Spur-Gear Pressure Pump, 90 Psi Discharge Pressure

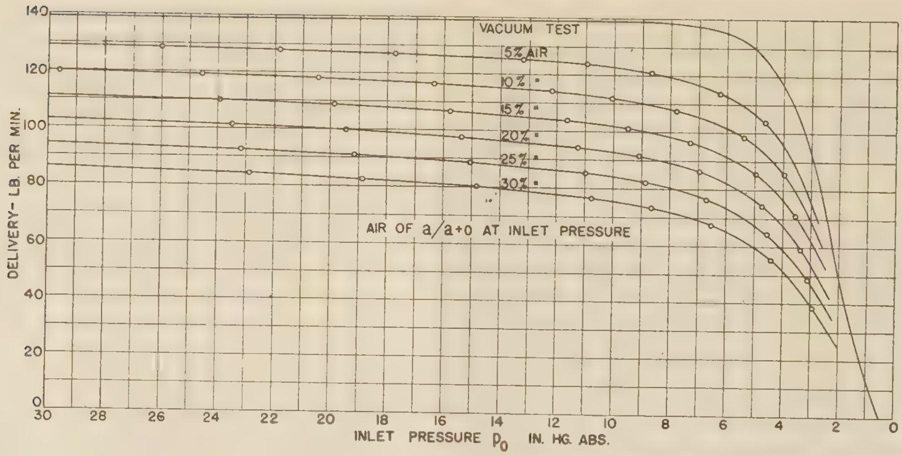

Fig. 11 Calculated Performance, Standard Spur-Gear Pressure Pump, 20 Psi Discharge Pressure

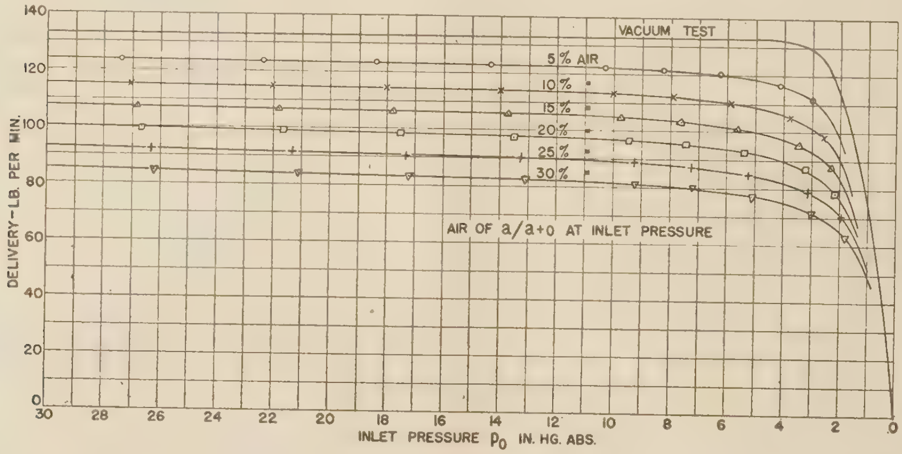

Fig. 12 Calculated Performance, Internal-Ported Experimental Pump, 90 Psi Discharge Pressure

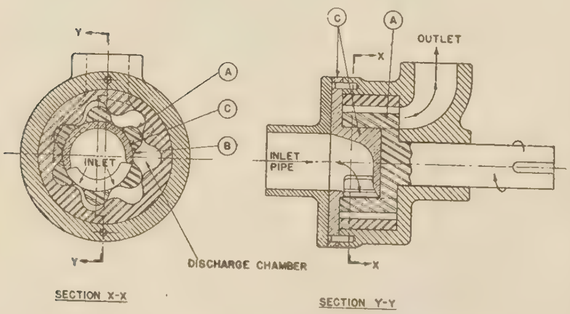

Fig. 13 Internal-Ported Internal-Gear Pump

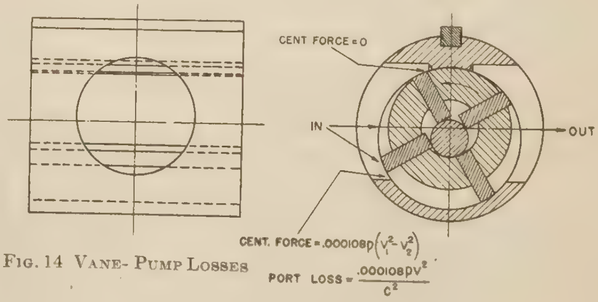

Fig. 14 Vane-Pump Losses

Fig. 11 shows a recalculation of the same pump for 20 psi discharge pressure, scavenging. Fig. 12 shows the 90-lb conditions for a pump, same delivery, but only 5 per cent clearance and a much lower intake loss. This pump is an internal-gear type, shown in Fig. 13, and avoids the centrifugal loss in the intake by taking in the oil at the center of the pinion; it is also possible to get very low values of clearance with the internal-gear design, since the tooth practically fills the tooth space; in a spur gear, clearance cannot be reduced below about 12 per cent.

Vane Pumps

The situation in vane pumps is very nearly parallel to the gear pumps; they both take in oil at the periphery, so are subject to centrifugal force; both have port loss. But there is no tooth loss in the vane pump.

Fig. 14 shows a diagrammatic view as commonly constructed. The centrifugal force at the beginning of the suction "stroke" is practically zero, as the oil between rotor drum and sleeve has no depth, but at the end of the stroke as the blade is just meeting the lower abutment, the depth is nearly twice the eccentricity so

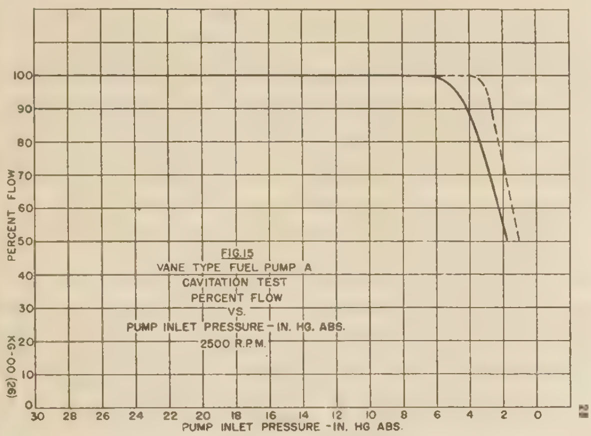

Fig. 15 Vane-Type Fuel Pump A, Cavitation Test

that the centrifugal force is nearly double that of a gear pump of corresponding diameter. The port, which is in the sleeve, is usually a round hole and therefore the area for filling, under the edge of the descending blade and the lower edge of the port, is decreasing much faster than the required rate of filling of the space between rotor blades and sleeve. Consequently, the port loss becomes quite substantial.

The cavitation test on such a pump is shown in Fig. 15 and is worse than that of a gear pump of the same diameter and speed. There is, moreover, little need for so much loss. A simple change in the port for better terminal velocities lowered the losses 30 per cent. This pump is typical of several made by different manufacturers, and it is obvious that the designers have not yet paid enough attention to the hydraulics of the operation.

Some General Data

The following is a summary of the pump situation at the present time:

Taking a pump of $1^1/_2$ in. gear or sleeve diameter, or a centrifugal of the same eye diameter, and running at 2700 rpm on aviation oil, the losses are about as follows, at the point of cavitation:

1 External gear pump, 7 in. Hg best, many designs now in actual flight up to 11 in. Hg.

2 Vane pump, 5 in. Hg best, most designs now in use 6.5 to 7 in.

3 Internal-gear pump, Fig. 13, 3.0 in. Hg.

4 Centrifugal boosters, as in use 3.5 to 5 in. Hg.

It is perfectly obvious that if we are to fly these pumps satisfactorily at 50,000 ft, which is the present aim, the total losses in suction piping from tank and to the interior of the pump must be less than 3.43 in., the barometric pressure at that altitude.

Since the piping may have losses from 1 in. Hg at full flow to as high as 7 or 8 in. in some bad cases, not much is left to fill the pump. Clearly the pumps must be designed for a total loss on the suction side of the pump which, including any pipe losses from the tank, does not exceed the barometric pressure at the maximum altitude desired.

We have several ways of attaining this end:

1 Design the pumps for low inlet loss.

2 Design the piping for lower loss than is now common, eliminating all sharp ells, restrictions, and quick disconnects, possible.

3 Put the pump drowned at the sump or tank, and avoid all inlet piping.

4 For those cases where a considerable change of piping and pump is not a feasible fix for designs now flying, use low-loss boosters at the sump or tank.

It is of course true that we can always take some advantage of the reduction of inlet loss when the pump operates below its cavitation point, which is usually the condition where failure will occur (see Figs. 10 and 12, for example), and that is where the pumps usually are operating in flight, since some entrained air is always present in the oil and probably always will be. The path for successful selection of a pump is therefore rather clearly indicated. Let us use the pressure pump as an example; to fly at 50,000 ft means 3.43 in. barometer. For a normal short line from tank to pump, we can assume the line can be designed for, say, 1.0 in. loss at 110 per cent of normal new engine flow. That will leave 2.43 in. for the pump loss. About 10 per cent air at tank at present should be allowed for a poor tank at low level of oil, and the pump can then be selected that will deliver the required 110 per cent of new engine flow, with 10 per cent air at the tank, and at 2.43-in. pump loss, from curves such as Fig. 7.

Even without any tests, these losses may now be quite accurately and safely calculated so that the design can be assured of success before building the job.

The foregoing discussion on pumps has only touched on major points; it is hoped that this sort of study will be carried further by those directly interested in building engines, pumps, and planes.

Discussion

R. A. Jeheber.[3] Referring to the section entitled, "Slip and Mechanical Losses," there are some statements which the writer feels are somewhat oversimplified and do not reveal what is actually happening in many cases. These are: "Every positive-displacement pump pumps the full displacement volume, . . . Slip-horsepower loss is therefore merely the difference between displacement horsepower and water horsepower as delivered. Friction horsepower, or mechanical loss, is the difference between brake-horsepower input and displacement horsepower." Figs. 3 and 4 of the paper are based on these statements.

The writer bases his opinion on past experience with reciprocating pumps; it should apply to rotary pumps as well, as they are the same type, that is, positive displacement. Several years ago when the majority of liquids pumped (mostly water) were practically incompressible, the author's statements would have corresponded closely to the truth. Developments in oil-refinery practice, however, soon showed that the word "slip" contained more than one type of loss.

If we start from the common accepted definition of "slip" as used by the author (displacement in gallons per minute minus delivery in gallons per minute), we can divide this slip into two main types as follows:

1 *Full Power-Loss Slip.* In other words liquid which leaks back to the suction after full discharge pressure has been built up in it. Examples of such slip are leakage past valves or pistons (in reciprocating pumps), or clearances (in rotary pumps) from full discharge pressure to suction pressure.

2 *No (or Low) Power-Loss Slip.* In this category we have such types as the following:

(*a*) Liquid which falls back to the suction before pressure builds up (for example, the liquid which falls back through the suction valves of a reciprocating pump while they close, before the build-up of pressure. Another example, rather extreme to illustrate the point, would be a four-plunger pump with one suc

[3] Director of Research, Mack Manufacturing Corporation, Plainfield, N. J. Mem. A.S.M.E.

tion valve stuck open so that one plunger, although it travels back and forth, does not do any effective work; assuming ideal operation for the other three plungers this positive-displacement pump would pump only three quarters of the full displacement volume).

(*b*) Liquid which cannot be drawn in because it has been displaced by air or vapor (an extreme example of this would be to connect air instead of the liquid to the suction valve[s] of one plunger. In a four-plunger pump, this would again mean that this positive-displacement pump would pump no more than three quarters of the full displacement volume).

(*c*) Liquid which cannot be drawn in because of expansion of the liquid trapped in the clearance space. To illustrate, let us imagine a pump handling compressible liquids which expand in the clearance space so that its plungers travel one quarter of their suction stroke before the suction valves can open; here again this positive-displacement pump would pump no more than three quarters of the displacement volume.

There are of course some volumetric losses which fall partly in the first and partly in the second category, such as, for example, liquid which leaks back while the pressure is only partly built up. In a 1000-lb differential-pressure pump, for instance, the amount which leaks back from 250 lb differential pressure might be one-quarter loss of the first type and three quarters loss of the second type.

This second type of slip is far from negligible for such liquids as hot oil or propane close to the boiling point where they are quite compressible and may also pull vapor.

As a matter of fact this was brought forcefully to the writer's attention in some tests with large reciprocating-plunger pumps where the effect of this second type of slip was so great that the brake horsepower required to drive the pump was less than the displacement horsepower based upon geometrical displacement (geometrical displacement of the pump in gallons per minute times differential pressure of pump in pounds per square inch divided by 1715). In plunger pumps there is not much argument as to what constitutes the geometrical displacement.

It might be said that this is a special case of a pump which functions partly as a compressor. This may be true, but there are enough modern applications in oil-refinery work and high-altitude flying to warrant its recognition.

The author is fully aware of this second type of slip, or should we say, volumetric loss; he recognizes some of it in other parts of his paper as "effect of air or other compressibles" also "effect of clearance volume and air;" only he does not call it "slip." This would suggest that the disagreement could perhaps be one of definitions and words rather than one of principle. Possibly the author has in mind making a correction to a "reduced displacement" which would be equal to the geometric displacement minus a certain value representing this second type of slip. If this is the case, it would be interesting to know how the various factors are evaluated in such a correction.

It is agreed with the author that theories used in the pump industry should undergo a general overhaul so that they can be adapted to problems encountered in present-day practice.

W. L. Weeks.[4] During the past 2 years or so the Wright Aeronautical Corporation has worked very closely with the author's organization on problems of air entrainment and of refinement of oil-pump design.

Since the investigation of inlet ports and tooth-loss effects, made by progressively cutting away the inlet side of a submerged pump, was one of the jobs they did for us, it is natural that we are in general agreement with this paper.

[4] Project Engineer, Wright Aeronautical Corporation, Paterson, N. J.

In the successful effort in making his paper a digest, some points are necessarily condensed. With some slight urging, the author might expand a few points. For instance, a more detailed concept of the centrifugal-loss line *a* in Fig. 7 might be of interest. The author might explain that, while for purpose of general discussion, this line may be considered straight, it actually is a slightly curved one as calculated for conditions beyond the start of cavitation. To us it is an interesting even if not a highly important point.

Another item which is interesting in some phases beyond the present exposition is the one the author calls "jet loss."

When we started the research with Gulf, we already knew inlet ports could be bad if too narrow. As a matter of fact, we had widened them in some pumps where the location of studs would permit. But the pump's volumetric efficiency at altitude still left much to be desired. At that time we had no evidence that the spur-gear pump was not as good a type as any, for the job; and we had investigated others.

The question of volumetric efficiency being tied in with pump gear-tip speed gave us an incentive to check empirical figures we had heard on tip-speed practice. All told, we felt that every factor in the action of a spur-gear pump should have its possibilities exhausted by research.

We soon found that our program dovetailed very well with some analytical studies the author had made. The now completed research leaves the spur-gear-pump ceiling quite predictable and the limitations of this type pump well established before we adopt other types.

A very clear picture of various loss factors is given in this paper, but one which concerned us a year ago is slighted. It deserves consideration.

The author has pointed out that loss by centrifugal force can be evaluated. He shows that loss in the unmeshing process of the gears, labeled "jet loss," is not only a difficult one to evaluate, but that it can easily be avoided by making the inlet port reasonably wide.

In the research with the mutilated and submerged oil pump, one factor we tried to clarify was that called shock loss, namely, the loss caused by the struggle of the liquid to get in behind a gear tooth as it speeds across, and even partly against, the path of liquid flow through the inlet port.

Referring to Fig. 7 of the paper, and the "test line *c*," it may be said that even in the flow plot of a pump with a very wide inlet port, this line *c* is still a fairly easy curve, showing a loss imposed aside from the computable port and centrifugal ones. And yet, with such a wide inlet port, the author says (and we certainly subscribe) the "jet loss" is practically negligible. A reasonable deduction might be, then, that test line *c* represents the loss due to shock of sudden change of direction of liquid entering the rotors, even during exposure of wide-open tooth spaces. In fact this interpretation is indicated in the report of the tests thus: ". . . . the shock losses are practically independent of the gear exposure." Also, in the discussion section of the report, where housing-inlet, centrifugal, and acceleration losses (the latter, the author has spoken of, in his present paper as jet loss) are treated, the last item is shock loss. But, although given last place in discussed losses, this loss is evaluated in the over-all, as follows: "The major portion of the loss indicated by the tests, after deducting the calculated centrifugal and inlet friction losses, is that due to shock brought about by sudden changes in the velocity and direction of oil flow." We certainly agree with the report statement.

It is believed the author will agree that the flow curves from the internal-ported gear-type pump, shown in section in Fig. 13, show more delay in cavitation than can be explained by centrifugal aid to rotor filling, and designedly low inlet-port loss, when

compared with the flow curve from a wide-ported spur-gear pump. This could be explained by the fact that, although in both types of pump the teeth move practically crosswise of oil flow, the cross-sectional shape of the pinion (or inner rotor) teeth and the profile of the teeth of the outer rotor in the I. P. pump are much cleaner in point of flow lines, than are spur-gear teeth to radial entry of liquid.

Does the author feel that this comparison further supports the status given to shock loss, or what we like to call tooth turbulent-entry loss, in spur-gear pumps, as stated in the test report?

In no sense is the foregoing questioned correlation pertinent to the status of the internal-ported pump in the aviation industry. If its outstanding volumetric efficiency at extremely low inlet absolute pressures is 99 per cent due to centrifugal aid to rotor filling, we do not care in the least. This new and cleverly developed type of pump has become available almost exactly when we are convinced that the inherent limitations of the spur-gear type shows it to be unsatisfactory for service at altitudes of 40,000 ft and upward. Although the author has not pushed this new development particularly, the writer takes the opportunity to say that we cannot afford to neglect its potentialities.

Author's Closure

Referring to the discussion by W. L. Weeks, possibly he has confused the losses in the inlet port with those which may occur in the inlet chamber between the pipe port and the gears. The normal losses in the port and the inlet chamber are, in all cases, relatively small for any reasonably well-designed pump, but the amount of the pinion arc that is exposed to the suction chamber is quite important. This situation has been discussed in the paper and is also covered by Mr. Weeks.

With regard to his point that the jet loss, by which the author means the flow between the tip of one tooth and the flank of the next, can be eliminated by making the inlet port reasonably wide, while this is not subject to exact determination, in general, the statements made in the paper and those by Mr. Weeks are in agreement. The jet loss still exists in reduced amount when the pump is in cavitation, but by exposing more arc of the teeth in the unmeshed region, a greater time is allowed for filling the incompletely filled teeth, and therefore the loss occurring in the meshed region may not be appreciable or even detectable. It is somewhat difficult to get exact terminology to describe these situations, but in Fig. 7 of the paper, the port loss as indicated in this figure includes any loss due to the pipe opening into the pump, the skin friction in the inlet chamber, and any velocity-head-type loss due to changes of direction of the stream from the usually circular inlet pipe to the usually rectangular configuration where the oil enters the gears. It will be seen from the diagram that the port or intake-chamber loss forms a relatively small percentage of the total when the pump is either not cavitating or is in low order of cavitation, but assumes a somewhat larger percentage of the total after the tooth loss has substantially vanished in the high order of cavitation. It is possible that Mr. Weeks has misunderstood the application of the term "jet loss." The author was thinking almost exclusively of the high-speed jet occurring between the tip of one tooth and the flank of the next, which is actually the bulk of the tooth loss. The other velocities elsewhere, particularly in the port and intake chamber, are relatively rather low.

With regard to Fig. 13 of the paper, part of the gain shown by this design of pump is of course elimination of centrifugal force. Another fraction of the gain is due to proper shaping of the gear-tooth ports for good pickup, just as in a centrifugal pump; and another fraction of the gain is due to the great reduction in clearance. His point is correct, that the proper shaping of the intake edge of the gear teeth, so as to introduce the liquid radially from the intake cup without shock, is quite important. The difference between a plain radial port and the ports as now shaped amounts to as much as $1^1/_2$ to 2 in. Hg. It is the author's feeling that the improvement in this pump is not traceable to any one cause but to several of the structural changes that have been accomplished. These are about as follows:

1 Elimination of centrifugal resistance.

2 Good turn conditions and faired shapes in the inlet cup.

3 Proper proportioning of the port area in the gear, together with proper angle of the inlet edge to reduce, as far as possible, any shock loss or change in direction of the fluid as it enters the teeth.

4 The reduction of clearance from approximately 20 per cent to around 5 per cent.

In response to Mr. Week's kind remark about the potentialities of this design, the author has rather avoided pushing this design too much, since this is a technical paper, and presumably quite unbiased. However, it is within the bounds of propriety to say that, at the present time, this pump has the lowest inlet loss of any we have investigated, including some centrifugal boosters.

In reply to Professor Schweitzer's oral comments the author is in agreement that, in some sense at least, displacement is a conventional term. However, the reason why we have adopted displacement in the form given in the paper is: (1) It is in exactly the same status as the displacement of a reciprocating pump and in all of the commercial rotary pumps, and therefore has very widespread use. (2) The distinction between slip and mechanical loss can only be made correctly upon the basis of this form of displacement. They do not correlate properly with the geometric displacement described by Professor Schweitzer which in effect corresponds in the present paper to the tooth space. Mr. Hambling of the Thornton Laboratories in England also applied Professor Schweitzer's method of using the geometric displacement, with the result that all analyses based upon volumetric efficiency give very deceptive values.

With regard to the assumption that slip may occur elsewhere than from the discharge side to the inlet, two points should be made as follows:

In the analysis as made in the paper and in 90 per cent of the pumps, there is no external shaft leakage because for a commercial pump the shaft is packed. The only case where shaft leakage is permitted is in aviation pumps where leakage along the shaft either passes back to a sump or travels through to another pump in the same case. It is believed Professor Schweitzer's assumption that some of the leakage may have come from oil carried into the tooth space on its way from suction to discharge, and leak out before it reaches the discharge, is an incomplete rationalization. It must be realized that, while some of this leakage is taking place all along this path, what material does leak out is from an intermediate-pressure region and the tooth must be refilled from some other source or the leakage would not take place. Now it is obvious that between the point where this particular quantity of oil has leaked back before completing the path from suction to discharge chambers, there is also pressure difference between this pressure and the discharge, and leakage comes down from discharge to this point to replace that which has leaked out at the lower pressure point. The net result is, therefore, that the entire displacement work is done on the total difference of pressure, and all leakage is truly represented by calculation on the total pressure difference between inlet and discharge. It is therefore clear that the work done on the liquid in the pump is always that of pumping the entire displacement against the difference of pressure between inlet and discharge and is not in any way affected by what place, in the pump, any particle of oil came from to join the leakage. This situation is not an approximation, it is entirely rigorous.

Effect of Combined High Temperature and High Humidity on the Corrosion of Samples of Various Metals

By W. L. MAUCHER[1] and B. W. JONES,[2] SCHENECTADY, N. Y.

The atmosphere around and in oil refineries is corrosive; and when these refineries are located in hot humid locations, this atmosphere attacks various metals actively. Four groups of eighteen selected metal specimens were subjected to this corrosive atmosphere in a refinery along the Gulf Coast. Two groups were located indoors and two groups outdoors for about one year. This paper is a report of how these specimens withstood this hot, humid, corrosive atmosphere.

VARIOUS metals when subjected to the combination of high temperatures, high saline humidity, and to the gases which are found around and in the various oil refineries along the Texas Gulf Coast exhibit a very high rate of corrosion. This condition has been the cause of much destruction to enclosing cases, structural members, and to electrical parts which are magnetic. This gas also rapidly attacks current-carrying parts such as copper, cadmium, silver, and other materials, so that the maintenance personnel in these plants would welcome any suggestions that would reduce this type of corrosion.

The present paper is a report of some work done to ascertain the degree of corrosion which this condition would have on eighteen selected materials or platings. Four groups of these eighteen selections per group were placed in four selected locations of an oil refinery which was adjudged to be the most severe plant in that area as regards corroding. Each specimen was hung on a porcelain-enameled hook so that the specimen did not touch any other member. No. 1 location was inside a power plant; No. 2 location was outside this same power plant, where the samples were subjected to the rain and sun; No. 3 location was inside a chilling pump house; and No. 4 location was outside this same chilling pump house.

A competent person inspected these seventy-two specimens at the expiration of 7, 14, 35, 72, 129, and 225 days at the test sites, and Tables 1 to 4, inclusive, are his report. These specimens were returned to the laboratory after 1 year exposure and W. L. Maucher who originally prepared them also inspected them on their return. Tables 5 and 6 are from his report. Therefore, these two reports give a comprehensive description of what occurred during this period of 1 year in a saline atmosphere containing a substantial concentration of sulphur acids, such as hydrogen sulphide and sulphurous and sulphuric acids, and we should be able to draw a few useful conclusions from the results. The list of metals as shown in the next column might be generalized as follows: (*a*) Five simple elements; (*b*) electroplated coatings on copper; (*c*) chromium alloys; (*d*) copper alloys; and (*e*) nickel alloys.

LIST OF 18 DISTINCTIVE SPECIMENS OF METALS

Specimen number	Material
1	Rolled copper
2	Rolled copper with 0.001 in. silver electroplate per side
3	Rolled copper with 0.001 in. lead electroplate per side
4	Rolled copper with 0.001 in. tin electroplate per side
5	Rolled copper with 0.001 in. cadmium electroplate per side
6	Rolled nickel
7	Type 410 12 per cent chrome steel passivated
8	Nichrome V
9	Lead
10	Type 302 18-8 stainless steel passivated
11	2S aluminum
12	2S aluminum anodized for 20 min
13	Cold-rolled steel
14	Fine silver
15	Monel
16	Everdur
17	Red brass (85 Cu, 15 Zn)
18	25 per cent chrome steel passivated

1 Of the four electroplated coatings on copper, the tin appeared to stand up best. The cadmium was completely gone and the silver showed heavy attack, as evidenced by the scaly black deposit which was probably silver sulphide. The lead plating did not look so bad on the surface but on closer inspection, deep pits of the copper base metal were noted.

2 Rating the chromium alloys in their order of merit, 18-8 stainless steel and Nichrome V average up about the same and rate about the best in this test. The 25 per cent chrome steel was somewhat worse, and 12 per cent chrome steel was decidedly inferior to the other three.

3 While aluminum, untreated, showed a uniform corrosion coating and the anodized aluminum had scattered pits, it is felt that this metal showed up nearly as well as did the 18-8 stainless steel. Considering price, it is the consensus of opinion that aluminum, anodized or otherwise properly treated for paint, and then given two or more coats of suitable aluminum paint, would provide one of the most corrosion-resistant combinations.

4 The test on copper and copper alloys showed conclusively that the high-copper metals have a low resistance to corrosion in this kind of atmosphere and should not be used unless protected by some protective coating.

5 A comparison of monel and nickel indicate that nickel was not attacked as much as was monel.

6 The extremely bad corrosion of the unprotected cold-rolled steel indicates the severity of the corrosive atmosphere at this location and emphasizes the necessity for maintenance painting on any steel parts.

7 Lead showed up outstandingly well but because of its low physical properties would not lend itself to many applications.

8 The interesting observation on silver was that, although it was heavily attacked, there was no sign of pitting or noticeable etching. The metal underneath the corrosion film was very smooth. On large knife switches, the silver surfaces continued to carry their currents satisfactorily.

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Contributed by the Aviation Division and presented at the Semi-Annual Meeting, Pittsburgh, Pa., June 19–22, 1944, of The American Society of Mechanical Engineers.

TABLE 1 RESULTS OF EXPOSURE OF SPECIMENS TO CORROSION GASES
(Field Inspections)

SPECIMEN NUMBER	2-20-40 7 DAYS EXPOSURE	2-27-40 14 DAYS EXPOSURE	3-19-40 35 DAYS EXPOSURE
INSIDE POWER PLANT			
1	Colored Red & Green. Spotted.	Gray Film Some Copper Showing.	Dark Gray
2	Dark Brown Coating. Spotted.	Coated Greenish Blue.	Dark Brown all over.
3	Slightly tarnished. Clear	Tarnished to Bluish Gray.	Slate Gray. Spotted.
4	Very Slightly Tarnished. Clear	Light Gray Coating.	Dull Gray. Clear
5	Bright. Several Dark Spots.	Bright. Silvery Color. Few Spots	Semi-Bright. Rainbow Colors.
6	Very Slightly Dulled. Clear.	Slightly Dulled. Clear	Dull Gray. Clear.
7	Very Slightly Dulled. Clear.	Slightly Dulled. Clear.	Dull. Clear.
8	Bright - Like New.	Bright - Clear.	Slightly Dulled. Clear.
9	Darkened. Clear	Darkened. Clear.	Dull. Clear.
10	Bright and Clear.	Bright - Clear.	Semi Bright - Clear.
11	Bright and Clear.	Bright - Clear.	Bright - Clear.
12	Bright and Clear.	Bright - Clear.	Bright - Clear.
13	Darkened - Clear.	Light Coat of Rust. All over.	Rust all over.
14	Very Tarnished. Spotted.	Very Tarnished.	Very Tarnished.
15	Tarnished. Clear.	Dull Gray. Clear.	Dull - Rust Color.
16	Rainbow Colored.	Tarnished - Spotted.	Very Dark with few Bright Colors.
17	Colored Green & Red.	Tarnished - Spotted.	Very Dark - Spotted.
18	Bright and Clear	Bright and Clear.	Semi Bright and Clear.
OUTSIDE POWER PLANT			
1	Rainbow Colored - Dark Spots.	Tarnished - Very Spotted.	Dull Gray. Black Spots (rough).
2	Blue Cast with Red Spots.	Tarnished. Brown Spots.	Dull Gray - Black Spots (Rough).
3	Gray - Spotted.	Dull Gray. Spotted.	Dull Gray - Black Spots (Rough).
4	Slate Colored.	Dull Slate Color	Dull Gray - Black Spots (Rough).
5	Bright-Some Green Spots.	Yellow Powdery Coating.	Yellowish Green -Rough
6	Darkened - Clear.	Dull, Many Dark Spots.	Black, Spotted, Rough
7	Large Rust Spots.	Rusted all over.	Rust all over.
8	Slightly Dull - Clear	Dulled - Mottled.	Dull Gray - Spotted.
9	Darkened - Some Spots	Dull - Spotted.	Dull - Spotted.
10	Bright and Clear	Rust Spots Showing, Dull.	Dull - Brown Spots.
11	Bright and Clear	Dull, Gray Powder Coating.	Dull - Rough
12	Bright and Clear	Bright - Clear.	Semi Bright, Rough Black Spots.
13	Rusted all over.	Rust all over.	Very Rusty all over.
14	Tarnished near Black	Tarnished Dull Gray	Tarnished near Black
15	Slight Tarnish, Clear.	Tarnished Dull Gray	Dull Gray - Rough
16	Tarnished - Spotted.	Dull - Spotted	Dull Gray - Rough
17	Tarnished - Spotted.	Dull - Spotted	Dull Gray - Rough
18	Bright and Clear	Bright - Clear	Some Bright Areas - Brown Spots, Rough.

TABLE 2 RESULTS OF EXPOSURE OF SPECIMENS TO CORROSION GASES
(Field Inspections)

SPECIMEN NUMBER	4-26-40 72 DAYS EXPOSURE	6-26-40 129 DAYS EXPOSURE	10-1-40 225 DAYS EXPOSURE
INSIDE POWER PLANT			
1	Black Sooty Film - Easily Rubbed Off.	Rough - Black	Rough - Black
2	Black, Spotted, Smooth.	Slightly Rough, Dull Black.	Slightly Rough - Plating O.K.
3	Slate Gray, Smooth, Some Spots.	Dull Gray - Smooth.	Smooth - Dull. Plating O.K.
4	Light Gray, Smooth Clear.	Dull Gray - Smooth.	Smooth - Dull. Plating O.K.
5	Yellowish & Iridescent- Smooth.	Yellowish - Smooth.	Smooth - Yellowish Plating Oxidizing.
6	Dull Gray, Clear, Smooth.	Dull Gray, Smooth, Clear.	Considerably Etched.
7	Dull Gray, Clear, Slightly Rough.	Dull Gray, Slightly Rough, Clear.	Slight Rust Showing.
8	Semi Bright-Slightly Rough.	Semi Bright, Smooth Clear.	Etched.
9	Dark Gray, Smooth, Clear.	Lead Color, Smooth Clear.	Lead Gray Color. Smooth.
10	Semi-Bright, Smooth Clear.	Semi-Bright, Smooth Clear.	Good Condition
11	Bright Smooth Clear	Bright Smooth Clear	Slightly etched.
12	Bright Smooth Clear	Bright Smooth Clear	Very Good.
13	Rusted all over, Very Rough.	Very Rusty & Rough	Very Rusty & Rough
14	Dark Smooth Spotted	Very Dark, Smooth, Spotted.	Very Tarnished - Smooth.
15	Black Smooth Clear	Black, Slightly Rough	Slightly Etched.
16	Black Smooth, Some Spots.	Black, Slightly Rough	Coated with Copper Sulphide.
17	Black Smooth Flaking	Bluish Black, Slightly Rough	Coated with Copper Sulphide.
18	Bright Clear Smooth	Bright - Smooth.	Very Good Condition.
OUTSIDE POWER PLANT			
1	Dark, Iridescent, Spotted, Rough	Blue-Black, Very Rough	Severely Corroded, Rough, Black
2	Dark Gray, Spotted Rough.	Dull Gray, Very Rough.	Severely Corroded, Dull Gray.
3	Light Gray, Rough, Black Spots.	Blue-Black, Slightly Rough.	Plating OK except when Samples Rubbed.
4	Light Gray-Rough, Black Spots.	Dull Gray, Slightly Rough.	Plating OK except when Samples Rubbed.
5	Yellow-Rough Flaking	Yellowish, Very Rough	Plating Wearing Off by Flaking.
6	Black, Rough, Spotted.	Tarnished Bronze Slightly Rough.	Thick Coating, Etched
7	Rusted all over. Rough	Rusted all over Rough.	Very Rusted & Pitted
8	Dull Gray, Slightly Rough, Spotted.	Dull Gray - Rough.	Thin Coating, Badly Etched.
9	Dull, Slightly Rough.	Lead Color, Slightly Rough.	Etched.
10	90% Area Rusted	95% Area Rusted, Rough.	Covered with Rust Spots

TABLE 2 (*Continued*)

	SPECIMEN NUMBER	4-26-40 72 DAYS EXPOSURE	6-26-40 129 DAYS EXPOSURE	10-1-40 225 DAYS EXPOSURE
OUTSIDE POWER PLANT	11	Dull, Very Rough.	Dull, Very Rough.	Dull, Rough Coating.
	12	Semi-Bright, Rough Black Spots.	Dull & Semi-Bright Spots. Rough.	Semi-Bright to Dull Rough.
	13	Very Rusty all over, Rough.	Rusted all over, very rough.	Very rusty.
	14	Tarnished to near black. Spotted Flaking.	Blue-Black. Smooth	Tarnished to Blue-Black.
	15	Dark Gray - Rough.	Blue-Black. Very Rough.	Severely etched.
	16	Flaked to Bright Surface. 75% Balance Rough.	Blue-Black. Very Rough.	Severely etched.
	17	Flaked to Bright Surface. 75% Balance Rough	Blue-Black. Very Rough.	Thick Coating
	18	90% Area Rusted. Rough.	95% Area Rusted. Very Rough.	Covered with Rust Spots.

TABLE 3 RESULTS OF EXPOSURE OF SPECIMENS TO CORROSION GASES

(Field Inspections)

	SPECIMEN NUMBER	2-20-40 7 DAYS EXPOSURE	2-27-40 14 DAYS EXPOSURE	3-19-40 35 DAYS EXPOSURE
INSIDE CHILLING PLANT	1	Very Spotted & Tarnished.	Red, Green & Copper Colors	Brown, Red & Gray Casts.
	2	Colored to varying shades of brown.	Mottled - Brown.	Varying shades of brown.
	3	Dark Gray, Clear.	Dark Gray, Clear.	Dark Gray, Clear.
	4	Light Slate Color, Clear.	Slate Colored, Clear.	Slate Colored, Clear.
	5	Bright - Clear.	Bright - Clear.	Bright - Clear.
	6	Bright - Clear.	Bright - Clear.	Bright, Few Brown Stains
	7	Dull - Clear.	Dull - Clear.	Dull Gray - Clear.
	8	Bright, Like New.	Bright - Clear.	Bright - Clear.
	9	Dull, Some Spots.	Dull, Spotted.	Dull, Clear.
	10	Dull, Clear	Dull, Clear.	Semi-Bright, Clear.
	11	Bright, Clear.	Bright, Clear.	Bright, Clear.
	12	Bright, Clear.	Bright, Clear.	Semi-Bright, Clear.
	13	Bright, Clear.	Bright, Pimply Spots	Bright, Pimply Spots
	14	Very Tarnished	Tarnished Black & Red Casts.	Tarnished Black & Red Casts
	15	Bright - Clear.	Bright - Clear.	Semi-Bright, Clear.
	16	Stained with Dark Brown Spots.	Bronze Color - Large Brown Spots	Brown-Red & Gray Mottled
	17	Stained with Dark Brown Spots.	Bronze Color - Mottled	Bronze Color, Red & Brown Mottled.
	18	This sample was Lost		

TABLE 3 (*Continued*)

	SPECIMEN NUMBER	2-20-40 7 DAYS EXPOSURE	2-27-40 14 DAYS EXPOSURE	3-19-40 35 DAYS EXPOSURE
OUTSIDE CHILLING PLANT	1	Very Tarnished and Spotted	Tarnished Black & Green	Black & Red Casts - Rubs Off
	2	Very Tarnished and Spotted	Tarnished, very dark, Spotted	Very dark, Spotted
	3	Tarnished to Dull Gray	Tarnished, Spotted	Gray, Spotted
	4	Tarnished Dull Gray	Tarnished, Dull Gray	Dull Gray, Spotted
	5	Tarnished to Green & Gray Colors	Gray, Mottled	Dull Gray, Rough, Powdery.
	6	Tarnished Dull Gray	Very Dull, Clear	Dull Gray, Yellowish, Rough
	7	Large Rust Spots	Rust all over	Rust all over
	8	Bright Like New	Dull Gray, Clear	Dull, Green Spots, Raised.
	9	Dull, Some Spots	Dull, Spotted	Dull
	10	Bright, Some Spots	Dull & Bright Areas	Dull and Bright Areas
	11	Bright & Clear one side. Spotted on South Side	Gray Powder Coat - Dull	White, Rough, Coating
	12	Bright & Clear	Some Gray Powder - Bright	Semi-Bright, Rough with white spots
	13	Rusted all over	Rusted all over	Heavy rust coating
	14	Very Tarnished	Tarnished, Black & Red Casts	Tarnished Black
	15	Tarnished & Spotted	Dark - Spotted	Black, Some Green Spots
	16	Very Tarnished and Spotted	Tarnished, Green Spots	Black, Some Green Spots
	17	Very Tarnished and Spotted	Tarnished to Black	Black, Some Geeen Spots
	18	Bright and Clear	North Side Bright, Mottled, South Side Brown Spots	North Side Bright, South Side Brown Spots

TABLE 4 RESULTS OF EXPOSURE OF SPECIMENS TO CORROSION GASES

(Field Inspections)

	SPECIMEN NUMBER	4-26-40 72 DAYS EXPOSURE	6-26-40 129 DAYS EXPOSURE	10-1-40 225 DAYS EXPOSURE
side Chilling Plant	1	Brown, Red & Gray Casts, Smooth	Bronze & Black Spots Smooth	Tarnished to Bronze and Black Spots
	2	Varying Light & Dark Shades of Brown, Smooth	Light & Dark Brown Smooth	Tarnished Plating O.K.
	3	Bluish Gray, Smooth	Lead Gray, Smooth	Gray, Plating O.K.
	4	Light Gray, Smooth	Light Gray, Smooth	Gray, Plating O.K.
	5	Bright, Smooth, Clear	Yellow (Pale), Smooth	Lemon Yellow, Plating O.K.
	6	Bright, Smooth, Some Brown Stains	Bright, Smooth, Few Spots	Dulled, O.K.
	7	Dull Gray, Smooth, Clear	Dull Gray, Smooth, Few Spots	Dulled, O.K.
	8	Bright, Smooth, Clear	Bright, Smooth	Dulled, O.K.
	9	Dull, Smooth, Clear	Lead Color, Smooth	O.K.
	10	Semi-Bright, Smooth Clear	Dull, Smooth	O.K. (Dulled)
side Chilling Plant	11	Bright, Smooth, Clear	Bright, Smooth	Bright, O.K.
	12	Semi-Bright, Smooth, Clear	Bright, Smooth	Bright, O.K.
	13	Bright, Pumply Spots	Bright, Smooth	Very Rusty
	14	Tarnished to Black and Red, Smooth	Nearly Black, Smooth	Tarnished Blue-Black
	15	Semi-Bright, Smooth Clear	Semi Bright, Smooth	Dull, Spotted
	16	Gray, Red & Brown Mottled, Smooth	Black & Bronze, Smooth	Tarnished - Varigated Colors
	17	Red & Brown, Smooth	Spotted Red & Bronze Smooth	Tarnished, Black and Bronze
	18	This sample was lost		

TABLE 4 (*Continued*)

	SPECIMEN NUMBER	4-26-40 72 DAYS EXPOSURE	6-26-40 129 DAYS EXPOSURE	10-1-40 225 DAYS EXPOSURE
Outside Chilling Plant	1	Gray - Smooth	Black to Bronze, Smooth	South Side Badly Attacked, North Side not so bad.
	2	Very dark, rough	Dark Brown, Spotted, Rough	Plating Good, Few Rough Spots
	3	Gray, Rough, Spotted	Lead Gray, Smooth	Plating Good, Few Rough Spots
	4	Light Gray, Rough Spotted	Gray, Very Rough, Spotted	Much of Plating Gone, Pitted.
	5	Dull Gray, Rough, Powder Surface	Yellowish, Slightly Rough	Plating all gone.
	6	Green & Yellow, Rough	Spotted, Smooth	Tarnished, Slightly Rough
	7	Rusted all over, Rough	Rusted all over Very Rough	Rusted all over
	8	Green Cast - Rough	Dull, Slightly Rough	Etched & Slightly Rough
	9	Dull, Smooth, Spotted	Lead Color, Smooth	Turned to Light Gray
	10	75% Area Rusted - Rough	80% Area Rusted - Rough	Covered with Rust Spots
	11	Dull - Very Corroded	Very Much Corroded, Rough	Very Corroded
	12	Dull, Rough	Very Much Corroded, Rough	Very Corroded (Not as bad as #11)
	13	Rusted all over, Rough	Rusted all over, Scaling off	Plate 50% Gone
	14	Tarnished Black, Spotted, Smooth	Tarnished & Black, Smooth	Tarnished, Smooth
	15	Black, Smooth	Very Rough	Tarnished & Rough
	16	Black & Brown, Rough	Very Rough	Very Rough Coating
	17	Black & Brown, Rough	Very Rough	Very Rough Coating
	18	North Side Dull, South Side 15% Area Rusted	North Side Dulled, South Side Rusted	South Side Rusted, North Side Some Rust

TABLE 5 LABORATORY INSPECTION RESULTS OF ONE YEAR OF EXPOSURE TO CORROSIVE GASES

SPECIMEN NUMBER	LOCATION A OUTSIDE #3 POWER PLANT	LOCATION B INSIDE #3 POWER PLANT
1	Heavy black corrosion incrustation. Metal etched and pitted.	Heavy incrustation of dark corrosion products. Metal etched.
2	Medium black corrosion. No pitting. Some silver plate still present.	Thin, uniform dark corrosion coating. No etching or pitting.
3	Uniform black corrosion. Some pitting.	Same as No. 2
4	Uniform black corrosion. No noticeable pitting.	Thin, uniform gray corrosion coating. No etching or pitting.
5	Cadmium plate all gone. Similar in appearance to #1.	Uniform yellow corrosion coating. No pitting or etching.
6	Uniform green and black corrosion. No pitting.	Uniform light green corrosion coating. No pitting or etching.
7	General rusting over all. Shallow pits.	Numerous scattered rust spots. Metal etched.
8	Thin green-gray corrosion over all.	Same as No. 6
9	Missing	Thin dark corrosion film. No pitting.
10	Scattered rust spots over all. Shallow pits.	No appreciable corrosion. Metal slightly dulled.
11	Moderate gray uncrustation. Uniformly etched.	Numerous pits scattered over surface.
12	Light corrosion with scattered shallow pits.	Practically perfect. One pit noted.
13	Badly corroded. Metal thinned by extreme rusting	Over all rust
14	Uniform black corrosion scale. No noticeable pitting	Thin uniform smooth black scale. No pitting
15	Heavy black incrustation. Metal etched but not pitted (not as good as nickel.)	Moderately heavy black incrustation. Metal etched.
16	Same as No. 1.	Same as No. 1.
17	Same as No. 1.	Slightly better than No. 1 or No. 16 but of the same order.
18	Scattered rust over all. Shallow pits.	No. attack. In excellent condition.

TABLE 6 LABORATORY INSPECTION RESULTS OF ONE YEAR OF EXPOSURE TO CORROSIVE GASES

SPECIMEN NUMBER	LOCATION C OUTSIDE CHILLING PUMP HOUSE	LOCATION D INSIDE CHILLING PUMP HOUSE
1	Heavy incrustation of dark corrosion products. Metal etched.	Black film overall. Somewhat etched.
2	Medium incrustation. No pitting. Some silver plate still present.	Uniform black coating. No etching or pitting.
3	Medium incrustation. Deep pits in basis metal.	Tarnished but otherwise O.K.
4	Fairly heavy incrustation. Shallow pits of basis metal.	No appreciable change. In excellent condition.
5	Medium incrustation. No cadmium plate left. No appreciable pitting.	Uniform yellow color. Otherwise O.K.
6	Moderate uniform corrosion products. No pitting.	Slightly tarnished. Otherwise in perfect condition.
7	Heavy rust overall. Metal etched with shallow pits.	Same as No. 4.
8	Thin gray surface film. No pitting or etching.	Same as No. 4.
9	Moderate uniform gray film over all. No pitting.	Tarnished and a few shallow pits.
10	Numerous scattered small rust spots which show shallow pits underneath.	Same as No. 4.
11	Moderate gray incrustation. Metal etched.	Same as No. 4.
12	Scattered corrosion pits.	Same as No. 4.
13	Very heavy rust.	Rust over all.
14	Thin uniform smooth corrosion scale (.0004" thick). No pitting.	Uniform black tarnish coating. No pitting or etching.
15	Moderately heavy corrosion products over all. Metal surface etched.	Tarnished but not appreciably etched or pitted.
16	Same as Copper (#1).	Same as #1.
17	Same as Copper (#1).	Similar to #1 & #16 but slightly better.
18	Missing	General rusting and pitting.

R. B. Mears.[3] We were very much interested in the paper by Mr. Maucher and Mr. Jones. Naturally, we were gratified by the good resistance to attack exhibited by the bare and anodically coated aluminum samples. These results are supported by a considerable amount of other data obtained both by our own laboratories and elsewhere. These other tests unite in revealing that many aluminum-base materials are highly resistant to outdoor exposure conditions, even in damp or seacoast locations. It is also well known that sulphur compounds do not attack any of the commercial aluminum-base alloys. Thus the high resistance of the aluminum-base alloys to atmospheric conditions coupled with their specific resistance to sulphur compounds indicates that oil-refinery people could well afford to give serious consideration to the use of aluminum alloys.

Data from extensive atmospheric exposure tests conducted by A.S.T.M. Committee B-3, Subcommittee VI, for periods up to ten years are now available. These tests were conducted at seven different outdoor exposure sites, including exposures at Altoona, Pa.; New York N. Y.; State College, Pa.; Phoenix, Arizona; Sandy Hook, N. J.; Key West, Fla.; and La Jolla, Calif.

Aluminum alloys 2S, 3S, and Alclad 17S-T proved highly resistant at all locations. Rather similar tests conducted by the Aluminum Research Laboratories indicate that some of the newer sheet alloys such as 52S, 53S, Alcald 3S, and Alclad 24S-T are also highly resistant in outdoor exposures.

Aluminum alloys have been used to some extent in roofs for oil tanks handling sour crudes. About 60 unpainted aluminum-alloy (3S) roofs have been used. In all cases it is understood that these roofs resisted both the outdoor atmospheric exposures and the moist air containing H_2S in the tank interiors very satisfactorily.

Because of the high resistance of aluminum alloys to sulphur compounds and to ammonia (or ammonium hydroxide), aluminum-alloy tubes can often be used advantageously in heat exchangers and condensers in refineries. For such applications, the use of Alclad 3S alloy tubing or of cathodic protection by zinc attachments is advantageous in minimizing the effects of corrosive cooling waters.

Aluminum wires are used for telephone lines in atmospheres containing hydrogen sulphide, because of their high resistance to attack by sulphur compounds.

Aluminum alloy (17S-T) nails are extensively employed for use in wooden structures in oil fields, again because of their resistance to sulphur compounds.

Large processing equipment for gas purification can often be made of aluminum alloys advantageously. Here the high resistance to both CO_2 and H_2S is valuable.

[3] Chief, Chemical Metallurgy Division, Aluminum Research Laboratories, Aluminum Company of America, New Kensington, Pa.

Authors' Closure

We appreciate Dr. Mears's discussion concerning the behavior of aluminum and its alloys under test and service conditions related to those which prevailed in this investigation. It is gratifying to note that the data obtained by the authors are in substantial agreement with Dr. Mears's extensive observations and experience.

An Investigation of Radial Rake Angles in Face Milling

By J. B. Armitage[1] and A. O. Schmidt,[2] Milwaukee, Wis.

The object of this investigation was to determine the effect of negative and positive radial rake angles in milling cutters upon the power required for the cutting action, the tool life of the cutter, the surface finish, and temperature of the workpiece. Tests were conducted with 2-in. face mills which had two brazed carbide tips. All cutters had the same dimensions except for the radial rake angles which included 30 deg, 15 deg, 6 deg positive; 0 deg, 6 deg, 12 deg, 20 deg, and 30 deg negative. The axial rake angle was 0 deg on all of the cutters. Cutting speeds were varied in seven steps between 106 and 785 fpm; different feeds were used with each speed and the depth of cut was constant at 0.125 in. Test bars were made of normalized S.A.E. 1055 hot-rolled stock. This steel was used throughout the investigation except for the tool-life tests in which S.A.E. 1020 was also used. Equipment and procedure used in the tests are discussed and the results illustrated by several graphs. Clearly defined conclusions are drawn from the investigation.

Discussion of Previous Investigations

Numerous articles recommending negative radial rake milling have appeared in technical publications and bulletins, most of them reporting on over-all performance. The following is stated in one such reference (1):[3] "While the cutting force with a 10-degree NEGATIVE rake is higher than that of a 10-degree positive rake cutter at low speeds, there is always a point at which they are equal, and if a sufficiently high speed is used, the force on the NEGATIVE rake cutter is less than on the positive rake cutter."

At the 1943 Annual Meeting of the Society, a paper was given by Hans Ernst (2), in which were shown several graphs of test data and diagrams with a single-point-tool dynamometer. The results indicated an increase of tool force at high speeds for a 10-deg positive radial rake angle and a decrease of tool force at high speeds for a 10-deg negative radial rake angle. It was stated: "Careful investigations over a wide range of speeds and with various work materials have shown that the power required with negative rake angles decreases steadily with increase in speed. In fact, it has been found in several cases that beyond a certain speed, the specific power consumption with negative rake angles is actually less than with positive rake angles."

At the same meeting, in a companion paper (3), Wallace E. Brainard showed that the tool forces do not vary with the cutting speed from 200 to 1500 fpm. In his investigation, he used an electrical strain gage to determine the tool forces.

For negative radial rake angles, 50 per cent increase in horsepower over present machines and cutting speeds from 375 to 1000 fpm are recommended by R. G. Owen (4). It is also stated that a feed of 0.0005 in. per tooth will dull a cutter after 3 to 5 min cutting time.

C. H. Bodner (5) suggests a heavy cutter body and finds negative blade angles of 10 deg for axial and radial rake the best, in spite of the fact that they require more power to drive the machine tool than positive angles. Climb-milling on machines without backlash and a feed of 0.006 in. per tooth are also recommended.

In a paper by Fred W. Lucht (6), which was presented at the 1943 Annual Meeting of the Society, the following information was given:

"As a rule, positive rake angles result in less power consumption and permit deeper cuts and heavier feeds than do negative rake angles, but the latter should be employed only when necessary to strengthen the cutting edge. Carbide blades 30 to 50 per cent thicker than those used for cast iron permit optimum results when milling steel. A minimum feed of 0.005 in. per tooth is also recommended for steel."

Procedure and Equipment for High-Speed Milling Tests

For this milling investigation a special setup had to be developed to measure the power at the cutting edge accurately at both low and high speeds. All tests were carried out on a Milwaukee 2-K vertical milling machine which had a wide range of feeds and speeds and permitted a maximum cutting speed of 785 fpm with a 2-in. face mill. For checking purposes, an Esterline-Angus recording wattmeter was connected with the milling-machine-drive motor. The surface temperature of the test specimen was measured immediately before and after cutting with an "Alnor" low-range thermocouple. Surface finish on the test bars was checked with a profilometer.

During this investigation, the power required by the tool was measured with a calorimeter from which the heat in the chips was determined. This method proved to be simple and reliable.[4] The measuring of tool forces has been generally effected by the use of mechanical, optical, and electrical means, while the temperature of the cutting edge during the cutting operation has been measured by thermoelectrical devices. In the reported test, distilled water is employed as a means of measuring the quantity of heat generated by a combination of friction and deformation during the cutting operation. Because of the effect which water may have in a milling operation, it was decided to cut the test bars dry and measure only the heat of the chips in the calorimeter. When work is transformed into heat or heat into work, a quantity of work is the mechanical equivalent of a quantity of heat.

Discussion and Interpretation of Results

The main advantage of the calorimetric method used in these tests is that it permits in a simple way the measurement of power at the cutting tool at any cutting speed. Prof. G. Schlesinger

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[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Production Engineering Division and presented at the Semi-Annual Meeting, Pittsburgh, Pa., June 19–22, 1944, of The American Society of Mechanical Engineers.

[4] For more detailed information see the paper, "Determining Tool Forces in High-Speed Milling by Thermoanalysis," by A. O. Schmidt, presented as a companion paper, at the Semi-Annual Meeting of the Society.

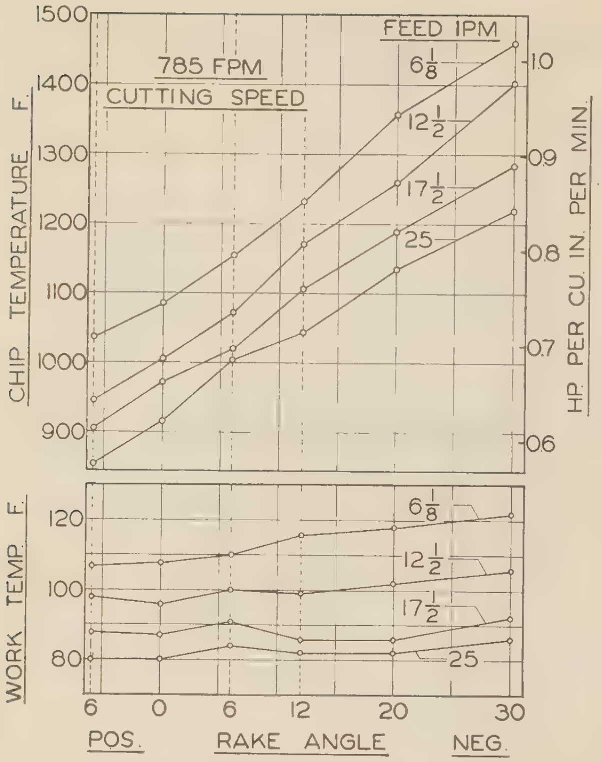

FIG. 1 GRAPHS SHOWING TEMPERATURE OF CHIPS, HORSEPOWER PER CU IN. PER MIN, AND SURFACE TEMPERATURE OF WORKPIECE

(Top: Temperature of chips and horsepower per cubic inch per minute in relation to radial rake angles at 785 fpm cutting speed and various feeds. Material cut is S.A.E. 1055, 1 in. diam, hot-rolled, and normalized. Depth of cut is 0.125 in. Bottom: Surface temperature of workpiece measured with thermocouple immediately after taking cuts plotted in upper graph. Room temperature is constant at 70 F.)

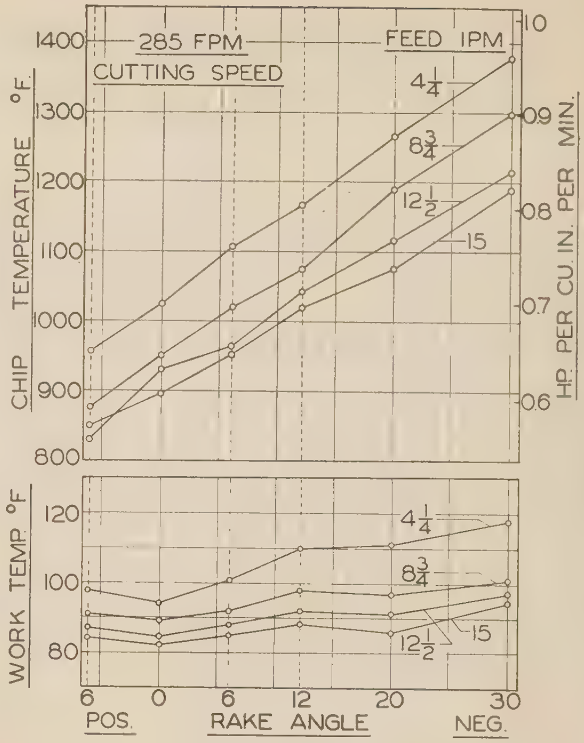

FIG. 2 GRAPHS SHOWING TEMPERATURE OF CHIPS, HORSEPOWER PER CU IN. PER MIN, AND SURFACE TEMPERATURE OF WORKPIECE

(Top: Temperature of chips and horsepower per cubic inch per minute in relation to radial rake angles at 285 fpm cutting speed and various feeds. Material cut is S.A.E. 1055, 1 in. diam, hot-rolled, and normalized. Depth of cut is 0.125 in. Bottom: Surface temperatures of workpiece measured with thermocouple immediately after taking cuts plotted in upper graph. Room temperature is constant at 70 F.)

(7) has stated that in customary shop tests in metal cutting, variations in the results up to 20 per cent may occur. There are inevitable differences in the elements of the test which are due to the following:

1 The efficiency of the motor which may vary from 5 to 8 per cent because of irregular heating due to load variations during the testing period.

2 The efficiency of the gears which may vary from 3 to 8 per cent because of difficulties in keeping the lubrication factors constant and to the heating caused by a change of load.

3 The variations of 5 to 10 per cent in the test material.

4 Variations in the heat-treatment of the tool, which may vary between 5 and 20 per cent in high-speed tools. (The authors have found that the same percentages will apply to brazed tungsten-carbide tips.)

Data plotted in the accompanying graphs constitute the average of at least three individual tests and are based upon the measurements taken with the calorimeter. This excludes errors due to motor and gear variations. Test bars were taken from the same piece of material and were normalized. A close check was maintained in order to eliminate those bars which would repeatedly give results more than 5 per cent above or below the average temperature value measured in the calorimeter for identical tests.

Care was taken to use carbide tips from the same batch and to have them carefully brazed and ground. Another source of error is the wear to which the carbide tip is exposed as soon as it begins to cut metal. During the power tests, tips were inspected after each cut on a test bar. Cutters were eliminated if they showed any sign of wear under the illuminated magnifying glass, and in most cases were used on three test bars removing 0.3 cu in. of metal and not on more than six test bars. During these tests, the cutters were considered to be sharp. A very close check on the cutting edge was also possible because a rise in the calorimeter readings and the surface-temperature readings would always indicate a hard spot in the test bar or wear and chipping of the cutting edge. In some cases this could also be noticed by a rise in the wattmeter reading and could be seen with the magnifying glass as wear on the cutting edge.

Changes in the power required by the two-lip milling cutter in relation to the rake angle were investigated first. The initial series of tests was conducted at a cutting speed of 427 fpm and varying feeds with the 6-deg positive radial rake and 0 deg, 6 deg, 12 deg, 20 deg, and 30-deg negative radial rake angles. A second series of more than 1000 tests was run under identical conditions except that the cutting speed was increased to 523 fpm.

In order to study the effect of the negative radial rake angles at lower speeds, tests were conducted at 285, 192, and 106 fpm cutting speeds. At these speeds the surface of the specimen would show conspicuous chatter marks for all negative radial rake angles larger than 6 deg. Considerable vibration was induced in the machine. At 106 fpm the surface of the specimen was also badly torn and cracked by the negative radial rake cutters.

Several series of tests were conducted at 638 and 785 fpm, and the results were found to agree substantially with those at the lower speeds. From the computed average temperature of the chips, it can be seen that the melting temperature of S.A.E. 1055 was not approached in any test. Observation indicates that

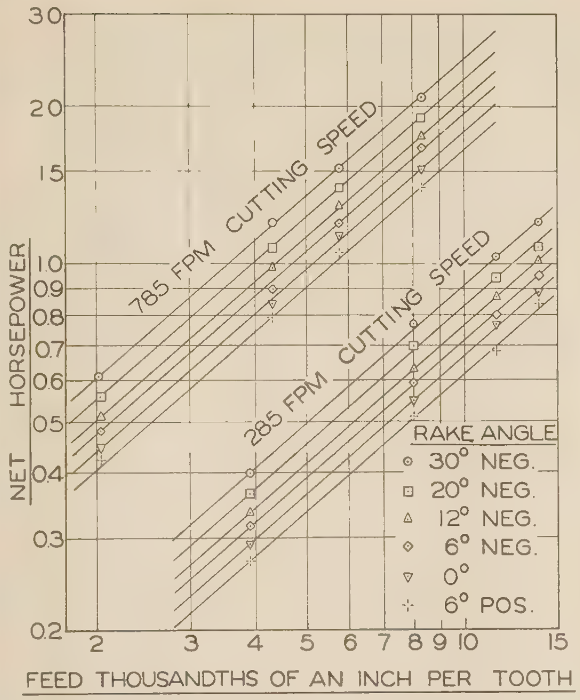

Fig. 3 Comparison of Net Horsepower Requirements of Different Radial Rake Cutters Operating at Constant Cutting Speeds, 0.125 In. Depth of Cut, and Various Feeds as Computed From Heat in the Chips

(Material, S.A.E. 1055; normalized; Bhn 205.)

the average temperature is the actual temperature of the chips only at fine feeds, while a heavier chip has a different temperature on each side, the higher one on the side away from the tool. This is indicated by the difference in color in individual chips.

When cutting tests are made at heavy feeds and at high speeds, chips and minute particles from the workpiece come off simultaneously. These small particles usually become so overheated, because of their smaller heat-carrying capacity, that they fly off as sparks, while the heavier chips are blue on the side away from the tool and still have the silvery steel color on the side which has just been separated from the workpiece.

From the temperature rise of the calorimeter the average temperature rise of chips, the horsepower per cubic inch per minute, and the net horsepower could be determined.

Values for average chip temperature and horsepower per cubic inch per minute thus arrived at are plotted in Figs. 1 and 2. It can be seen that the chip temperatures increase as the radial rake angle becomes more negative. Because these negative radial rake angles required more net horsepower, it follows that there was more heat generated during the cutting operation. The plotted temperatures further substantiate this statement. Chip temperatures were computed from the calorimeter readings for a constant specific heat. Actually, the specific heat of steel must be assumed to increase at high temperatures and the true chip-temperature values are lower than those plotted.

From the surface temperatures of the workpiece which were measured with the "Alnor" low-range thermocouple immediately before and after the cutting of the test bar, it can be seen that workpiece temperatures depend less upon the radial rake angle than upon the feed (see lower part of Figs. 1 and 2). The temperature of the workpiece and chips varies inversely as the feed. This general tendency was also found to be true in the tool-life tests when the same test bar was used continuously until the cutter had failed.

The net horsepower values, as computed from the calorimeter temperature rise, in relation to the feed per tooth as used in the tests are plotted with a logarithmic scale in Fig. 3, for a cutting speed of 785 and 285 fpm. This relationship between the values for different radial rake angles at other cutting speeds was found to remain substantially the same.

For a given radial rake angle, the increase in net horsepower with an increase in feed has a relationship which is always definite and can be determined from the logarithmic graphs as

$$\text{hp} = CnNf^a$$

in which

C = constant depending upon the radial rake angle
n = number of cutter revolutions per minute
N = number of teeth in cutter
f = feed in inches per tooth
a = exponent determined as 0.85 from values plotted on log-log paper

Net horsepower values computed by the foregoing formula were found to agree within close limits with actual test results. This was a further check that the tool forces were not reduced by negative radial rake angles or high cutting speed, but remained constant for a given chip cross section and changed with the radial rake angle, as shown in Figs. 12 and 13.

During these tests conducted at 427, 523, 638, and 785 fpm cutting speeds, various feeds, and a constant depth of cut, the following observations were made:

1 In taking the first six cuts with sharp cutters on a test bar the power requirement of the cutter with positive radial rake is less than that for cutters with negative radial rake angles.

2 After the negative and positive radial rake cutters have cut an equal amount of metal and all show wear on the cutting edge, the negative radial rake cutters will require less power.

3 In comparison with positive radial rake cutters, the cutter with negative radial rake angles will stand up longer and show less wear.

4 A cutter with negative radial rake angles is more effective at higher cutting speeds because of its inherent additional strength at the cutting lip.

5 Cutters with 20- and 30-deg negative radial rake angles left more pronounced chatter marks on the workpiece and frequently induced machine vibration.

6 Observations for tests run at 427, 523, 638, and 785 fpm agree substantially for tests run at 285 and 192 fpm cutting speed. However, surfaces showed conspicuous chatter marks for all negative radial rake angles above 6 deg even at low feeds.

7 At 106 fpm the surface of the specimen was badly torn and cracked by all negative radial rake cutters. In every specimen, the edge at which the cut was completed was ragged and chipped below the surface plane, indicating that the edge of the test bar had been broken rather than milled off. Induced vibration of the machine was very noticeable during these low-speed tests with the various negative radial rake cutters.

8 The surface temperature of the workpiece decreases with increasing feed rate and is proportional to the temperature of the chips.

The horsepower per cubic inch of metal removed per minute by the cutters with different radial rake angles at a cutting speed of 785 fpm is plotted in Fig. 1. The highest values were obtained with the 30-deg negative radial rake cutter and the lowest with the 6-deg positive radial rake cutter. In spite of the fact that data points are somewhat scattered due to the inevitable test variations, it can be seen that there is a tendency of the horse-

power per cubic inch per minute to decrease with increasing feed. This really means that the work required to remove a cubic inch of metal is less with a coarse feed than with a fine feed. One good reason for the better power economy at coarser feeds is the fact that a coarse feed F, which is twice the fine feed f, is separating a smaller suface in relation to the volume removed. As can be seen in the simplified diagram, Fig. 4, the fine feed will separate a sur-

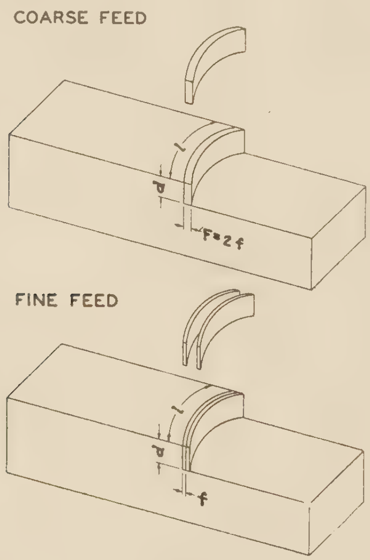

Fig. 4 Relationship Between One Coarse Chip and Two Fine Chips in Removing Same Amount of Metal

face area of $2\,(f + d) \times l$ which is $(d \times l)$ larger than the surface area of separation for the coarse feed which is $(2f + d) \times l$. But in each case the same volume of metal will be removed and a coarse feed will result in more economical power consumption.

The advantage in power economy gained with heavier feed is illustrated in Fig. 5, in which the feed in inches per tooth is plotted on the abscissa, 1 f being the feed which is so fine that there is only minimum deformation in the workpiece and chip. For this reason the cutting force is assumed to be directly proportional to the area of the surfaces of separation and to the volume of metal removed. On the right-hand ordinate are given values for the cutting force per chip and also area of the surfaces which are separated when cutting a particular chip. Volume of the chip is indicated on the left-hand ordinate. If it were possible to separate the metal chips from the workpiece in one plane only, e.g., without disturbing the adjacent layers in the metal, the increase in the cutting forces with increase in feed per tooth would be very small and would coincide with the increase in the area of the surfaces of separation. The volume of metal removed will increase more rapidly as can be seen from the upper line.

However, besides separation of the metal planes, compression, bending, friction, and other phenomena in a metal-cutting operation which can be observed as deformation also occur. Since this deformation in the chip and in adjacent layers of the workpiece is proportional to the feed, the tool forces become greater with an increase in feed.

When the volume of metal being cut is removed in fine chips of 1 f thickness, the force line would coincide with the volume line because of the large proportional increase in the areas of separation.

The ordinate b-c between the first and second curve can be considered as representing the increase in the cutting force due to deformation of the heavier chips. If the force of separation were the only force necessary to cut metal, then the cutting force would be directly proportional to the surface areas of separation, and the increase in cutting force would equal a-b. It is evident that one thick chip is more economical than several fine chips in removing an equal volume of metal from the difference in the ordinate of the same chip for the force and volume curve. The volume curve would indicate the sum of the forces necessary to remove 10 fine chips of 1 f thickness, while the force curve represents the actual force necessary to remove one thick chip of 10 f thickness. The difference, c-d, in the height of the ordinate equals the decrease in force in the latter case.

For this particular feed of 10 f, the increase in volume of the metal removed is 1000 per cent in comparison to a feed of 1 f, while the increase of the cutting force is only 720 per cent. If there were no deformation in the chip and workpiece, the cutting force would have increased only slightly less than 10 per cent be-

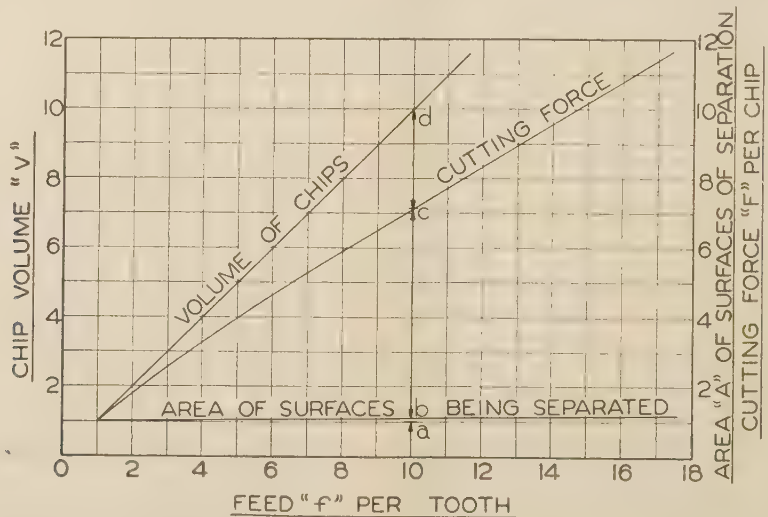

Fig. 5 Relationship of Volume of Chips Removed, Area of Surfaces of Separation, and Cutting Force to Feed per Tooth of Cutter

cause it would have equalled the small increase in the area of surfaces being separated. Therefore, this one chip of 10 *f* required almost 7 times as much force due to deformation as was required for its separation.

But no matter how fine a chip is cut, there is always more work done in deforming the metal than in separating it. As just pointed out, separation of metal is only a small part of metal-cutting. This can be seen from the appearance of heavier steel chips, which will often retain the steel color in the cutting plane and be heavily oxidized on the side away from it, indicating that more heat was generated through deformation than cutting of the chip.

From the heat values of the calorimeter, the average temperatures of the various chips were computed. Increasing the chip thickness will lower the average temperature because, with increasing feed, the volume of the chips increases more rapidly than the cutting forces. Therefore, the average temperature of the chips will vary inversely as the chip thickness.

The graph in Fig. 5 has been plotted with values of depth of cut and feed corresponding to the ones used in the tests, as have the force values. If another ratio of depth of cut to feed were to be used, the line for the areas of separation would be steeper for a smaller ratio and have still less slope if the ratio between depth of cut and feed were larger.

Comparative Tool-Life Tests

During the tests for power consumption, it was found that the positive radial rake cutters would frequently fail after the first cut on the test bar. For this reason numerous comparative tool-life tests were made following the procedure subsequently described. Since the cutters with 20- and 30-deg negative radial rake angle induced too much vibration, very few tool-life tests were run with them. Cutters with 15- and 30-deg positive radial rake were excluded from the tool-life tests because they failed too easily.

Three readings on test bars of S.A.E. 1055 (the same as used in the tests for power consumption) were taken first. Then 16 cuts which removed 28 cu in. of metal were made on S.A.E. 1020. Following this, three test bars were again cut for comparison. Depth of cut was 0.125 in. for all tests; the feed was constant at 15 ipm with a cutting speed of 523 fpm.

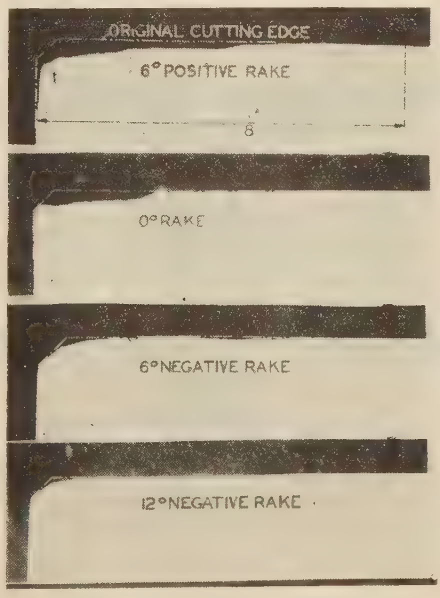

Fig. 6 Silhouettes of Cutting Edges of Various Cutters in Plane Parallel to Axis of Rotation Taken on a Jones and Lamson Optical Comparator

(Each cutter has removed 28 cu in. of S.A.E. 1020 cold-rolled steel and 0.59 cu in. of S.A.E. 1055 at a cutting speed of 523 fpm, a feed of 0.0075 in. per tooth, and 0.125 in. depth of cut.)

Fig. 7 Two-Bladed Face-Milling Cutter Which Was Used in Tests and Was Provided With Blades of Various Radial Rake Angles

The following observations and measurements are typical and representative of the results in the comparative tool-life tests:

1 After the same amount of metal had been cut with each cutter the increase in power consumption was as follows:

6-deg positive radial rake cutter......34 per cent increase in power
0-deg radial rake cutter..............24 per cent increase in power
6-deg negative radial rake cutter.......16 per cent increase in power
12-deg negative radial rake cutter.......12 per cent increase in power

2 The wear on the cutting edge of the 6-deg positive, 0-deg and 6-deg negative radial rake cutters was due to minute chipping of the carbide tips. On the cutting edge of the 12-deg negative radial rake cutter only fine abrasion was visible, see Fig. 6.

3 Average profilometer readings in microinches on the surface of the test bars at the beginning and end of test were as follows:

Cutter	Reading at beginning	Reading at end
6-deg positive radial rake	140	53
0-deg radial rake	23	26
6-deg negative radial rake	45	80
12-deg negative radial rake	22	80

4 The average surface temperature rise of the specimen would increase between 200 and 400 per cent for the last cuts taken with the partly failed and worn cutter.

Comparative Tool-Life Tests of Cutters Provided With Double Radial Rake Angle Tips

It was established without doubt by numerous tests at cutting speeds ranging from 100 to 800 fpm that the tool forces are higher

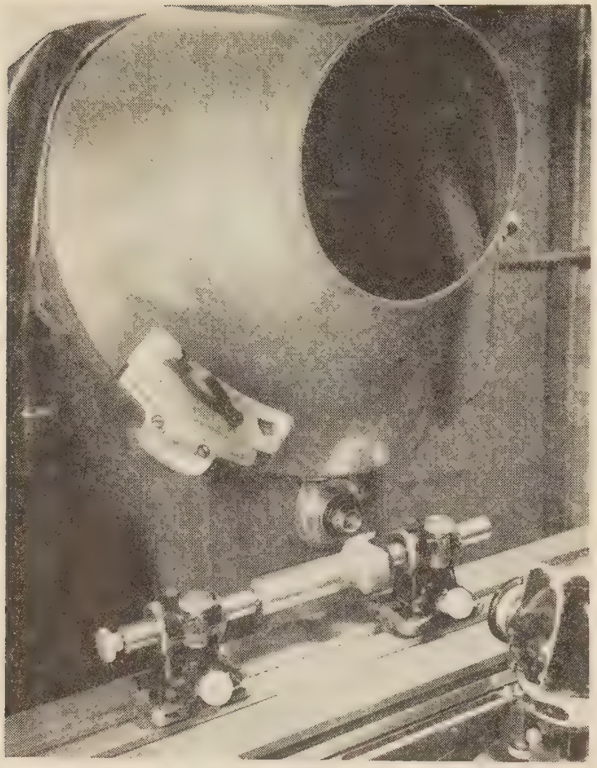

Fig. 8 Jones and Lamson Optical Comparator Used for Investigation and Recording of Tip Failure

for cutters with negative radial rake angles than for positive radial rake angles. However, the cutters with negative radial rake angles will stand up much better and their power requirements will increase more slowly than those with positive radial rake angles. This increase in power consumption during the cutting operation is due to the wear to which any cutter is exposed while in operation. Only because of the more rapid wear may it actually happen that a positive radial rake cutter will entail higher cutting forces than one with negative radial rake angles.

From Fig. 5, it is evident that the cutting forces are invariably several times greater due to deformation of the metal than to separation in the cutting plane. This deformation is always present no matter which angles are provided on the cutter and will be more evident with negative radial rake angles than with positive radial rake angles. A very sharp tool would induce little deformation in the chip or workpiece but tool life would be very low if one would go to the extreme and use a metal-cutting tool similar to a razor blade.

There are therefore two major requirements for a cutter which will give satisfactory cutting action and tool life:

1 A cutting edge which will stand up and not wear or chip readily.

2 A tool shape which will reduce the deformation in the chips and workpiece.

As the cutting speed and hardness of the workpiece material increase, these two requirements assume greater importance. The first requisite can be satisfied with negative radial rake angles and carbides on the cutting tips; the second is achieved by simultaneously providing a positive radial rake angle for improving chip flow.

In accordance with the foregoing conclusions, it was found that cutters which had positive secondary radial rake angles with negative primary radial rake angles imposed at the cutting edge gave better tool life and required less power than cutters with negative radial rake angles. In the text the term "primary" radial rake refers to the radial rake angle at the cutting edge. "Secondary" radial rake refers to the radial rake angle behind the primary radial rake angle (see Fig. 9, *A* and *C*.)

Similar designs for rake angles on single-point tools have been used by DeLeeuw, E. G. Herbert, and Klopstock and are discussed in their application to high-speed tools by O. W. Boston (8). The use of negative and positive rake angles on carbide lathe tools for the machining of steel has recently been reported by Einar Almdale (9).

For the experimental investigation of the double radial rake angles, the first two cutters tested had been given a negative primary radial rake angle of 12-deg, 0.005 in. wide at the cutting edge of a 15-deg positive secondary radial rake tooth. This cutter was tested at 523 fpm cutting speed and compared with cutters of 6-deg positive radial rake angle and 12-deg negative radial

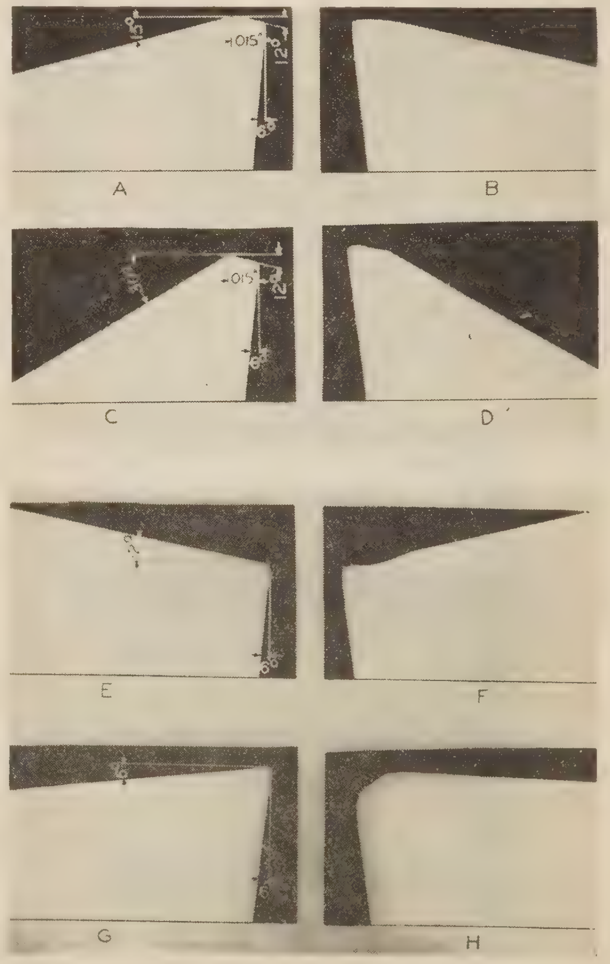

Fig. 9 Cross-Sectional Silhouettes of Carbide Cutting Tips in a Plane Normal to Axis of Rotation Before and After They Had Removed Equal Amounts of Metal

(Double radial rake cutter blades are shown in *A*, *B*, *C*, and *D*. The 12-deg negative radial rake angle is the primary radial rake angle and the 15- or 30-deg positive radial rake angle behind it is the secondary radial rake angle. Negative and positive radial rake angles are shown in *E*, *F*, *G*, and *H*. Illustrations at left show the angles of a sharp cutter blade, while those at right show type of wear which was most frequently encountered in tips provided with the respective radial rake angles.)

rake angle. The feed per tooth in this test was 0.0075 in. and it was observed that the cutter with the double radial rake angles used less power than the cutter of 6-deg positive radial rake angle. It was also standing up better, but not as well as the cutter with 12-deg negative radial rake angle. After the cutter had removed a comparatively small amount of metal, a crater started to form behind the cutting edge and eventually caused the edge to fracture.

The next comparative test was made with the same cutters except that the negative primary radial rake angle of 12 deg on the double radial rake cutter was increased in width to 0.007 in. The result was practically the same as in the first series of tests.

A third series of tests was conducted with identical cutters. The ones with double radial rake were given a negative primary radial rake of 12 deg, 0.020 in. wide, and the feed per tooth was 0.0105 in. In these tests, the double radial rake cutter required approximately the same power as the 6-deg positive radial rake cutter at the beginning of the test. Later, after it had machined the same amount of metal as the others, its power requirements were less than either of the other cutters. At this point the 6-deg positive radial rake cutter had failed completely and the 12-deg negative radial rake cutter had worn more than the double radial rake cutter. A comparison of the actual power consumption shows that after the double radial rake cutter had machined 16.8 cu in. of metal its power requirement was equal to that of the 12-deg negative radial rake cutter at the beginning of the test. Workpiece temperature was also lower than for the other cutters.

After these tests a large number of cutters with 30-deg positive secondary radial rake angles provided with a negative primary radial rake angle were tested. The positive secondary radial rake angle of 30-deg behind the negative primary radial rake angle of 12-deg brought about an additional decrease in tool force, power consumption, and tool wear, Figs. 9, 12, and 13.

The most obvious indication of improved cutting action was the shape and color of the chips. Those made by a double radial rake angle cutter showed little deformation, Fig. 14. They were almost flat (cutting S.A.E. 1020) and the color ranged from straw for the 15-deg positive secondary radial rake cutter with 12-deg negative primary radial rake to white for the 30-deg positive secondary radial rake cutter with 12-deg negative primary radial rake, Fig. 16.

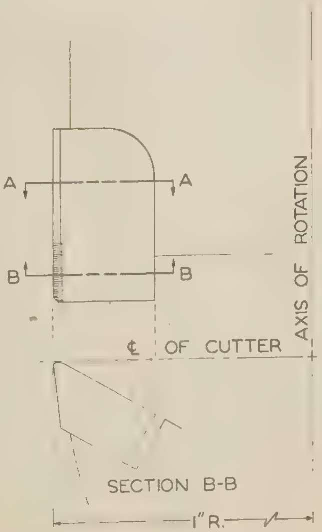

Fig. 10 Diagram Indicating Location of Cross Sections Shown in Fig. 9

(Section *A-A* is new blade as shown on the left side in Fig. 9, while section *B-B* is worn portion shown on right side.)

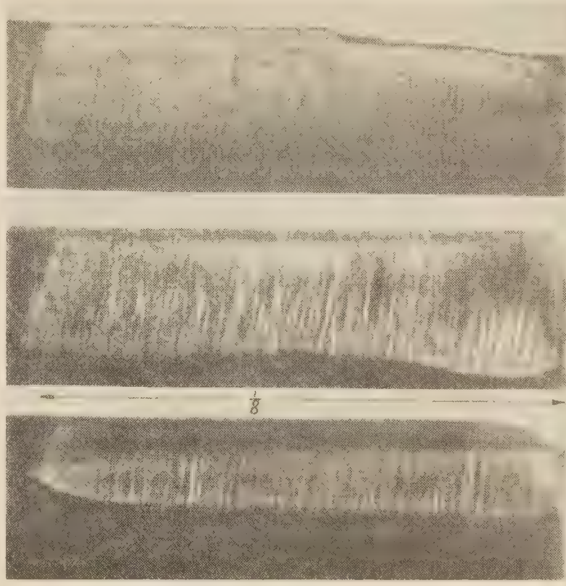

Fig. 11 Types of Wear and Failure on Different Cutting Tips Made of Tungsten Carbide

(Top: Positive and 0-deg radial rake angles. Center: Negative radial rake angles. Bottom: Double radial rake angles. All these tips had been working equal amounts of metal.)

Surface Finish and Temperature of Workpiece

In order to obtain a measure of the surface finish of the workpiece produced by milling cutters with various radial rake angles, profilometer readings were taken of the finish of each S.A.E. 1055 test bar or every third one in the tests which were repeated more than three times. In either case the first test bar cut with a new cutter and the last one of the test or series of tests were always checked for surface finish with the profilometer.

After comparing several thousand readings of surface finish produced at different speeds and feeds, the following statements can be made:

The best surface finish of 8 to 50 microinches is attained at cutting speeds from 500 to 800 fpm and a small feed of 0.0005 to 0.001 in. per tooth. At these feeds no difference could be established between the surfaces produced by the different cutters with various radial rake angles. However, when the feed was increased from 0.003 to 0.012 in. per tooth, the negative radial rake cutters gave a better finish than the cutters with positive radial rake. The best finish at these speeds was produced by the 12-deg negative radial rake cutter and had profilometer readings from 20 to 70 μ in. while the 6-deg positive radial rake cutter produced a finish from 30 to 145 μ in.

For a series of tests made at 427 fpm, the cutter with the 12-deg negative radial rake angles produced the best surface finish.

At speeds of 285, 192, and 106 fpm, with feeds from 0.0036 to 0.0130 in. per tooth, the surface finish was a little better with the 6-deg positive radial rake cutter than with the negative radial rake cutters. All the surface finishes were very poor, ranging from 90 to 245 μ in. for the 6-deg positive radial rake cutter to 170 to 290 μ in. for the 12-deg negative radial rake cutter.

It was found that the 6-deg positive radial rake cutter would produce a better surface at tne higher cutting speeds after the cutting edges had become slightly worn.

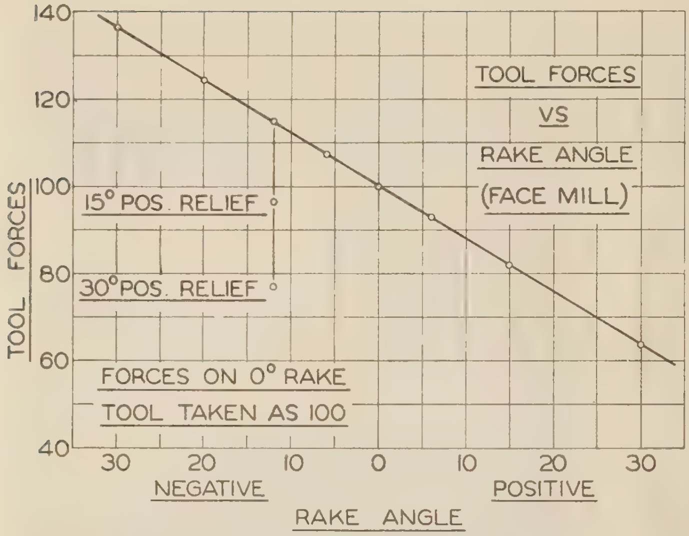

Fig. 12 Effect of Various Radial Rake Angles on Tool Forces Acting on a Face-Milling Cutter

(Points indicated as 15- and 30-deg positive relief represent cutters which had a negative primary radial rake angle of 12 deg, and a positive secondary rake angle of 15 or 30 deg.)

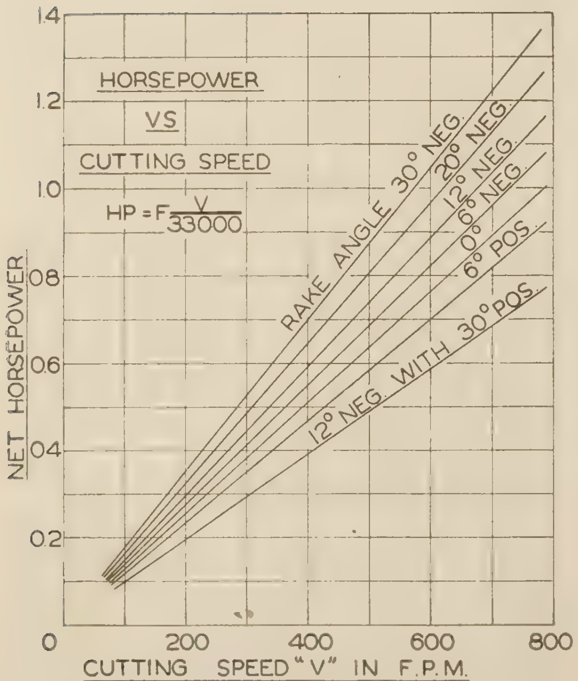

Fig. 13 Variation of Net Power Requirements With Cutting Speed for Various Radial Rake Angles

(Line designated 12-deg neg. with 30-deg pos. represents the cutter with 12-deg negative primary radial rake angle and a 30-deg positive secondary radial rake angle.)

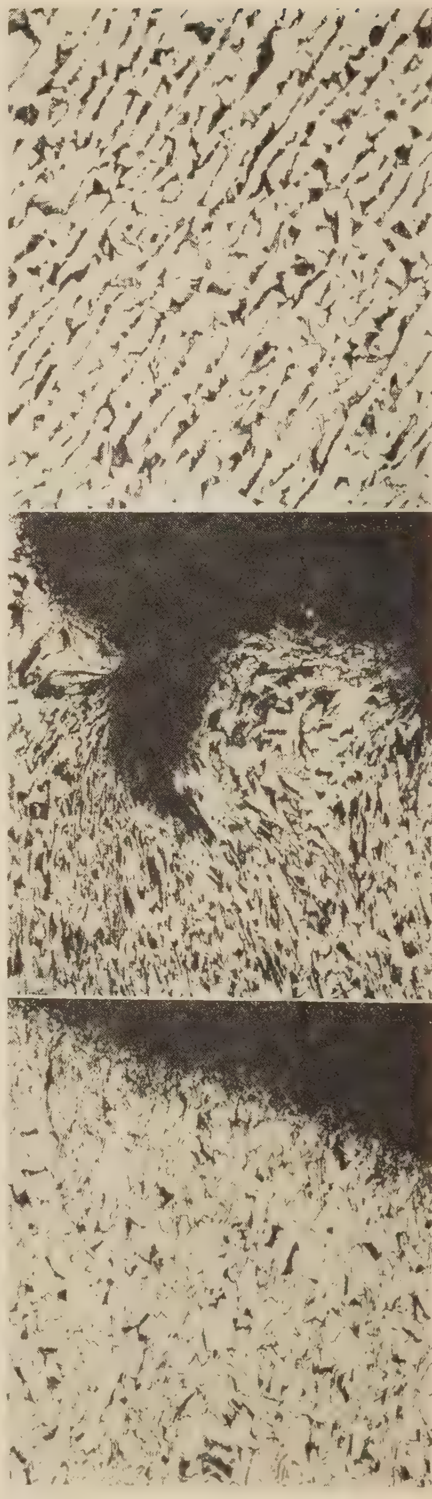

Fig. 14 Photomicrographs of S.A.E. 1020 Cold-Rolled Steel Used in Tool-Life Tests; ×100

(Top: Microstructure of parent metal. Center: Microstructure of badly deformed chips as produced by a 12-deg negative radial rake angle. Bottom: Microstructure of chip produced by 12-deg negative primary radial rake angle and 30-deg positive secondary radial rake angle. Very little deformation is visible.)

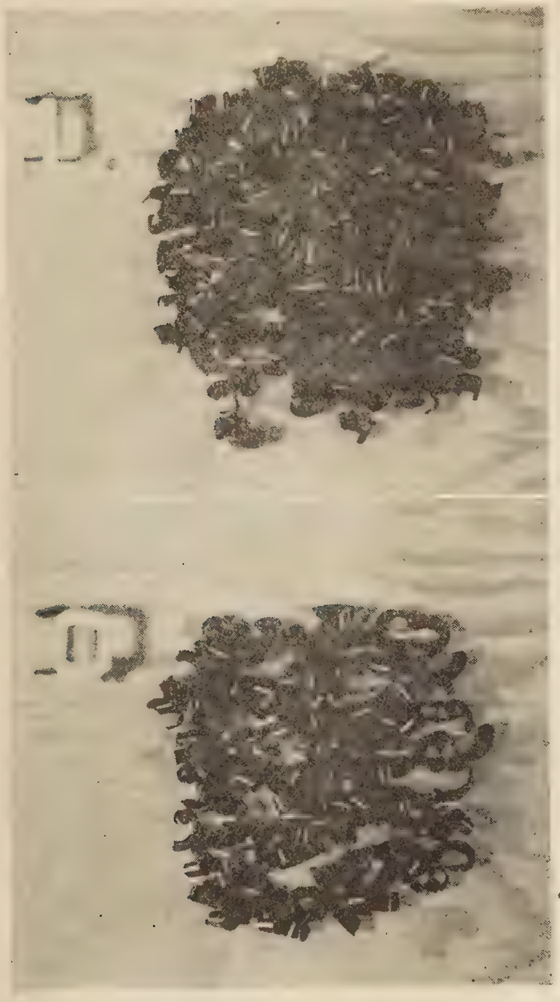

Fig. 15 Chips From S.A.E. 1020 Produced at 523 Fpm Cutting Speed With Feed of 0.0105 In. per Tooth

(Top: By cutter with 12-deg negative radial rake angle. Bottom: By cutter with 6-deg positive radial rake angle.)

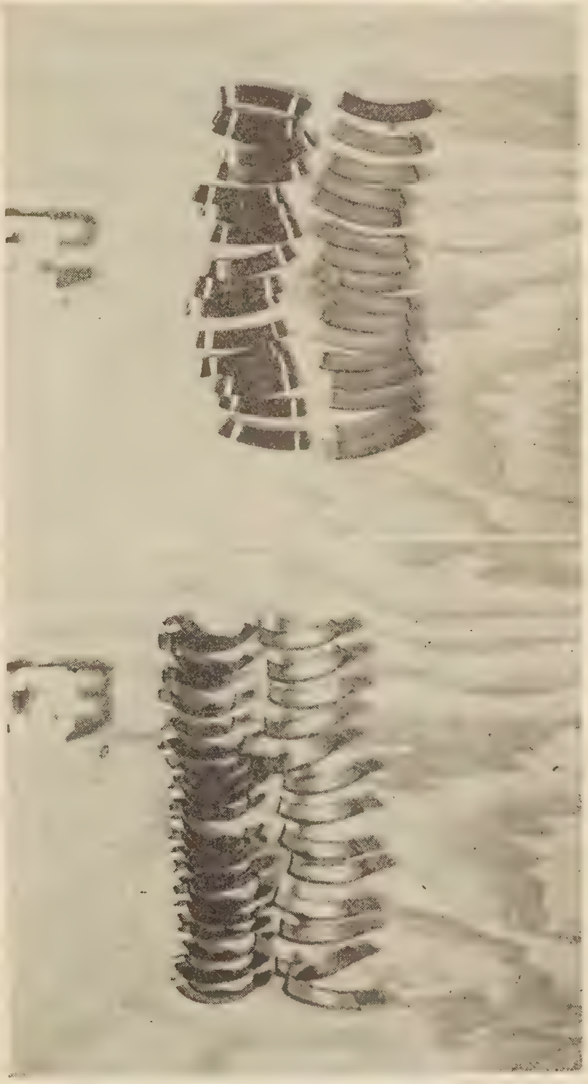

Fig. 16 Chips From S.A.E. 1020 Produced at 523 Fpm Cutting Speed With Feed of 0.0105 In. per Tooth

(Top: By cutter with 12-deg negative primary radial rake angle and 15-deg positive secondary radial rake angle. Bottom: By cutter with 12-deg negative primary radial rake angle and 30-deg positive secondary radial rake angle.)

Fig. 17 Cutter With 12 Blades Provided With 30-Deg Positive Secondary Radial Rake and Negative Primary Radial Rake of 6 Deg

The surface temperatures of the test bars were measured immediately before and after the cut. It can generally be stated that temperature of the workpiece is influenced by the radial rake angles of the cutter but is more dependent upon the cutting speed and feed. Surface temperature of the workpiece will vary directly as the cutting speed and inversely as the feed, see lower part of Figs. 1 and 2. The cutter which cuts with a smaller force, e.g., the 6-deg positive radial rake angle, will also entail a lower surface temperature as long as its cutting edge is sharp. As soon as its cutting edge is slightly worn, the surface temperature of the workpiece will increase beyond the temperatures generated by cutters with negative radial rake angles which have removed an equal amount of material. A cutter with negative radial rake angles, because of better wearing qualities, will generally give a somewhat lower surface temperature in workpieces than a positive radial rake cutter.

The foregoing statements apply only in those cases in which

Fig. 18 Different Types of Chips, as Produced With Cutter in Fig. 16, When Cutting a 4-In. Test Bar of S.A.E. 1020

(Top: Washerlike chips cut at a feed of 30 ipm. Center: Chips cut at 30 ipm by same cutter after a corner angle of 20 deg had been added. Bottom: Chips from same cutter, 20-deg corner angle, operating at 60 ipm feed.)

the chips will not be deposited upon the workpiece and will not transfer heat to the workpiece after their separation from the stock.

Practical Application of Double Radial Rake Angle

Performance of the double radial rake angle cutters with the experimental setup indicated a definite superiority over cutters with a single positive or negative radial rake angle. To check the action of double radial rake angles in a production setup, an 8-in-diam cutter, Fig. 17, was designed for the machining of mild steel. This cutter was made for comparison with existing cutters with negative radial rake angles and therefore was provided with 12 teeth. Blades made of solid carbide were ground and lapped so they could be held by wedges in ground slots in the cutter body. No axial rake angle or corner angle was provided. The positive secondary radial rake angle of the blades was 30 deg, and a negative primary radial rake of 6 deg, 0.015 in. wide, was ground on the cutting edge.

This cutter was operated at a cutting speed of 465 fpm and a feed of 30 ipm, which resulted in a maximum chip thickess of 0.0113 in. A $4 \times 4 \times 12$-in. block of S.A.E. 1020 cold-rolled steel with a Brinell hardness of 160 was used as the workpiece. Flat circular chips similar to spring washers and almost white in color were cut, see top of Fig. 18. After the cutter had machined approximately 300 cu in. of metal, signs of wear were detectable chiefly from the tendency of chips to assume a bronze tint. Interference of chips between the cutting blade and workpiece led to chipping of the cutting edge.

To increase cutter strength, a corner angle of 20 deg was ground on the blades. Chip clearance was improved by setting the blades out farther. The cutting edge was given a negative primary radial rake of 6 deg, but the width was increased to 0.025 in. This cutter was operated at various cutting speeds with feeds ranging from 35 to 60 ipm. Consequently, the maximum chip thickness was varied between 0.019 and 0.023 in. per tooth. Depth of cut was held constant at 0.125 in.

Chips cut with the second cutter were flat, semicircular, and almost white in color, see center of Fig. 18. After signs of wear began to appear on the cutting edge, the chips began to show a bronze coloring. One tip of the cutter was slightly damaged after operating for two hours by a chip which was carried around and caught between the blade and workpiece.

As can be seen from the chips in the lower part of Fig. 18, the chief disadvantage in this cutter design was inadequate chip clearance which allowed the chips to hit the cutter body and curl sharply at one end when the feed was 60 ipm.

However, these tests were sufficiently conclusive to prove that the double radial rake angle cutters require less power than the cutter with negative radial rake only and still provide a maximum amount of strength in the cutting edge.

The main advantage of the double radial rake on a face milling cutter will be that one cutter of basic design, e.g., 30-deg positive radial rake, can be used for mild-steel and nonferrous metals by grinding a primary negative radial rake on the cutting edge in the case of steels or a positive primary radial rake in the case of nonferrous metals. A cutter with a 15-deg positive secondary radial rake and a negative primary radial rake will be a stronger cutter and could be used in machining steels with a hardness in excess of 200 Bhn.

Regrinding of the double radial rake cutter will not require the removal of as much of the carbide tip as either a positive or negative rake cutter.

Conclusions

Negative radial rake angles were found to produce stronger

cutting tips, the cutting edges of which are generally not as liable to fail from impact and shock when entering a workpiece of hard material as are the cutting edges formed by positive radial rake angles.

Power required at the cutting edge is higher for the negative radial rake cutter than for cutters with positive radial rake angles. This holds true for conventional cutting speeds as well as higher cutting speeds up to 785 fpm.

Cutters with negative radial rake angles will stand up longer at the higher speeds than positive radial rake cutters under identical conditions.

Wear and failure at high speeds on the cutting edge of a positive radial rake angle cutter will soon increase its power consumption above that of a cutter with negative radial rake angles.

Best surface finish in high-speed milling is produced with a negative radial rake cutter at cutting speeds between 500 and 800 fpm. A feed of 0.001 in. per tooth or less will give the smoothest surface but will also result in more rapid abrasion at the cutting edge than with coarser feeds.

At 192 and 106 fpm cutting speed, the surface of the test specimens was badly torn and cracked by the negative radial rake cutters. This did not happen at higher cutting speeds.

Average temperature of the chips produced by ordinary feed does not approach the melting temperature of steel even at high speeds.

Heat in the workpiece and chips is due more to deformation of the metal than to separation in the cutting planes.

When removing a chip there is always more work done in deforming the metal than in separating it. The various elements like compression, bending, work-hardening, shearing, and friction enter into what is commonly called "metal-cutting."

A cutter with negative radial rake angles will generally give a somewhat lower surface temperature in the workpiece because it does not wear as rapidly as the positive radial rake cutter. However, as long as the positive radial rake cutter has a sharp cutting edge it will maintain lower temperatures in the workpiece.

A cutter with a 15- or 30-deg positive secondary radial rake angle and provided at the cutting edge with a negative primary radial rake angle 1 to 2 times the width of feed per tooth was found to be a more effective cutter, since it combined the increased strength of the cutting edge afforded by negative radial rake angles and the lower power requirement of the cutter with positive radial rake angles.

When running tool-life tests, it was found that the tool life of the carbide tips may vary as much as 50 per cent for the same cutters under identical conditions. This was due to variations of the quality between batches of carbide inserts of the same brand. Since tungsten from different sources is used by the carbide makers, this may be one of the most likely causes for the variation in cutter performance.

From the net horsepower at the cutter, as determined by the calorimeter, the tool forces were computed for the different rake angles by the formula

$$\text{hp} = \frac{F \times V}{33{,}000}$$

$$F = \frac{\text{hp} \times 33{,}000}{V}$$

This force F is the resultant force of several component forces, and it can be seen in Fig. 12 that it increases with the negative radial rake angle and decreases with the positive radial rake angle. It also shows how much the tool forces on a cutter with 12-deg negative primary radial rake are reduced by the application of a positive secondary radial rake angle of 15 or 30 deg.

The graph in Fig. 13 illustrates that the horespower is a straight-line relationship to the cutting speed when depth of cut and feed are constant. The force F has a particular value for each radial rake angle and chip cross section. This value does not vary with cutting speed.

ACKNOWLEDGMENT

Mr. J. R. Roubik of the Department of Engineering Development, Kearney & Trecker Corporation, assisted in carrying out and evaluating the reported tests. Grateful acknowledgment is also made to J. P. Bunce for his help in checking the manuscript.

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Effect of Grain Size and Subzero Treatment on Productivity of Four High-Speed Steels

By S. M. Depoy,[1] Dayton, Ohio

In the investigation reported, the effect of grain size and subzero treatment on the productivity of four high-speed steels was studied by means of a series of turning tests on one alloy steel at a preselected uniform hardness. The material being cut, the shape of the tool bit, and the speed, feed, and depth of cut on the turning operation were held constant. Standard and subzero treatments were used at different hardening temperatures to develop different grain sizes and different hardening products. The results obtained show that the grain size, carbide solution, and type of martensite formed in the tool have a very marked effect on its cutting ability. It appears that subzero treatment is much more effective, when large grain sizes are developed.

The purpose of the research was as follows:

1 To determine the difference in cutting ability between four types of high-speed steel.

2 To determine the difference in cutting ability which may exist between standard-treated cutting tools and those that are subzero-treated between subsequent tempers.

3 To determine the effect of ultimate grain size on the cutting ability of cutting tools.

The equipment and materials used in the research were as follows:

1 Model C Monarch toolroom lathe, 14 in.

2 Battery of Ajax electric salt-bath furnaces.

3 Subzero unit; simple container filled with dry ice and methanol.

The materials utilized consisted of one alloy machinery steel, N.E. 8620, and four types of high-speed tool steel. The source, heat number and chemistry of these materials are given in Table 1.

TABLE 1 SOURCE, HEAT NUMBER, AND CHEMISTRY OF STEELS USED IN TESTS

Spec. no.[a]	Trade name	Source	Heat no.	Chemical composition, per cent: C	Mn	Si	P	S	Cr	W	Mo	Va	Co	Ni
NE-8620	NE-8620	Timken Steel & Tube Div.	...	0.20	1.03	0.29	0.014	0.016	0.49	...	0.17	...	...	..
M-2	Electrite Double Six	Latrobe Electric Steel Co.	20392	0.85	0.29	0.21	0.028	0.012	4.12	5.78	4.57	1.59	..	..
M-36	Electrite CO-6	Latrobe Electric Steel Co.	44038	0.86	0.31	0.33	0.022	0.014	4.07	5.39	4.41	1.65	8.63	..
T-1	Electrite no. 1	Latrobe Electric Steel Co.	21003	0.73	0.34	0.32	0.029	0.009	4.07	17.76	...	1.07	...	..
T-5	Electrite Super Cobalt	Latrobe Electric Steel Co.	44041	0.85	0.29	0.25	0.027	0.012	4.30	18.03	0.78	1.94	8.56	..

[a] Specification for high-speed steels in this column were recently adopted by three large automotive manufacturers.

Procedure Followed in Tests

All high-speed steel bits ($^3/_8$ in. square × 3 in. long) were hardened in the Ajax salt-bath furnaces using Houghton liquid heat N.D. salt and Houghton liquid heat 2400 C salt in the preheat and high-heat furnaces, respectively. Park No. 100 salt was used in the quenching bath. The following are the treatments used:

Treatment No. 1. Preheat at 1650 F for 30 min; high heat at 2175 F for $2^1/_2$ min; quench at 1100 F for 3 min; cool in air to 100–150 F; wash thoroughly (hot water not less than 150 F); double temper at 1050 F, 2 hr + 2 hr.

Note: Care was taken that the bits cooled to room temperature between the temper cycles on all treatments.

Treatment No. 2. Preheat at 1650 F for 30 min; high heat at 2225 F for $2^1/_2$ min; quench at 1100 F for 3 min; cool in air to 100–150 F; wash thoroughly; double temper at 1050 F, 2 hr + 2 hr.

Treatment No. 3. Preheat at 1650 F for 30 min; high heat at 2275 F for $2^1/_2$ min; quench at 1100 F for 3 min; cool in air to 100–150 F; wash thoroughly; double temper at 1050 F, 2 hr + 2 hr.

Treatment No. 4. Preheat at 1650 F for 30 min; high heat at 2300 F for $2^1/_2$ min; quench at 1100 F for 3 min; cool in air to 100–150 F; wash thoroughly; double temper at 1050 F, 2 hr + 2 hr.

Treatment No. 5. Preheat at 1650 F for 30 min; high heat at 2325 F for $2^1/_2$ min; quench at 1100 F for 3 min; cool in air to 100–150 F; wash thoroughly; double temper at 1050 F, 2 hr + 2 hr.

Treatment No. 6. Preheat at 1650 F for 30 min; high heat at 2350 F for $2^1/_2$ min; quench at 1100 F for 3 min; cool in air to 100–150 F; wash thoroughly; double temper at 1050 F, 2 hr + 2 hr.

Treatment No. 7. Preheat at 1650 F for 30 min; high heat at 2410 F for $2^1/_2$ min; quench at 1100 F for 3 min; cool in air to 100–150 F; wash thoroughly; double temper at 1050 F, 2 hr + 2 hr.

A duplicate of each standard bit was made, and these duplicates were treated at 105 F below zero (—105 F). The bits were subjected to the subzero treatment between the two tempers. The refrigeration cycle was 3 hr. An Izod impact specimen was treated with each bit, and the relative impact values determined. Microsections were made on each bit, and the grain count was

TABLE 2 HARDNESS, IMPACT VALUE, AND GRAIN COUNT OF BITS TREATED AT —105 F

Material	Treatment	Hardness, Rockwell: Standard	Subzero	Impact, ft-lb: Standard	Subzero	Grain count (intercept): Standard	Subzero
M-2	1	63.5	65.0	33	29	9½	10
M-2	2	63.5	66.5	42	15	10½	6½
M-2	3	64.0	67.0	7	6	8	7
M-36	1	64.0	66.5	18	12	8½	8
M-36	2	64.0	67.0	21	7	8	6½
M-36	4	64.5	67.5	4	4	4	4½
T-1	3	63.0	65.0	31	32	11	11½
T-1	5	63.5	65.5	20	51[a]	10	12
T-1	7	63.5	66.5	8	8	5	6
T-5	4	64.0	66.5	15	13	10½	7½
T-5	6	64.5	68.0	4	4	6	6
T-5	7	65.5	67.0	4	4	5½	5

[a] Questionable; no explanation for high value.

[1] Metallurgist, Delco Products Division, General Motors Corporation.

Contributed by the Research Committee on Metal Cutting Data and Bibliography and presented at the Semi-Annual Meeting, Pittsburgh, Pa., June 19–22, 1944, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

determined by the intercept method. Table 2 records the hardness, impact value, and grain count on each bit.

Following the heat-treatment all bits were ground to conform to the following specifications:[2] Back rake, 8 deg; side rake, 14 deg; front relief angle, 6 deg; side relief angle, 6 deg; end cutting angle, 6 deg; side cutting angle, 15 deg; nose radius, $^3/_{64}$ in. Care was taken that all bits were ground exactly alike.

When the bits were completely finished a 5-in-diam bar of N.E. 8620 of 240 Bhn, was set up in a 14-in. Model C Monarch toolroom lathe, and a preliminary cut made on it to remove the hot-rolled surface, reducing it to 4.950 in. diam. The bar was 23 in. long, having an available machining surface of 20 in. By varying the revolutions per minute (rpm) with respect to the diameter, a surface speed of 200–219 fpm was maintained, according to the following formula:

$$\text{Surface speed} = 0.262 \times \text{diam of bar} \times \text{rpm of chuck}$$

The cutting point of the tool was set on the center of the bar, and the shank of the tool was at 90 deg to the center axis of the bar. The depth of cut was held uniform at 0.050 in., and the feed at 0.0075 in. per revolution. The operation time of each bit was timed with a stop watch. The point of failure of each bit was obvious and abrupt. After failure, each bit was reground and made ready for a repeat test. All 24 bits were tested five times. It should be noted that small pieces of the tool bit were left in the bar at the point of failure. It was necessary to remove these and the "glazed" surface before starting another test bit.

Results of Tests

After tabulating the operating time on each bit, the average time was calculated for the five test runs. In order to present the comparison of the results on the various materials and heat-treatments, an arbitrary figure, which will be referred to as the "productivity" of the tool, was developed. This figure was arrived at by calculating the volume of metal removed, according to the following formula:

$$\text{Volume of metal removed (cu in.)} = \text{depth of cut} \times 12 \times \text{speed (fpm)} \times \text{feed} \times \text{operating time}$$

With the feed depth of cut constant, the formula reduces to

$$\text{Volume of metal removed} = 0.0045 \times \text{speed} \times \text{operating time}$$

or

$$V = 0.0045\, ST = \text{productivity} = P$$

Thus

$$P = 0.0045\, ST$$

From this formula the productivity was calculated for each test run, and the average productivity determined for each bit.

TABLE 3 AVERAGE OPERATING TIME AND PRODUCTIVITY OF TOOLS TESTED

Material	Treatment	Average operating time, min: Standard	Average operating time, min: Subzero	P: Standard	P: Subzero	Increase in P, subzero over standard, per cent
M-2	1	15.85	17.60	15.00	16.88	12.5
M-2	2	20.36	29.20	19.80	27.38	38.3
M-2	3	29.29	45.32	27.20	42.60	56.5
M-36	1	3.22	3.50	2.98	3.22	8.0
M-36	2	20.86	28.95	19.10	27.40	43.5
M-36	4	32.18	47.38	30.85	46.00	49.0
T-1	3	4.26	4.69	4.00	4.35	8.5
T-1	5	15.18	20.10	14.30	18.55	29.8
T-1	7	20.52	29.02	18.83	26.58	41.0
T-5	4	2.41	3.18	2.28	3.05	33.7
T-5	6	13.65	18.61	13.30	17.43	31.2
T-5	7	34.21	49.15	31.80	46.30	45.5

Table 3 lists the average operating time and the average productivity for each tool tested.

Interpretation of Results

From examination of Table 3, it is immediately apparent that the low refrigeration treatment is immensely effective. Further, it may be noted that refrigeration is much more effective on the steel when it is austenitized at comparatively high temperatures.[3] It may also be noted that the higher austenitizing temperatures produce a superior cutting tool even when the tool is not subjected to refrigeration. However, this does not mean that high-speed tool steels may be heated at or over the top of the austenitizing range promiscuously. It must be remembered that these treatments were controlled very closely both regarding time of soak and temperature range. A small increment of prolonged soaking time or higher temperature would cause incipient fusion, thus producing a "burned structure." This structure would be extremely brittle and would be practically useless in operation.

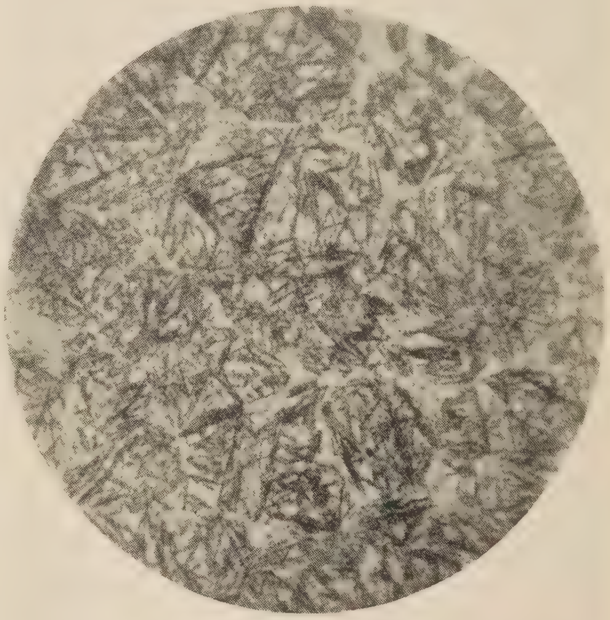

Fig. 1 M-36 Steel, No. 4 Standard Heat-Treatment
(Aqua regia etch; ×1000.)

The superiority of the bits that are austenitized at the top of the range is probably due to the more complete solution of the carbides contained in the steel. Microsections of these treatments show very little free carbide areas. In contrast to this, the bits that were austenitized at temperatures on the low side of the range or below, revealed very high percentages of free carbides. The carbides are much more effective as abrasive resistors when they are in solution. Consequently, it should be the object of the hardener to get as much solution as possible. There are patented and so-called "secret processes" which utilize this fact as a basis for their treatment. However, they use very high temperatures (2500–2700 F) in the austenitizing cycle and depend solely upon the operator to remove them before damage is done to the tool. It is believed that the same results may be obtained by close temperature control at the top of the austenitizing range.

Photomicrographs reveal a very definite structure is caused by subzero treatment. This structure is decidedly different from that obtained by standard treatments. Further, this in-

[2] "The Effect of Hardness on the Machinability of Six Alloy Steels," by O. W. Boston and L. V. Colwell, Trans. American Society for Metals, 1943, vol. 31, pp. 955–979.

[3] "The Transformation of Retained Austenite in High Speed Steel at Subatmospheric Temperatures," by P. Gordon and M. Cohen, Trans. American Society for Metals, 1942, vol. 30, pp. 569–591.

Fig. 2 Steel of Fig. 1 Subzero-Treated Between Tempers
(Aqua regia etch; ×1000.)

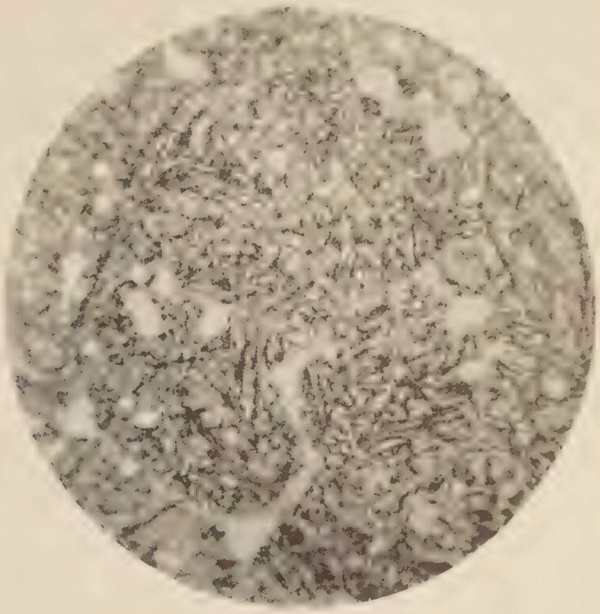

Fig. 3 M-2 Steel, No. 3 Standard Heat-Treatment
(Aqua regia etch; ×1000.)

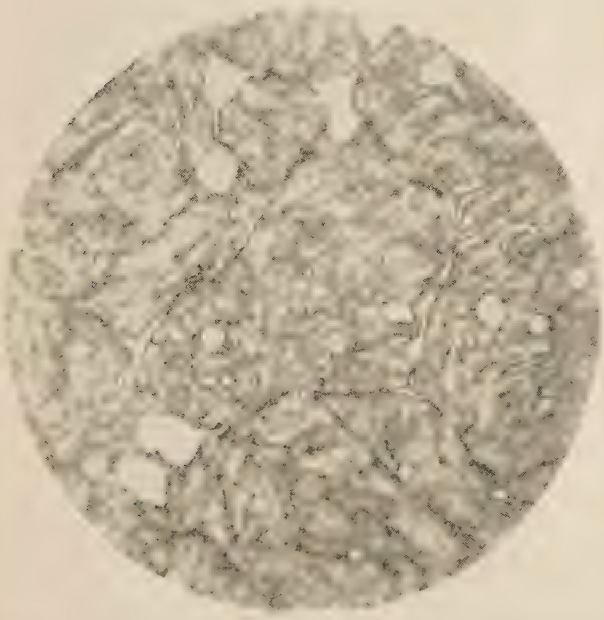

Fig. 4 Steel of Fig. 3, Subzero-Treated Between Tempers
(Aqua regia etch; ×1000.)

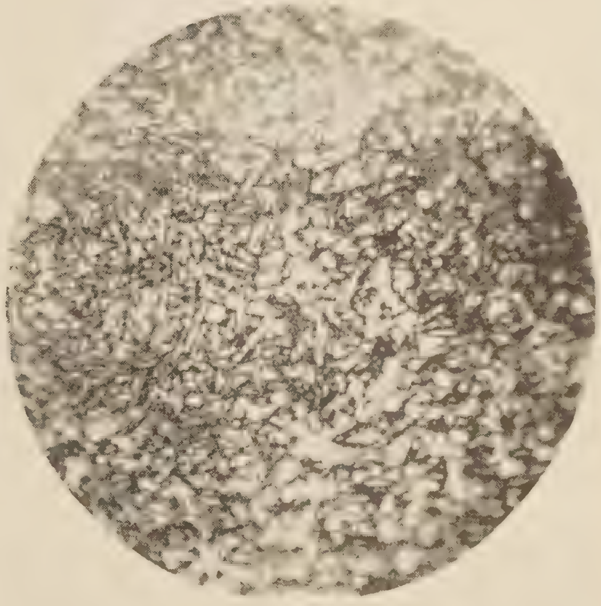

Fig. 5 M-2 Steel, No. 2 Heat-Treatment, and Subzero-Treated at —85 F Between Tempers
(Aqua regia etch; ×1250.)

vestigator has not been able to produce the desired structure or productivity on tools that are treated at —90 F or higher.

Fig. 1 is a photomicrograph of M-36 steel treated with No. 4 treatment, but not subzero-treated. The familiar acicular formation of the martensite is apparent. Fig. 2 reveals the structure obtained in the same steel with the same treatment only that it has been subzero-treated between the tempers. It may be seen that the martensite is no longer heavy acicular, but rather fine, giving a peppery structure. The same phenomenon is present in Figs. 3 and 4. The microsections were taken of M-2 steel using No. 3 heat-treatment. Fig. 3 is standard-treated, and Fig. 4 is subzero-treated. The apparent absence of the familiar carbides in Figs. 1 and 2 is the result of the extreme austenitizing temperature. This temperature has caused almost all of the carbides to go into solution.

The formation of the fine martensitic structure on the refrigerated material is believed by the author to be caused by the intense internal strains and pressures caused by the extreme low temperature. It is suggested that these pressures also caused the transformation of austenite to martensite to go to completion. Fig. 5 is a microsection of M-2 steel which was treated ac-

cording to No. 2 treatment, and refrigerated at —85 F, between the tempers. The heavy acicular martensitic structure is still present and blocky austenite may be seen. It may also be noted that the results obtained on cutting tests were very much the same as on an unrefrigerated tool. Consequently, it is believed that the pressures set up at —85 F are not sufficient to affect the structure of a material that has high alloy content, and the sluggish transformation properties that are present in high-speed steels.

Conclusions

The following general conclusions have been arrived at by a study of the results of this research:

1 High austenitizing temperatures should be used to produce the best structure for metal removal. However, the control of these temperatures and the austenitizing time must be accurate.

2 Refrigeration treatment of high-speed steel at —100 F or lower between the tempers will produce a superior tool.

3 Low austenitizing temperatures leave too many undissolved carbides, thus producing an inferior tool. Therefore, they should be avoided.

4 For cutting operations on hard alloy material, where intense heat is generated at the cutting edge, cobalt-bearing high-speed steel will generally prove superior.

Acknowledgment

Acknowledgment is given to Mr. Ray P. Kells of the Latrobe Electric Steel Company for furnishing the high-speed steel used in this research and the many suggestions that he made to the investigator.

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2 "The Effect of Hardness on the Machinability of Six Alloy Steels," by O. W. Boston and L. V. Colwell, Trans., American Society for Metals, vol. 31, 1943, pp. 955–979.

Machinability of Plain-Carbon, Alloy, and Austenitic (Nonmagnetic) Steels, and Its Relation to Yield-Stress Ratios When Tensile Strengths Are Similar

By E. J. Janitzky,[1] Chicago, Ill.

It is possible by the use of yield-stress ratios of plain-carbon, alloy, and austenitic (nonmagnetic) steels of the same tensile strength to obtain an index of machinability for rough-turning. The relation between Taylor speed and yield-stress ratios of the same tensile strength is expressed mathematically in the paper. A graphical presentation is also given. The yield-stress ratios of steels of the same tensile strength allow co-ordination of machinability of the types of steel listed by the same method, a distinct advantage over the use of any one of the four components of the tensile test alone, or a combination of stress and strain.

The quest for an index of machinability began with Taylor's[2] establishment of the conception that the measure of machinability may be expressed by the cutting speed at which a standard tool will last for periods of 20 min, 1 hr, or more.

Taylor also found that the change of life of a high-speed steel tool, resulting from a change in the cutting speed, could be represented approximately by an empirical equation

$$VT^n = C \qquad [1]$$

in which

V = cutting speed, fpm
T = tool life, min
n = an exponent
C = constant which is dependent for its numerical value upon exact cutting conditions (other than speed). It will vary with the form and size of tools, the material cut, the steel and treatment of the steel from which the tools are prepared, the feed and depth of cut

Curves from which V is deduced are generally called TV curves.

Taylor also developed an empirical relation for carbon steels between cutting speed and the tensile properties of the steel tested by using the tensile strength and elongation. The equation gives some indication regarding machinability but, with the advent of alloy steels, it lost its usefulness. Taylor, however, was not satisfied with the evaluations by this equation.

The present tendency to judge machinability by tensile strength, yield stress, Brinell hardness number or reduction of area, separately, is not satisfactory. Two steels with equal tensile strength or equal yield stress may have different machining properties depending upon the ratio of yield stress to the tensile strength. Likewise two equal Brinell hardness numbers do not necessarily signify equal machinability.

[1] Consulting Metallurgical Engineer, Carnegie-Illinois Steel Corporation, South Works.

[2] "On the Art of Cutting Metals," by F. W. Taylor, Trans. A.S.M.E., vol. 28, 1906, pp. 31–279.

Contributed by the Research Committee on Metal Cutting Data and Bibliography and presented at the Semi-Annual Meeting, Pittsburgh, Pa., June 19–22, 1944, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

Relationship of Stress and Strain

Before going into the discussion of machinability of steel and its relation to yield-stress ratios when the tensile strengths are similar, it is advisable to consider the relation of stress and strain as they manifest themselves in the tensile test.

A statistical analysis of the relation of the yield-stress ratio to the deformation ratio may be helpful in arriving at a better understanding of the subject.

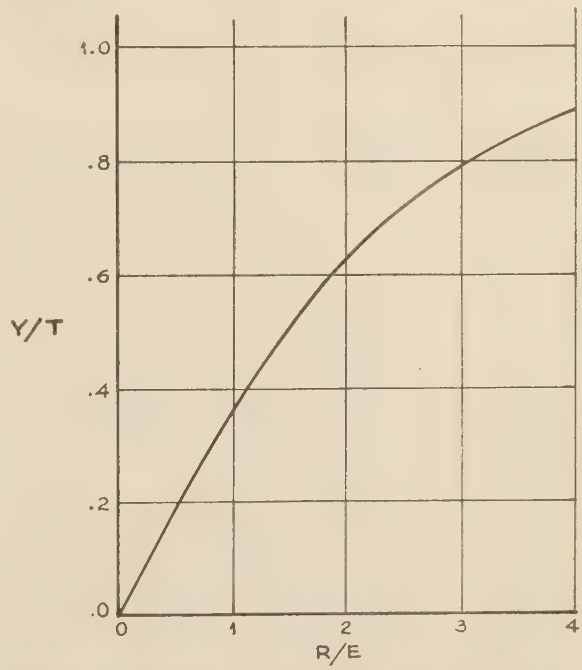

Fig. 1 Illustrating Relation of Deformation Ratio to Yield-Stress Ratio

Yield-stress ratio is defined as the ratio of yield stress to tensile strength, and deformation ratio, the ratio of reduction of area to elongation.

By plotting yield-stress ratios ($Y:T$) against deformation ratios ($R:E$), one obtains a curve as shown in Fig. 1.

The curve may be expressed by the equation

$$\frac{Y}{T} = 1 - e^{-0.416}\left(\frac{R}{E}\right)^{1.133} \qquad [2]$$

where

T = tensile strength, psi $\times 10^{-3}$
Y = yield stress, psi $\times 10^{-3}$
R = reduction of area, per cent
E = elongation, per cent, in a gage length 4 times diameter
e = base of natural logarithm

The equation was developed by using yield stress at a set of 0.2 per cent gage length and a pulling speed of 0.11 ipm.

From Equation [2], we perceive that, for a given deformation ratio, there is a corresponding yield-stress ratio. We also perceive that at constant tensile strength with increasing deformation ratio, the yield-stress ratio increases and, consequently the yield stress and vice versa.

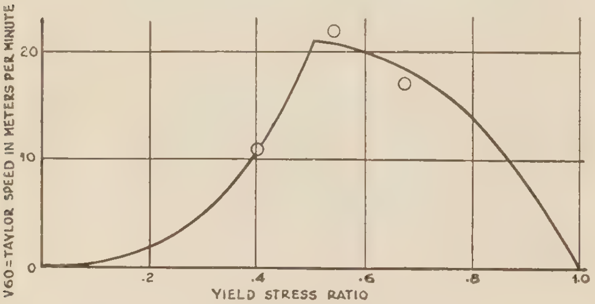

Fig. 2 Representing an Isorotic Cycle for a Tensile Strength of 64.5 Kg Per Sq Mm or 91.5 Psi $\times 10^{-3}$ and Different Yield-Stress Ratio

From the foregoing observations, it is also evident that co-ordination of machinability with any of the four properties reported in making the tensile test is impossible and leads to erroneous conclusions. The use of indentation hardness for co-ordination of machinability properties may also result in misleading conclusions. An austenitic steel (nonmagnetic) may possess the same Brinell hardness numeral as a quenched and tempered or annealed plain-carbon or alloyed steel and have different tensile properties and, therefore, different machinability properties.

As already mentioned, steels of the same tensile strength may possess different yield-stress ratios and, consequently, different machinability properties. Thus a plot of different yield-stress ratios, but of the same tensile strength, against Taylor speed appears to be the clue to the present enigma.

However, some steels of the same tensile strength but different yield-stress ratio may possess the same Taylor speed. This coincidence manifests itself in nonmagnetic steels of austenitic structure or very low-carbon steels of ferritic structure; that is, in steels of a yield-stress ratio lower than 0.5. Thus the coincidence suggests a family of reversing cycles of the same tensile strength but different yield-stress ratios.

When Taylor speeds are plotted on the ordinate and yield-stress ratios of the same tensile strength on the abscissa, a reversing cycle is obtained.

The cycle, as illustrated in Fig. 2, starts at a yield-stress ratio of 0 and ascends up to a yield-stress ratio of 0.5. From this point the cycle descends to the X axis and terminates at a yield-stress ratio of 1.0.

In Fig. 2, the cycle represents steels of a tensile strength of 64.5 kg per sq mm or 91.5 psi $\times 10^{-3}$, including a nonmagnetic

TABLE 1 COMPILATION OF DATA ON MATERIAL UNDER TAYLOR TEST (LANGENBACH)

1	2	3		4	5	6	7
No.	S.A.E. grade	Heat-treatment, deg C		Tensile strength, kg per sq mm	Yield stress, kg per sq mm	Taylor speed V_{60}, m per min Observed	Calculated
1	1020	N 920 C 1688 F		37.5	22.0	39.5	39.2
2	1025	N 880 C 1616 F	T 700 C 1292 F	46.3	26.3	31.5	30.7
3	1035	O 850 C 1562 F	T 720 C 1328 F	55.0	28.0	26.5	25.5
4	1045	O 850 C 1562 F	T 720 C 1328 F	64.7	34.0	21.5	20.9
5	1070	A 720 C 1328 F		75.4	39.5	19.0	17.4
6	1075	O 850 C 1562 F	T 650 C 1202 F	83.5	43.2	18.0	15.5
7	1080	O 850 C 1562 F	T 650 C 1202 F	90.0	45.7	16.5	14.1
8	X-1020	W 820 C 1508 F	T 650 C 1202 F	51.5	30.5	24.0	26.7
9	X-1030	W 820 C 1508 F	T 650 C 1202 F	63.1	38.8	20.0	20.5
10	T-1340	O 850 C 1562 F	T 680 C 1256 F	74.2	50.5	15.0	15.5
11	2320	O 850 C 1562 F	T 670 C 1238 F	58.6	39.2	19.0	20.9
12	2520	O 850 C 1562 F	T 670 C 1238 F	61.8	41.7	17.5	19.4
13	2330	O 830 C 1526 F	T 500 C 932 F	65.6	45.0	17.0	17.7
14	5130	O 840 C 1544 F	T 550 C 1022 F	81.5	53.0	14.0	14.5
15	5140	O 840 C 1544 F	T 550 C 1022 F	87.5	65.6	12.5	11.6
16	5160	O 840 C 1544 F	T 550 C 1022 F	96.6	70.3	11.5	10.7
17	3230	O 850 C 1562 F	T 670 C 1238 F	58.5	42.0	18.5	19.2
18	3330	O 820 C 1508 F	T 680 C 1256 F	76.3	56.5	14.0	13.3
19	3340	O 820 C 1508 F	T 620 C 1148 F	93.4	72.5	10.0	9.4
20	5 per cent Ni 1.5 per cent Cr	O 820 C 1508 F	T 550 C 1022 F	118.5	94.0	7.5	6.7

Note: N = normalized, A = annealed, O = oil-quenched, W = water-quenched, T = tempered.

stainless steel of the S-18-8 grade with a yield-stress ratio of 0.4 (own datum).

The cycle may be identified as reversing parabolic cycle with the ascending branch of the parabola (a cubic parabola) having its vertex at 0 yield-stress ratio and the descending branch (an ordinary parabola) having its vertex at the maximum V_{60}, corresponding to a yield-stress ratio of 0.5.

Thus we may express the ascending part of the cycle

$$V_{60} = c_1\left(\frac{Y}{T}\right)^3 \qquad [3]$$

and the descending part of the cycle

$$V_{60} = c_2\left[0.25 - \left(\frac{Y}{T} - 0.5\right)^2\right] \qquad [3a]$$

where

$\frac{Y}{T}$ = yield-stress ratio (Y = yield stress, T = tensile strength)
V_{60} = Taylor speed in meters or feet per minute
c_1 and c_2 = factors

Calculating the factor c_2 from data of Langenbach,[3] given in

[3] "Die Zerspannbarkeit-Kennziffer V_{60} in ihrer Beziehung zur Zugfestigkeit und Streckgrenze beim Schrupp-Dreh Vorgang von legierten und unlegierten Staehlen," by F. Langenbach, Technische Hochschule, Aachen, 1932.

Table 1, and plotting it against the corresponding tensile strength, we obtain a hyperbolic relation that is

$$c_2 = \frac{12{,}500}{T^{1.2}}$$

By interpolating the data from the stainless steel

$$c_1 = \frac{25{,}000}{T^{1.2}}$$

Thus Equation [3] will read

$$V_{60} = \frac{25{,}000}{T^{1.2}} \left(\frac{Y}{T}\right)^3$$

and Equation [3a]

$$V_{60} = \frac{12{,}500}{T^{1.2}} \left[0.25 - \left(\frac{Y}{T} - 0.5\right)^2\right]$$

In order to transfer the equation into feet per minute when T is expressed in psi $\times 10^{-3}$, $c_1 = 116{,}642$ and $c_2 = 58{,}464$. The factors c_1 and c_2 in this special case are for the dimension of the tool, for its heat-treatment and chemical composition, the depth of cut, and feed.

Credence to these equations is given in Table 1. Data of observed values are in column 6 and the approximated values are in column 7. A family of cycles in steps of 10 kg per sq mm or 14.2 psi $\times 10^{-3}$ is also given in Fig. 3, for comparing relative machinability.

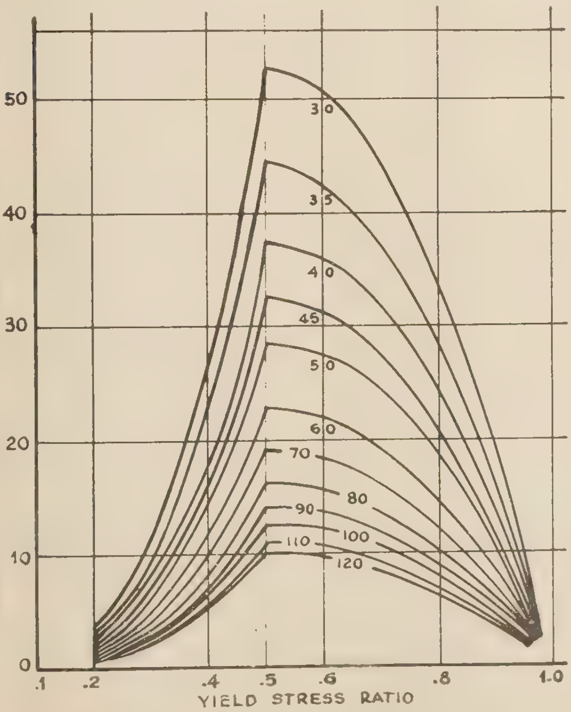

Fig. 3 Family of Isorotic Cycles for the Graphical Approximation of Taylor Speed (V_{60})
(Numerals at each cycle are kg per sq mm.)

It will be noted by comparison of calculated and actual Taylor speed, that variation in chemical composition does not influence machinability, provided that the steels can be heat-treated to have the same tensile properties. In other words, if two steels are heat-treated to have the same tensile properties (tensile strength, yield stress, elongation, and reduction of area), assuming such heat-treatment is possible, they will be equal in regard to machinability. Leaded, sulphurized, or phosphorized steels, or steels containing other elements as selenium, etc., are exceptions and must be co-ordinated as such. Another method of co-ordination may be applied by including tool angles, that is, the most efficient tool angle for yield-stress ratios of the same tensile strength.

Co-ordination of cold-worked steels of the same tensile strength but different yield-stress ratios may also be accomplished. It is known that cold-drawn or cold-rolled bars machine easier than bars in their natural conditions. It is also known that by cold deformation the yield-stress ratio and also the tensile strength are raised above those in their prior condition. Thus, the machinability of cold-work-processed steels may be also co-ordinated.

When using the same tool and the same feed and depth of cut for steels cold-work-processed to the same tensile strength but different yield-stress ratios, machinability will increase with increasing yield-stress ratio. However, the higher the yield-stress ratio of the steel before cold-processing, the less increase in machinability is to be expected, and cold-work-processing becomes impracticable or technically impossible.

In order not to confuse the conception of machinability, one has to consider that it is a different matter when speed of machining is the primary object and surface finish secondary. In the first case it is desirable to have a lower yield-stress ratio than in the second case, when the material is of the same tensile strength.

Cutting-speed variations also influence the surface finish, that is, the higher the permissible speed the better the surface appearance.

Details of Machinability Test

Following are the details of the machinability test on which the development is based. The data were taken from F. Langenbach's machinability test and selected not only because of the wide range of analyses represented, but also because of precautions used in making the tests, and the fact that only a limited zone was used for making both the machinability tests and the tests for physical properties. As the material under test (bars from 6 to 8 in. diam, and from 60 to 120 in. long) was in the heat-treated condition, it is very important how deep the ma-

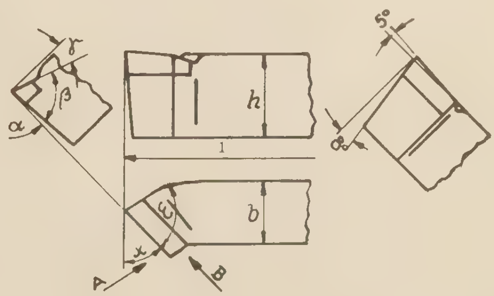

Fig. 4 Dimensions of Tool
(Arrows A and B direction of view. h = 1.574 in.; b = 1.18 in.; 1 = 13. in.; α = 6 deg; β = 65 deg; γ = 19 deg; x = 45 deg; ϵ = 80 deg.)

chining was performed in regard to the diameter or the location of the tests taken for tensile strength. Fortunately, the machining depth was limited to 2 inches maximum in order to avoid discrepancies on account of hardness gradients, and the tensile-test specimens were taken longitudinally, close to the surface at the ends of the rounds (gage length, 5 times the diameter); yield stress at 0.2 per cent of gage length.

The machining was of the roughing type and dry. The tools were of 18-4-1 composition and the dimensions are given in Fig. 4. The recorded data on machinability represent a depth of cut of 0.236 in. and a feed of 0.039 in. per revolution. The Taylor speed, V_{60}, was obtained by Taylor's method from the TV curves. Each of the TV curves was duplicated and the data so obtained were averaged. If the duplicate tests were not in agreement a triplicate test was run.

The motive of Langenbach's work was to investigate the influence of yield stress on the machinability of steels, as he was aware of the fact that machinability of steels of the same tensile strength decreased with increasing yield stress, however, a co-ordination was not attempted. The present paper deals with the co-ordination. It is perhaps not out of place to mention that the second object of these tests was to establish whether there was a difference between tipped tools when compared with standard tools. Therefore, the machinability of each steel was determined with tipped tools and with standard tools and was found to be independent of the type of tool used.

The Influence of Through-Metal on the Heat Loss From Insulated Walls

By VICTOR PASCHKIS[1] and M. P. HEISLER,[2] NEW YORK, N. Y.

The heat flow through insulation containing "through-metal," or "thermal short-circuits" has been investigated. The heat flow through such a structure (occurring, for example, in the insulation of ship hulls) can be considerably larger than would be found from adding to the heat flow through the insulation that contributed by the "through-metal," as if the two were independent. Thus the apparent effectiveness of an insulation can be very much smaller than would be calculated from the thermal conductivity alone. From the many different forms of thermal short-circuits, one specific type has been selected for investigation, i.e., that of metal strips or fins extending through insulation which is covered on its surfaces by sheet metal. General curves are developed and their application explained, from which an "increase factor" for any condition (conductivity and thickness of the insulation, conductivity and thickness of the strips, outside film conductance) can be read. The "increase factor" is defined as the ratio of the heat flow through an actual structure, compared with the flow that would occur if the insulation and heat flow through metal strips were independent of each other. The method of developing these general curves and the basis of the experimental technique used are explained in two Appendixes.

Statement of Problem

FREQUENTLY, in the practical application of thermal insulation "through-metal" has to be used. The term "through-metal" is here used for any metal extending from the hot surface to the cold one. Because metal has a very much higher conductivity than any insulating material, such through-metal acts like a thermal short-circuit, and therefore greatly influences the flow of heat through a structure. The present paper deals with some problems in connection with the heat losses caused by "through-metal."

Out of the great variety of possible shapes and sizes of such through-metal, one arrangement has been selected as the object of the investigations; the case of a strip of metal connecting the hot surface with the cold surface. Fig. 1 is an isometric sketch of this arrangement. It shows an insulation (of thickness L) covered by sheet metal a and b (thickness W_M). The insulation is assumed to extend infinitely in the x- and y-directions, but in the x-direction it is interrupted by infinitely long strips c (thickness $2\,W_M$) which are spaced at regular intervals W_I. (In case of block insulation W_I represents the width of an individual block.) Arrangements as shown in Fig. 1 are frequently used. Typical examples are the insulated hulls of ships, walls of stratochambers, and industrial ovens with metallic walls. Similar problems occur in almost all insulating practice. By way of example, the following cases may be mentioned: Metal-enclosed bricks such as are used in electric furnaces and in open-hearth furnaces; pipe flanges extending through the pipe insulation; anchor bolts and pipes used in boiler insulation; and mortar joints in brick walls.

It is obvious that such through-metal influences the heat loss of the structure, and also that, at any appreciable distance from the surface, the metal will influence the temperatures in the adjacent insulation.

In determining the heat loss from a structure incorporating through-metal, the oldest and empirical way is to estimate the increase of heat loss due to the metal and account for this increase by an additive, or by a multiplication correction factor, found from experience with similar arrangements.

A distinct step forward is the separate calculation of the heat flowing through the insulation and through the metal, and adding both heat flows. This method is based upon the assumption that the metal does not influence the temperatures in the surrounding insulation. This assumption, however, is in many cases not tenable, as will be evident from this paper. The correct approach must take the lateral flow into consideration.

A general solution for this problem, covering any thickness L of the structure, any spacing W_I of the strips, any thickness W_M of the metal, as well as any conductivity of insulation k_I and metal k_M, is the main object of this report. Only three assumptions have been made which limit the generality:

1 The film conductance on the hot surface is considered to be infinite (this means that the hot surface has the same temperature as the adjacent air (ambient).

2 The thickness of the metallic strips is twice that of the sheet metal covering the surfaces.

3 The thermal properties (conductivities and cold-side film conductance) do not change with temperature.

Lateral Heat Flow

Assume a wall of thickness L (Fig. 2). The hot side is maintained constantly at a temperature t_h, the cold side at a temperature t_{cs}.

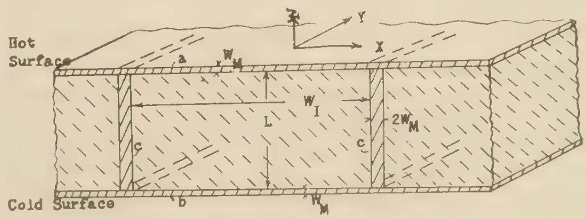

Fig. 1 Isometric View of Investigated Structure

Now if the wall is homogeneous, then independent of the thermal conductivity of the material of the wall, the temperature at any given distance l from the hot surface always has the same value as given by

$$t = t_{cs} + \frac{L - l}{L}(t_h - t_{cs}) \quad \ldots\ldots\ldots\ldots\ldots [1]$$

If (Fig. 3) two materials with thermal conductivities k_1 and k_2

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[2] Assistant in the Heat and Mass Flow Analyzer Laboratory, Department of Mechanical Engineering, Columbia University.

Contributed by the Heat Transfer Division and presented at the Spring Meeting, Birmingham, Ala., April 3–5, 1944, of The American Society of Mechanical Engineers.

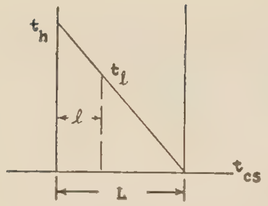

Fig. 2 Temperature Drop Through Homogeneous Wall

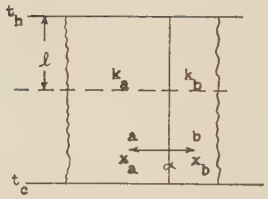

Fig. 3 Interface Between Two Materials

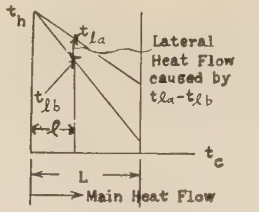

Fig. 4 Temperature Drop Causing Lateral Flow

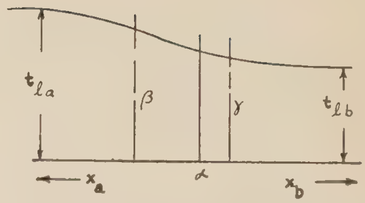

Fig. 5 Temperature Distribution Near Interface

are subjected in parallel to the same temperature difference, then their temperatures at a distance l from the hot surface are the same. Between points of equal temperature, no heat transfer can of course occur.

Matters are different however if the cold surface, instead of being held at a constant value, is (Fig. 4) facing an ambient of constant temperature t_c. Now the temperature at point l is not only a function of the quantities appearing in Equation [1] but also of the conductivity k of the material and the film conductance h at the cold surface. The temperature at l can now be found from

$$t_l = t_c + (t_h - t_c)\left(1 - \frac{l}{1 + \frac{k}{h}}\right) \quad \ldots\ldots\ldots\ [2]$$

Considering again the arrangement in Fig. 3, it is obvious that, since the two materials have different thermal conductivities, there will result different temperatures at each and any level l. Fig. 4 contains two such temperature lines for materials of different conductivities. At distance l from the hot surface, material a will have temperature t_{la} and material b temperature t_{lb}. Of course heat will flow from the point of higher temperature to that of lower temperature. This flow of heat is perpendicular to the main flow, which goes from the hot to the cold surface. It is called hereafter the "lateral" heat flow. The lateral heat flow is proportional to the difference of temperatures $(t_{la} - t_{lb})$. This difference in turn depends upon the relative magnitude of the values k_a/h and k_b/h, as becomes obvious from Equation [2]. If h approaches infinity ($h = \infty$ is the mathematical expression for the surface having the same temperature as the ambient), Equation [2] becomes identical with Equation [1]. No lateral heat flow occurs. If h approaches zero (which is the mathematical expression for a wall covered on the cold side by an absolute insulator), t_l becomes t_h and the entire wall is at uniform temperature equal to the hot-surface temperature.

The lateral heat flow does not leave the temperature field unchanged. If there is no contact resistance between the two materials (Fig. 3), the interface will show a temperature drop line, which lies between that for material a and material b. Adjacent strata will have almost the same temperatures, and only at some distance from α will the influence of the other material be no longer noticeable. Fig. 5 shows schematically the temperatures at level l. The distances x are plotted as abscissas on both sides of position α; temperatures are plotted as ordinates. If in either material the temperatures are taken close to α, say at β and γ, then different temperatures are obtained from those which would be expected if no lateral heat flow occurred; namely, different from t_a and t_b.

Referring to Fig. 1, let $W_I = 2.5$ in., $L = 9$ in., and the thickness $2\ W_M$ of the metal strips c be $^1/_8$ in. Let the conductivity of the insulation be 0.25 Btu/ft, hr, F, the conductivity of the metal strips and sheets be 25 Btu/ft, hr, F, and the film conductance h be 2 Btu/sq ft, hr, F.

In Fig. 6 the thickness L is plotted as abscissa, and temperature as ordinate. The two dotted lines show the temperatures that would exist if no lateral flow occurred. Because of the lateral flow, however, the temperatures approach each other. Near the metal, the line s prevails and in the center of the brick, the line c.

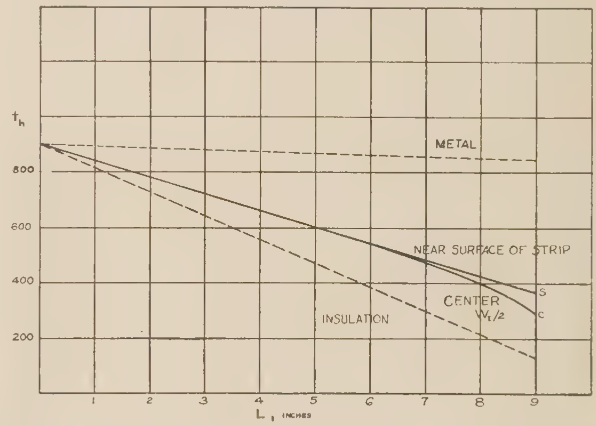

Fig. 6 Temperature Distribution With "Through-Metal"

List of Variables

In order to obtain a general solution, a survey of the variables which influence the heat flow was made. The heat flow Q for a structure, as shown in Fig. 1, is governed by the following:

1 The temperature difference between the hot surface and the cold ambient.
2 The net spacing W_I of the strips.
3 The thickness L of the insulation.
4 The thickness W_M of the strips.
5 The conductivities k_I of the insulation and k_M of the strip.
6 The surface-film conductance h.

Instead of plotting a great number of values of Q for many different combinations of the properties involved, it appeared simpler to compare the actual heat loss with a standard value. Although different possibilities for this value are available, it was decided to compare Q with the value that would be obtained if no lateral heat flow occurred. The ratio thus observed

$$\frac{\text{Actual heat loss with lateral heat flow}}{\text{Heat loss with no lateral heat flow}}$$

is called the "increase factor," and is abbreviated hereafter by I.

In order to determine the heat loss for any actual structure, such as Fig. 1, proceed as follows:

1 Determine the heat loss Q_s, i.e., the heat loss assuming no lateral heat flow.

2 Determine the value of I (see "Presentation of Results").

3 Multiply Q_s by I; this gives the actual heat loss Q with lateral heat flow.

4 Q_s can be found from Equation [3], which represents the basic law of steady-state heat flow

$$Q_s = (t_h - t_c)\left(\frac{W_I}{\frac{L}{k_I} + \frac{1}{h}} + \frac{W_M}{\frac{L}{k_M} + \frac{1}{h}}\right) \qquad [3]$$

In this equation, as in the entire paper, it is assumed that the strips repeat at equal distances, so that it is sufficient to consider one space W_I and one strip c.

Method of Experiments

All experiments described in this paper were carried out by the "electric analogy method" on the "heat and mass flow analyzer" at Columbia University. This method is based on the identity of laws governing the flow of electricity and of heat in a body. Appendix 1 contains a more detailed discussion and description of the method. Here it may suffice to say that in the analogy:

Electric conductivity represents thermal conductivity.
Voltage represents temperature difference.
Current represents heat flow.

The application of the method consists of building an electrical circuit which is analogous to the structure, measuring the current through the structure for applied voltage (which corresponds to the temperature difference across the structure), and translating the measured current into heat units.

Influence of Individual Variables

It is of interest to study the influence of the different variables just mentioned on the increase factor I.

The temperature difference $t_h - t_c$ has no influence on I. Q_s as well as Q is proportional to the temperature difference, and therefore, I, which by definition is $I = \frac{Q}{Q_s}$, is independent of temperature.

If W_I becomes very small and approaches zero, the influence of the insulating material as compared with the metal strips vanishes. Consequently, for $W_I = 0$, I becomes unity. On the other hand, if W_I becomes very large, the influence of the strips vanishes, and the structure behaves as if it consisted only of insulating material. For $W_I = \infty$, therefore, I again approaches unity. Between $W_I = 0$ and $W_I = \infty$, however, I must have values larger than 1. At some value a maximum will occur.

If L is very small, the thermal resistance of the insulation as well as of the strip becomes very small; the major part of the resistance then lies in the surface film. There will be almost no temperature differences within the structure; therefore very little lateral heat flow occurs, and, as a limit, I becomes unity (for $L = 0$). If, on the other hand, L becomes extremely large, the resistance in the film becomes insignificant; and all points at the same distance l from the hot surface have the same temperature, independent of the material involved. Again no lateral heat flow can develop, and I becomes unity when L equals infinity. The character of the curve is the same as that shown in Fig. 7, for W_I.

It is obvious that a change of W_M will result in a similar curve. If W_M approaches zero, almost all heat flow is limited to the insulation. If on the other hand W_M becomes exceedingly large the influence of the heat flow through the insulation can be entirely neglected, making I equal to unity.

If either of the two conductivities (k_I or k_M) becomes zero or infinite, the importance of the part of the structure having the other conductivity becomes either alone dominant or entirely insignificant, e.g., let $k_I = 0$. Then only the heat flow through

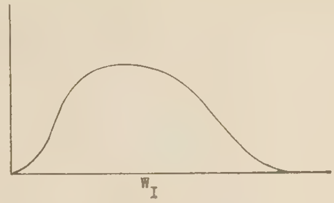

Fig. 7 Characteristic Shape of I—W_I Function

c, the metal strips, with the width W_M remains. If $k_I = \infty$, the heat flow through c becomes entirely insignificant. (If k_M changes, the situation is correspondingly reversed.) Consequently, either of the two conductivities will yield a curve for I of the same character as that shown in Fig. 7 for W_I.

It has been mentioned that for $h = 0$ and for $h = \infty$, the value of I becomes unity, while for other values of h, I is larger than unity.

Summarizing, it can be stated that the curve for I plotted against any single variable (the other variables held constant) yields curves of the same general character: For the two limiting cases, the variable being zero or infinity, I is unity; for all values of the variable in between, I is larger than unity.

Presentation of Results

The presentation of I as a function of six different variables would call for a vast number of charts, which would take long to plot, and then would still be unwieldy to use. Fortunately, however, the six variables can be grouped in four parameters, which will be mentioned. Thus plotting is greatly simplified. The deduction of these parameters, and the proof of their validity, is given in Appendix 2.

The four parameters are as follows

$$f_1 = \frac{W_M}{W_I}, \quad f_2 = \frac{W_I}{L}, \quad f_3 = \frac{k_I \cdot L}{k_M \cdot W_M}, \quad f_4 = \frac{k_I}{h \cdot W_I} \qquad [4]$$

The values of the parameters to be used in the charts were so selected that at least the following limits for the individual variables are covered by the graphs:

$$\begin{aligned} W_I &= 1/12 \text{ to } 1\,1/4 \text{ ft} \\ W_M &= 1/192 \text{ to } 1/24 \text{ ft } (= 1/16 \text{ to } 1/2 \text{ in.}) \\ L &= 1/6 \text{ to } 1\,1/2 \text{ ft} \\ k_I &= 0.025 \text{ to } 1.5 \text{ Btu/ft, hr, F} \\ k_M &= 2.5 \text{ to } 200 \text{ Btu/ft, hr, F} \\ h &= 1 \text{ to } 8 \text{ Btu/sq ft, hr, F} \end{aligned}$$

For practical purposes, a greater range is covered, since any two of the variables will seldom approach their limit at the same time. Because of the relationship in Equation [4] any one of the values can become larger, if at the same time the other values change accordingly. The values are so selected that f_1 to f_4 can be obtained even if the individual magnitudes are as unfavorable as possible. For example f_1 has been investigated within the limits 0.004 to 0.5 $\left(\text{i.e., } \frac{1}{192 \times 1.25} \text{ to } \frac{12}{24}\right)$.

The results are presented in Table 1 and in charts. In all charts, values of I are plotted as ordinates and values of f_4 as abscissas. Each chart contains several curves, each curve holding for a different but constant value of f_3.

In order to enable proper interpolation, charts are presented for four different values of f_1 and four different values of f_2. Inasmuch as f_1 and f_2 can vary independently, sixteen different combinations are necessary.

The four values of f_1 are 0.004, 0.020, 0.100, 0.500.

The four values of f_2 are 0.25, 0.5, 0.5, 1.5, 7.5.

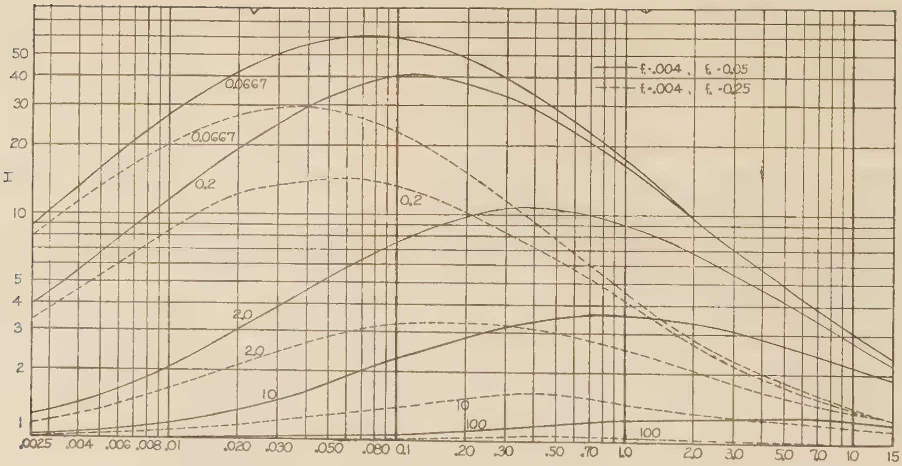

FIG. 8 INCREASE FACTOR I PLOTTED AGAINST f_4 FOR VARIOUS VALUES OF f_3

As explained, each value of f_1 can combine with any of the four values of f_2.

A first survey of the results may be obtained from Table 1, which contains the maximum values of I for different combinations of the values for f_1, f_2, f_3.

The results are shown in Figs. 8 to 14, inclusive. Table 2 shows the combinations of the different parameters to be found in the various figures.

TABLE 1 MAXIMUM VALUES OF "INCREASE FACTOR" I

f_1	f_2	0.05	0.25	1.5	7.5
0.004	f_3 = 0.0667	60.0	29.3	8.90	2.30
	0.2	41.0	14.2	3.40	1.56
	2.0	10.8	3.32	1.36	1.09
	10	3.62	1.64	1.05	1
	100	1.30	1.06	1	1
	500	1.0	1	1	1
	f_3 = 0.0667	15.4	12.1	3.2	2.03
	0.2	14.0	6.9	2.3	1.35
0.020	2.0	6.1	2.5	1.29	1
	10	2.69	1.4	1	1
	100	1.17	1	1	1
	500	1	1	1	1
	f_3 = 0.0667	5.14	4.17	2.26	1.25
	0.2	4.50	3.20	1.64	1.09
0.100	2.0	2.80	1.61	1.05	1
	10	1.67	1	1	1
	100	1	1	1	1
	500	1	1	1	1
	f_3 = 0.0667	1.21	1.168	1.27	1.02
	0.2	1.175	1.122	1.09	1
0.500	2.0	1.097	1.037	1	1
	10	1.028	1	1	1
	100	1	1	1	1
	500	1	1	1	1

TABLE 2 COMBINATIONS OF DIFFERENT PARAMETERS ILLUSTRATED IN FIGS. 8 TO 14

f_1	f_2 = 0.05	0.25	1.5	7.5
0.004	Fig. 8	Fig. 8	Fig. 9	Fig. 9
0.020	Fig. 10	Fig. 10	Fig. 11	Fig. 11
0.100	Fig. 12	Fig. 12	Fig. 13	Fig. 13
0.500	Fig. 14	Fig. 14	Fig. 13	

Wherever Table 1 shows amaximum (for I) of 1.0, no curve appears in the graphs, because I has the value 1 over the entire range of f_4.

Discussion of Experimental Results and Application of Charts

Discussion. High values of I occur with low values of f_1 and f_3. The meaning of f_1 and f_2 is so simple (from Equation [4]) that no comment appears to be necessary. The parameter f_3, however, is a complex expression

$$f_1 \cdot f_2 \cdot f_3 = \frac{k_I}{k_M}$$

A low f_3, with other f's unchanged means therefore a very high-grade insulating material with steel, or a medium insulating material with copper for the strips, etc. At high values of f_3, I becomes appreciable only under extreme conditions for the other f values (f_1 low, f_2 medium).

The parameter f_4 definitely shows a critical range in the neighborhood of 0.5. It determines the spacing of the strips and indicates that the spacing exerts its influence not directly but in combination with the thickness of the insulation. If I becomes smaller with decreasing f_2 (i.e., if the spacing of the strips becomes smaller), the heat losses do not necessarily decrease also. The heat losses are the product of the value $Q_s \cdot I$. This is important to keep in mind, because a drop in I might lead to the erroneous assumption that the heat loss drops.

Application of Curves. The curves shown in Figs. 8 to 14, inclusive, enable the determination of the actual heat loss for any condition covered by the general program outlined under "Statement of Problem." The use of the dimensionless parameters f_1 to f_4 makes the trends less easily recognized. Their use, however, is the only means of presenting the results in a practicable way. The use of the curves is best demonstrated in three examples which will cover two different insulating materials. The examples will also show how to use the curves to detect the influence of change in design.

Example 1: An insulation, L = 8 in. (0.667 ft) thick, is reinforced by steel strips $2W_M$ = $^1/_8$ in. (0.0104 ft) thick. The insulation is covered on both sides by steel sheets of the thickness W_M = $^1/_{16}$ in. (0.0052 ft). The temperature of the hot side of the structure is 570 F; the ambient facing the cold side is 70 F. The film conductance on the cold side is 2 Btu/sq ft, hr, F. The thermal conductivity of the insulating material is 0.08 Btu/ft, hr, F; that of the steel is 26 Btu/ft, hr, F.

(A) What is the heat loss if the strips are spaced W_I = 8 in. (0.667 ft) apart?

(B) How is the heat loss changed if the spacing changes to W_I = 4 in., 6 in., 12 in., 16 in.?

In order to answer these two questions, the values of the f's must be determined. The parameter f_3 is not influenced by W_I, and therefore has a constant value, f_3 = 0.394. The values of f_1, f_2, f_4 change with W_I, as shown in Table 3.

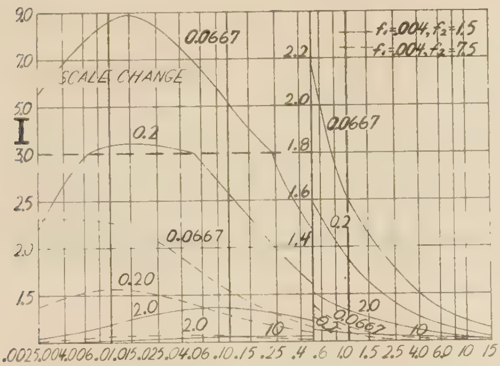

FIG. 9 INCREASE FACTOR I PLOTTED AGAINST f_4 FOR VARIOUS VALUES OF f_3

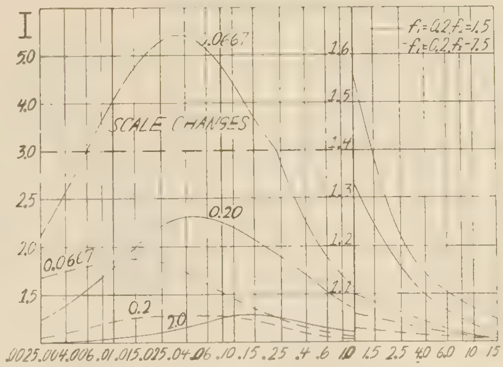

FIG. 11 INCREASE FACTOR I PLOTTED AGAINST f_4 FOR VARIOUS VALUES OF f_3

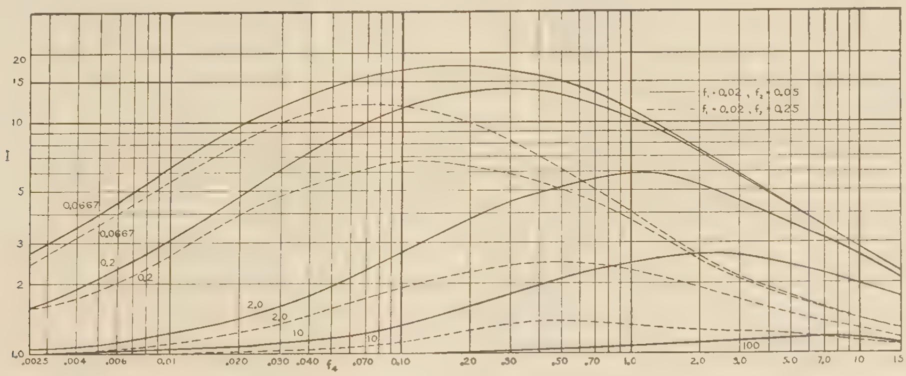

FIG. 10 INCREASE FACTOR I PLOTTED AGAINST f_4 FOR VARIOUS VALUES OF f_3

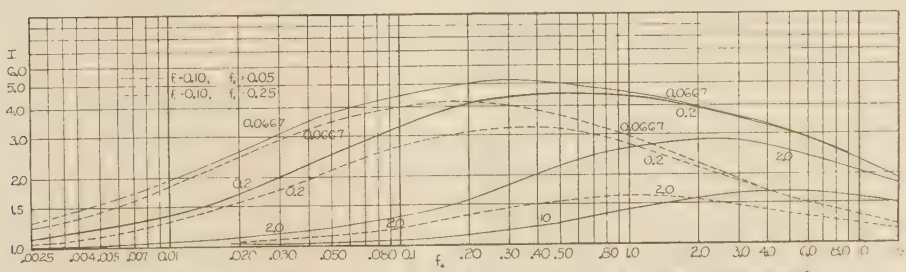

FIG. 12 INCREASE FACTOR I PLOTTED AGAINST f_4 FOR VARIOUS VALUES OF f_3

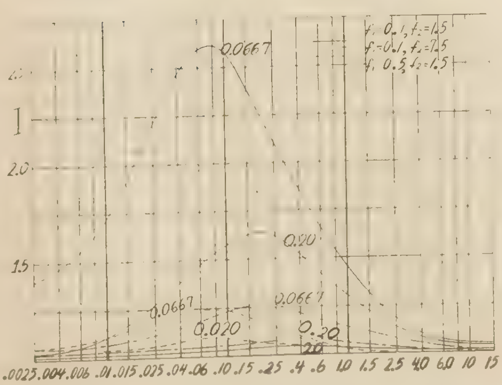

FIG. 13 INCREASE FACTOR I PLOTTED AGAINST f_4 FOR VARIOUS VALUES OF f_3

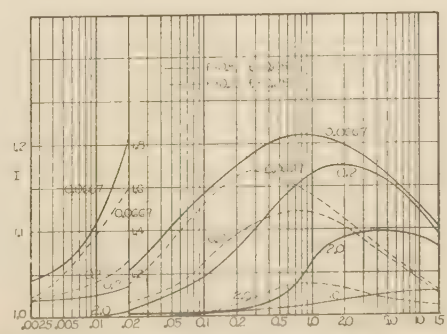

FIG. 14 INCREASE FACTOR I PLOTTED AGAINST f_4 FOR VARIOUS VALUES OF f_3

TABLE 3 VALUES OF PARAMETERS f_1, f_2, AND f_4

W_I	f_1	f_2	f_4
4	0.0156	0.5	0.12
6	0.0104	0.75	0.08
8	0.0078	1.0	0.06
12	0.0052	1.5	0.04
16	0.0039	2	0.03

The values for f_4 can be read directly on the abscissa of the charts; the values of the other f's, however, cannot be read directly. Therefore, the I values for these f's have to be found by interpolation. This interpolation is shown here for $W_I = 8$ in. The values of I for $f_3 = 0.394$ are not directly available in the charts. In order to interpolate, cross-curves have to be drawn (see Fig. 15). Values of f_3 are plotted as abscissas, and I as ordinates. Three curves (full lines) are drawn, each for a

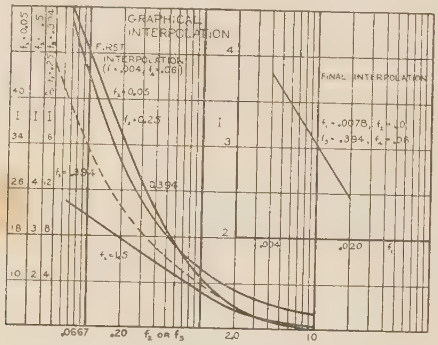

FIG. 15 GRAPHICAL INTERPOLATION

different value of f_2. (Each of the three curves has a different ordinate scale; for curve f_2 use the scale marked $f_2 = 0.05$, etc.) All curves are for $f_1 = 0.004$.

The values of I for $f_3 = 0.394$ (abscissa) are now read as follows:

	I for $f_3 = 0.394$
Curve $f_2 = 0.05$	22.7
Curve $f_2 = 0.25$	8.6
Curve $f_2 = 1.5$	2.45

The value of f_2 in the example ($W_I = 8$ in.) is $f_2 = 1$. For this value interpolation is again necessary. Because $f_2 = 0.25$ and $f_2 = 1.5$ are too far apart for accurate interpolation, a curve with all three values is drawn (dashed curve); f_2 is plotted on the abscissa, I on the ordinate. At $f_2 = 1$, the value of $I = 3.8$ is read on the ordinate. This is the increase factor I for $f_1 = 0.004$ $f_2 = 1.0$, $f_3 = 0.394$, and $f_4 = 0.06$.

A similar chart (not included in this paper) may now be drawn for $f_1 = 0.02$. From this chart, the value of I for $f_1 = 0.02$, $f_2 = 1$, $f_3 = 0.394$, $f_4 = 0.06$ is found to be 2.45.

Now the final interpolation is made (see insert in Fig. 15); f_1 is plotted as abscissa, while the ordinates give values of I for $f_2 = 1$, $f_3 = 0.394$ and $f_4 = 0.06$. By interpolating between the values for $f_1 = 0.004$ and $f_1 = 0.020$, the value for $f_1 = 0.0078$ can be read; $I = 3.22$.

This means that due to lateral heat flow, the heat loss for the structure described in case (A) is 3.22 times as high as if there were no lateral flow.

In order to determine the actual heat loss, the heat flow through the metal and through the insulation have to be calculated

$$\text{Metal heat flow } \frac{(570 - 70)^1/_8 \times {}^1/_{12}}{\dfrac{8}{12 \times 26} + \dfrac{1}{2}} = 9.90 \text{ Btu per hr}$$

$$\text{Insulation } \frac{(570 - 70) \times 8 \times {}^1/_{12}}{\dfrac{8}{12 \times 0.08} + {}^1/_2} = 37.6 \text{ Btu per hr}$$

$$\text{Total} \quad 9.9 + 37.6 = 47.5 \text{ Btu per hr}$$

The actual heat loss is $47.5 \times 3.08 = 146$ Btu per hr

The influence of the spacing (case B) has been studied by similar interpolations. In order to compare the results, it seemed advisable to reduce all of them to a width of 1 ft; inasmuch as the length (of the infinite structure) under consideration is also 1 ft, the results then apply to 1 sq ft. The reduction is carried out by dividing the heat loss by ($W_I + 2W_M$), where W_I and W_M are expressed in feet.

The results are shown in Fig. 16, in which spacing W_I is plotted as abscissa. Two ordinate scales are used; one for heat flow in Btu/sq ft, hr, the other (dimensionless) for I. Three curves are shown: the full line is for I; the dotted line is for the heat loss without lateral flow (sum of flow through metal and through insulation); the broken line is the actual heat loss with lateral heat flow.

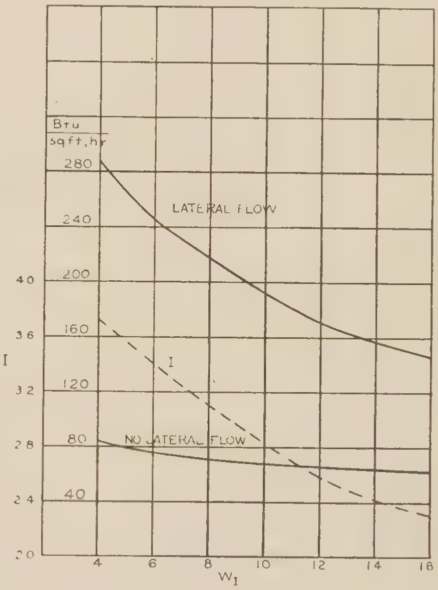

FIG. 16 VARIATION OF HEAT LOSS AND INCREASE FACTOR WITH SPACING OF STRIPS; $h = 2$

Example 2: What is the influence of the cold-side film conductance for the wall described in Example 1? How do the heat losses change if $h_0 = 1.5$, 2.5, or 3.5 instead of 2 as in Example 1?

Changing the film conductance influences only f_4; nevertheless all the interpolations have to be repeated. The results are given in Fig. 17, which shows the actual heat loss and the increase factor, both plotted as ordinates against the spacing W_I of the strips. Each curve is for a different value of h indicated on the curve. Changing the film conductance within the range of the example has relatively small influence on either the heat loss or the increase factor; more than doubling the value of h (3.5 instead of 1.5) increases the heat loss only something like 20 to 35 per cent. Of course conditions may be different for a different value of k_I or L.

Example 3: What is the influence of the thickness of insulation on increase factor and heat loss? What is the influence of the thermal conductivity of the insulation?

The results for this example are shown in Fig. 18. The ex-

ample has been carried through for a spacing of the strips, $W_I = 8$ in.; thickness of the metal, $W_M = {}^1/_{32}$ in.; conductivity of the metal, 26 Btu/ft, hr, F; film conductance, $h = 2$ Btu/sq ft, hr, F.

Thickness L of the insulation is plotted as abscissa. The ordinates have two scales, i.e., one for the heat losses per square foot (Btu/sq ft, hr), one dimensionless for I. The broken lines show the heat loss without lateral heat flow, the full lines the heat loss with lateral heat flow, and the dashed lines the increase factor. There are two curves of each type, i.e., one for a conductivity of the insulation $k_I = 0.08$, the other for $k_I = 0.05$. The values of the conductivities are noted on the curves. The increase factor for $k_I = 0.05$ goes through a maximum, while the corresponding curve of total heat loss does not show a maximum.

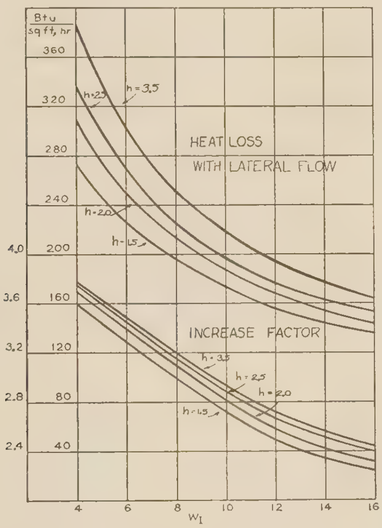

FIG. 17 VARIATION OF HEAT LOSS AND INCREASE FACTOR WITH SPACING OF STRIPS W_I FOR VARIOUS VALUES OF BOUNDARY CONDUCTANCE h

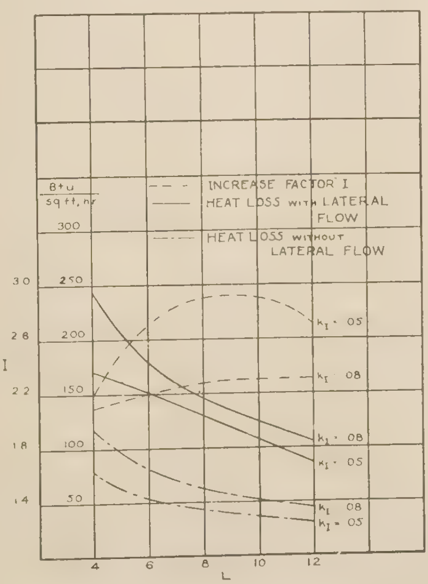

FIG. 18 VARIATION OF HEAT LOSS AND INCREASE FACTOR WITH THICKNESS L, OF INSULATION FOR DIFFERENT THERMAL CONDUCTIVITIES k_I OF THE INSULATION

Conclusions and Verifications. The extremely high values of the increase factor look alarming. Some preliminary tests indicate that the larger part of the additional heat loss can be attributed to the influence of the steel enclosure on both surfaces of the insulators. This should be further investigated, and (in further tests) if it should prove to be true, important design factors may result, e.g., insulation inserted between the strips and the covering sheets would do much to decrease the loss. From these considerations it also follows that a contact resistance between insulation and through-metal, or insulation and covering sheets, would not affect the results materially. Further investigations to verify this point would be desirable.

By way of a general confirmation as to the order of magnitude of the increase factor, reference is made to an article by J. G. Bergdoll (1),[3] in which the author states that in insulating a strato chamber it has been found that, due to thermal short circuits, the actual heat loss was 4 times as high as expected for the conductivity and thickness of the insulation.

SUMMARY

1 The heat flow through a structure composed of an insulation covered by sheet metal and interspersed with metal strips (through-metal) is discussed.

2 The cause of "lateral heat flow" occurring at any plane or level parallel to the surface of the structure, is explained and related to the cold-side film conductance.

3 An "increase factor" is defined as the ratio of the actual heat loss, including lateral heat flow, to the heat flow that would occur if no lateral heat flow existed.

4 The factors influencing the increase factor are discussed.

5 General curves giving the increase factor for various conditions are presented.

6 The use of these general curves is explained by means of examples. The examples show in a readily understandable way the influence of the thermal conductivity of the insulation, its thickness, the spacing of the metal strips, and the film conductance in specific cases.

7 In two Appendixes which follow the experimental technique and the mathematical deductions for the paper are presented.

Appendix 1

ELECTRIC ANALOGY METHOD

Ohm's law for the flow of electricity through a conductor is expressed by

$$e = ir \quad [5]$$

where

e designates voltage
i designates current
r designates resistance

Similarly, Fourier's law for steady-state heat flow through a body can be written

$$t = Q \cdot R \quad [6]$$

where

t designates temperature difference
Q designates rate of heat flow
R designates thermal resistance

Both the electrical resistance and the thermal resistance are

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

proportional to the length L of the path traveled by the medium (current or heat) and inversely proportional to the area S of flow; both are proportional to the resistivity (ρe and ρt) of the material involved (in case of heat flow, it is more common to use the reciprocal value of the resistivity, namely, the thermal conductivity, $k = \frac{1}{\rho t}$). Therefore, r and R can be expressed by

$$r = \rho e \cdot \frac{L}{A} \quad \text{[7a]}$$

$$R = \frac{1}{k} \cdot \frac{L}{A} \quad \text{[7b]}$$

From the mathematical identity of Equations [5] and [6] and that of Equations [7a] and [7b], it follows that thermal phenomena can be studied by carrying out certain electrical measurements. (Incidentally, the method is widely inclusive and can be used to analyze the transient state as well as the steady state.)

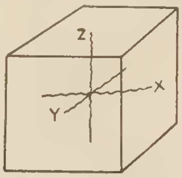

Fig. 19 View of Lump With Resistors Representing Three-Dimensional Flow

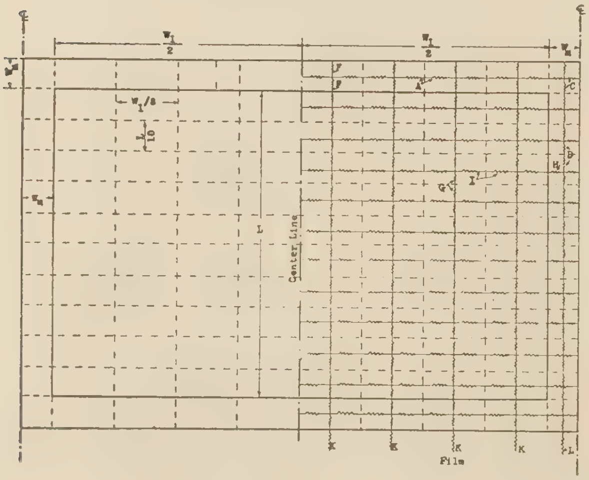

Fig. 20 Diagram Showing Sectioning Layout and Resistors

The equipment at Columbia University, known as the "heat and mass flow analyzer," has been especially devised to handle complex problems of the transient state. The method and the laboratory have been described elsewhere (2). As is evident from this paper, it can be usefully employed for analyzing complicated steady-state problems.

One method of studying steady-heat flow by electric analogy consists in setting up a body which is geometrically identical to the body subjected to heat flow and filling this body with a semiconductor. Electrical measurements then disclose the heat flow, even in odd shapes. Langmuir (3) probably was the first to use this method.

For the present experiments, the analyzer, primarily designed for transient flow, has been used. In this method the body which is to be analyzed is thought of as cut into a number of sections or lumps. The resistance in each of the three directions is considered concentrated in the three axes, as indicated by the isometric sketch, Fig. 19.

The structure being of infinite length in the Y direction, there can be no heat flow in the Y direction, and the problem reduces to one involving only two dimensions.

Thus a network is built up having "horizontal" resistors for flow in the direction X, and "vertical" resistors for flow in the direction Z. Fig. 20 shows the wiring diagram. Because of symmetry, it is sufficient to build up a circuit representing $W_I/2$ and W_M. L has been divided into 10 sections, and $W_I/2$ into four sections. Fig. 20 shows the geometrical meaning of the sections. It is a cross section, showing the same view as the isometric sketch, Fig. 1, with the limits of the sections noted. The film conductance on the cold surface is represented by additional resistors. The letters at each resistor refer to Appendix 2, where the deduction of the dimensionless units is discussed.

A word of comparison with the other electrical method (with geometrical similarity) may not be amiss. The "lumping" method used here is probably more complicated than the "similarity" method. With only 115 sections available, the "lumping" is rather crude. (Previous experiments (4) have shown, however, that this does not affect the accuracy to any appreciable extent.) One great advantage, on the other hand, may be claimed for the "lumping" method; it is easily possible to study separately the influence of the thermal conductivities of the various materials involved.

Appendix 2

Derivation of Dimensionless Units

If two different thermal arrangements lead to the same circuit, they have the same value I. If, once a network is established, all resistors are changed in the same ratio, e.g., all resistors are doubled or halved, the outcome of the experiments is not changed; merely the "scale factor" between Ohm's law and Joule's law is different.

The first step then in deriving the dimensionless units consists in writing the equations for all resistors, Equations [9] to [17]. Resistors having the same letter always have the same value and are changed in value only after the individual test is over.

The next step consists in forming resistor ratios. Here all resistors were compared with R_I as "standard." Any of the other resistors (R_A, R_B, etc.) would have done just as well; all resistors, however, must be compared with the same "standard" (R_I), Equations [18] to [25]. If in these equations certain variables always appear in the same grouping, then this group may be taken as one parameter. Thus the four parameters were empirically evolved. In writing Equations [9] to [18], a value z has been used; z is the "scale factor" transforming heat units into electrical units. The scale factor z in (megohms × Btu/hr, F) ties in each individual resistor with the thermal resistance it represents. For example, in the case of resistance R_G; the thermal resistance R_{Gt} for this section is expressed by

$$R_{Gt} = \frac{8}{10} \frac{L}{k_I \cdot W_I \cdot 1} \quad \text{[8]}$$

This equation is easy enough to find. The total thickness L is divided into 10 sections; one section has for length of the heat path only $L/10$; similarly, the width is divided into 8 sections. Hence, the area is proportional to $W_I/8$. A length in the direction Y of 1 ft is considered. If L and W_I are expressed in feet, k_B in Btu/ft, hr, F, then R_{Gt} is expressed in F, hr/Btu. In

order to obtain the resistance R_G in megohms, R_{Gt} has to be multiplied by z

$$R_G = \frac{8 \cdot z \cdot L}{10 \cdot k_I \cdot W_I} \qquad [9]$$

Similarly the other Equations [10] to [17], for the resistors have been found.

$$R_I = \frac{10 \cdot z \cdot W_I}{8 \cdot k_I \cdot L} \qquad [10]$$

$$R_A = \frac{1 \cdot z \cdot W_I}{8 \cdot k_M \cdot W_M} \qquad [11]$$

$$R_D = \frac{1 \cdot z \cdot L}{10 \cdot k_M \cdot W_M} \qquad [12]$$

$$R_L = \frac{z}{h \cdot W_M} \qquad [13]$$

$$R_K = \frac{8 \cdot z}{h \cdot W_I} \qquad [14]$$

$$R_C = \frac{z \cdot W_M \cdot 1}{k_M \cdot 2 \cdot W_M} = \frac{z}{2k_M} \qquad [15]$$

$$R_H = \frac{z \cdot W_M \cdot 10}{k_M \cdot 2 \cdot L} \qquad [16]$$

$$R_F = \frac{z \cdot W_M \cdot 8}{k_M \cdot 2 \cdot W_I} \qquad [17]$$

Now by comparing each of these resistors with R_I it can readily be found that the variables always appear in given ratios, and only in these ratios. They therefore are selected as parameters. They were given in Equations [4], which, for convenience, are repeated here

$$f_1 = \frac{W_M}{W_I};\ f_2 = \frac{W_I}{L};\ f_3 = \frac{k_I \cdot L}{k_M \cdot W_M};\ f_4 = \frac{k_I}{h \cdot W_I} \qquad [4]$$

With these parameters, equations for various resistors read

$$R_G = 0.64 \cdot \frac{R_I}{f_2^2} \qquad [18]$$

$$R_A = 0.1 \cdot f_3 \cdot R_I \qquad [19]$$

$$R_D = 0.08 \frac{f_3}{f_2} \cdot R_I \qquad [20]$$

$$R_L = 0.8 \frac{f_4}{f_1 \cdot f_2} \cdot R_I \qquad [21]$$

$$R_C = 0.4 \cdot f_1 \cdot f_3 \cdot R_I \qquad [22]$$

$$R_H = 4 \cdot f_1^2 \cdot f_2 \cdot f_3 \cdot R_I \qquad [23]$$

$$R_F = 3.2 \cdot f_1^2 \cdot f_3 \cdot R_I \qquad [24]$$

$$R_k = \frac{6.4 f_4}{f_2} \cdot R_1 \qquad [25]$$

The four parameters used are by no means the only ones possible. There are certainly other combinations available. The validity of the parameters chosen was checked twice; once by "dimensional analysis" (5), which shows possible selections of parameters. A second practical test was made by setting up one set of conditions. Then one value, say W_I, was changed. Following Equations [4] this means a change of W_M and L if f_1 to f_4 are to remain unchanged, which in turn calls for other changes. By systematically changing the conditions and corresponding circuits, one must return to the original circuit if the parameters are to be correct. The method of finding the parameters, as just described, is generally applicable in working with the heat and mass flow analyzer, and is in many cases very promising. For specific cases, it yields answers avoiding lengthy and extended mathematical analysis. In other cases of course it would be easier to determine the (dimensionless) parameters mathematically and arrange the experiments accordingly.

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Discussion

C. E. Ernst.[4] The influence of through-metal in limiting the effectiveness of insulation has long been realized, but this paper presents what is apparently the first concrete information on the extent of this influence. A true appreciation of the importance of such information can be gained from a consideration of the economics involved.

Thermal insulation for any specific condition should be applied in a thickness consistent with its cost and efficiency and the cost of heat-energy loss. In other words, the economic thickness is that thickness at which the sum of the cost of insulation per year and the cost of heat-energy loss is a minimum.

Through-metal, however, has the effect of reducing the efficiency of insulation, and the greater the amount of through-metal the less the thickness of insulation that can be justified. In other words, if the amount of through-metal is sufficient to nullify the effect of the insulation, the elimination of the insulation can be economically justified.

While, for the present, construction methods apparently do not allow complete elimination of thermal short-circuiting paths through insulation, common sense tells us that it is certainly of economic advantage to reduce them to a minimum.

Herein lies the importance of the present work which is a good start in the right direction. Further work along these lines should lead to better design and construction methods for the incorporation of thermal insulation into all kinds of heated and refrigerated equipment with the efficiency of the insulation in place approaching that of the material alone, while maintaining the required mechanical strength of the structure.

G. M. Dusinberre.[5] The practical use of the authors' curves is considerably restricted by the lack of generality in the basic assumptions. For example, one of the metal walls will generally be designed for structural strength and will be rela-

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[5] Department of Mechanical Engineering, Virginia Polytechnic Institute, Blacksburg, Va. Now on duty at U. S. Naval Academy, Annapolis, Md. Mem. A.S.M.E.

tively thick, while the other wall will merely serve as a retainer for the insulation and will be relatively thin. The through-metal will often, as in ship construction, serve a structural purpose in stiffening the strength bulkhead, and the web thickness will be related to the thickness of this bulkhead.

Another serious lack of generality lies in the assumption that the surface resistance can be neglected on one side. There are a great many practical cases where this cannot be done.

Fortunately, the engineer who has to grind out an answer for this type of problem has at his disposal the greater flexibility, generality, speed, and lucidity of the numerical method outlined by Emmons.[6]

The writer worked the authors' Example 1A by this method, using the same assumptions except that the temperature gradient in the thin direction of the metal was neglected.[7] The time required was 2 hr for a 4-in. and a 2-in. network spacing. Additional solutions on the 2-in. spacing would take about $^1/_2$ hr each. This is probably more rapid than the authors' method, considering the intricate interpolation procedure which they describe.

In the numerical method, we work directly in temperatures, and trends are readily recognized. The authors note that the opposite is true when working with parameters. In the numerical method we acquire, incidentally, some other information of value, such as the maximum cold-surface temperature. This is a matter which insulation engineers are sometimes called upon to guarantee.

The temperature-distribution pattern was found to be as indicated in Fig. 21. This pattern is reversed on either side, by symmetry.

The heat flow can be calculated from the various layers. For example, from the cold surface to ambient, we have a mean temperature difference of 123 F. The width of a unit panel is $^2/_3$ ft and the surface coefficient is 2. Then $123 \times {}^2/_3 \times 2 = 164$ Btu per hr. Checking with the second layer in Fig. 21, we have

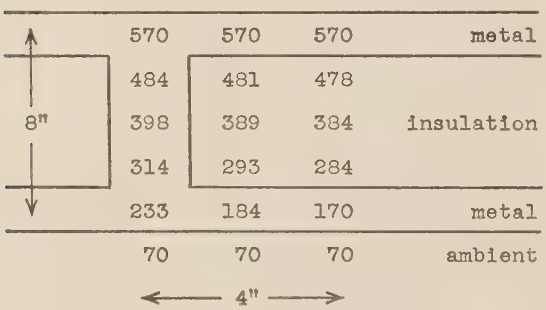

Fig. 21

a temperature difference of 84 F in the metal, and a mean temperature difference of 94 F in the insulation. Then $84 \times 26 \times 6 \times {}^1/_{96} = 137$ Btu per hr for the steel, and $94 \times 0.08 \times 6 \times {}^2/_3 = 30$ Btu per hr for the insulation, a total of 167 Btu per hr. The hot-side layer figures 169 Btu per hr, which is close enough to give the answer as 167 ± 3 Btu per hr.

This does not agree with the authors' figure of 146 Btu per hr. If the authors would publish a temperature-distribution diagram for this example, it might indicate where the discrepancy lies.

H. G. Elrod, Jr.[8] For the conditions outlined by the authors, the heat flow per section may be obtained from the following formula

$$Q = \frac{t_h - t_c}{\dfrac{L}{k_I} + \dfrac{1}{h}}\left[W_I + \frac{k_M}{k_I}(2W_M) - \left\{ \frac{(2W_M)}{L\,W'_M}\left(1 - \frac{k_I}{k_M}\right) \frac{\dfrac{\tanh \alpha \dfrac{W_I}{2}}{\alpha} + \dfrac{k_M}{k_I}(W_M)}{\alpha \tanh \alpha \dfrac{W_I}{2} + \beta W_M} \right\} \right]$$

where

$$\alpha^2 = \frac{\dfrac{k_I}{L} + h}{k_M W_M'}$$

and

$$\beta = \frac{\dfrac{k_M}{L} + h}{k_M W_M'}$$

W_M' the thickness of the metal walls need not be one half of the thickness of the through-metal strips.

The foregoing formula was derived by neglecting the temperature drop in the thin direction of the metal, and the lateral heat flow in the insulation. The first assumption is obviously allowable; the second assumption, because it arbitrarily restricts the heat flow, should cause the formula to yield values of heat flow smaller than actually found. Nevertheless, the formula values consistently exceed those shown by the authors in Fig. 17. On the other hand, the formula agrees well with the numerical method. The following is a comparison between formula and numerical method for the authors' example (1*a*):

	Formula	Numerical method	Authors' value
Over-all heat flow, Btu per hr....	165	167 ± 3	146
Cold-surface temperatures,[a] deg F	168	170	
	184	184	
	238	233	

[a] Obtained using a temperature-distribution formula based upon precisely the same assumptions as used for the formula given in this discussion.

The writer's formula can be applied to the case where the metal-wall thicknesses and film coefficients are the same on both sides of the insulation. For this case, L should stand for one-half of the thickness of the insulation, and $t_h - t_c$ should be replaced by one half the total temperature difference.

The solution for cylindrical through-metal (bolts and pipes) has already been illustrated.[9]

Authors' Closure

Mr. Ernst stresses the fact that through-metal greatly increases the heat losses of insulated structures, and that the value of insulation is therefore decreased. Examination of this problem shows that the economic thickness of insulation is greater for structures with thermal short circuits than for those without. Quite apart from any numerical examination, pure reasoning can prove that this must be so. With increasing thickness, the influence of the thermal short circuit decreases; therefore, greater thickness results in a smaller increase factor. However,

[6] "The Numerical Solution of Heat-Conduction Problems," by H. W. Emmons, Trans. A.S.M.E., vol. 65, 1943, p. 607–615.

[7] This gradient works out to be on the order of 0.01 deg F in the present example. It seems surprising that the authors used two resistances in series to subdivide such a relatively small quantity.

[8] Department of Marine Engineering, U. S. Naval Academy, Annapolis, Md. Jun. A.S.M.E.

[9] "Applied Mathematics in Chemical Engineering," by T. K. Sherwood and C. E. Reed, McGraw-Hill Book Company, Inc., New York, N. Y., 1939, p. 207.

no increase in thickness within practical limits can offset the increased losses due to through-metal. Therefore, every care should be taken in design of equipment to avoid through-metal as far as possible. After the limit has been reached, the economic thickness of the insulation should be determined, and this can best be done by using curves such as are contained in this paper.

With regard to the example checked by Messrs. Dusinberre and Elrod, it should be noted that in the paper an increase factor of 3.08 was inadvertently introduced in place of the actual value 3.22 found from the graphical interpolations. If 3.22 is used, as should have been done, the heat loss becomes $3.22 \times 4.75 = 153$ Btu per hr. This is about 7 per cent less than the loss calculated by the discussers, and the difference must be attributed to the many steps in the interpolation. This was verified by setting up the network for this specific case and measuring the heat loss, which was found to be approximately 165 Btu per hr.

In specific reply to Mr. Dusinberre's comments it can be agreed that the present curves do not cover all cases which he mentions as important. If only one case at a time has to be solved, the use of the numerical method may be helpful, if no network or analyzer is quickly available. If, however, a larger number of such cases have to be solved, the development of curves would be desirable, because the practicing engineer will rarely find the time for the calculations entailed by the numerical method.

Mr. Dusinberre's statement that in the numerical method trends are more readily recognized than when working with parameters is misleading. The statement that trends are not readily seen when working with parameters refers to the trends due to the individual variables, as conductivity, spacing, etc. Such trends cannot be found by solving only one problem, either by the numerical method or on the analyzer. They can be detected only by solving a number of problems sufficient to cover the range it is desired to investigate. The best presentation of a large number of tests is by dimensionless parameter as used in this paper.

The maximum cold-surface temperatures can be found on the analyzer without additional work. They could be presented in graphs similar to those shown in the paper.

Network spacings of 2 in. and 4 in., as used by Mr. Dusinberre, are sufficient in the present example. In many instances, they will not yield accurate results, in which case the work involved by the numerical method increases considerably.

In his footnote,[7] Mr. Dusinberre overlooks the fact that the network, as shown in Fig. 20, refers not only to the example discussed by him but also to broader cases in which the subdivision as shown is necessary because the temperature gradient in the thin direction of the metal cannot be neglected.

Mr. Elrod's formula is interesting, and it would be desirable that he publish proof. The assumptions made are rather broad for certain conditions, and the value of the equation would be enhanced if Mr. Elrod would determine the limits (perhaps expressed in f_1, f_2, f_3, and f_4) within which his formula can be safely used.

Formulas for Calculating the Temperature Distribution in Electrical Coils of General Rectangular Cross Section

By T. J. Higgins,[1] Chicago, Ill.

In a recent paper (1)[2] Prof. Max Jakob outlined a procedure for calculating the temperature at any point T, the maximum temperature T_m, and the average temperature T_a in an electrical coil maintained at uniform surface temperature. Mr. Jakob derived explicit formulas for certain simple coils, i.e., the circular cylindrical coil, the infinite-plate coil, the spherical coil. The present paper is devoted to effecting formulas for T, T_m, and T_a for the very important case of the toroidal coil of general rectangular cross section. Following Jakob, the procedure is to pose determination of T as a boundary-value problem in the mathematical theory of heat; effect the formal solution for T; and obtain the corresponding formulas for T_m and T_a by use of their known relationship with T. Illustrative of application of the general formulas derived in the present paper, the maximum and average temperatures in a coil of given dimensions are calculated and are found to be in excellent agreement with the known measured values.

1 General Remarks

A current-carrying toroidal coil of rectangular cross section, composed of many turns of insulated wire of circular or of rectangular cross section, is an important component of many electrical devices. Commonly, the coil is either exposed to the atmosphere, e.g., the field coils of a dynamo, or is submerged in a liquid, e.g., the coils of a transformer. Usually, conditions are such that the temperature distribution in the coil is substantially independent of the angular co-ordinate of a system of cylindrical co-ordinates, the axis of which is the axis of the coil. If so, the temperature distribution is two-dimensional, i.e., it is the same over any cross section of the coil. Only this case will be considered.

In use, the heat generated in the coil results in a general temperature rise throughout the coil. In consequence, a point of maximum temperature is to be found in the cross section of the coil. By virtue of the assumed angular symmetry, this maximum temperature is the maximum temperature in the coil. Because the permissible maximum coil current is that which produces a specified maximum temperature in the coil, and because the "hot" or "operating" resistance of the coil depends upon, and can be calculated from, the average (mean) temperature of the coil, formulas for predetermining the temperature distribution, maximum temperature, and average temperature are of considerable interest to the designer of toroidal coils. Unfortunately, but few such formulas are available to him in the literature.

Undoubtedly, this paucity of formulas stems from the analytical difficulties posed by the various factors which affect the temperature distribution in the coil. In the main, the temperature at an arbitrary point in the coil is determined by (*a*) the geometry of the coil, (*b*) the surface conditions, (*c*) the current density in the metal, (*d*) the temperature variations of the thermal conductivities of the insulation and of the metal, and (*e*) the temperature variation of the electrical resistivity of the metal. If each of these quantities is specified analytically, determination of the temperature distribution can be set as a problem in the mathematical theory of heat. Theoretically, a rigorous solution of this problem is possible. Practically, such a solution is mathematically intractable.

Thus for specification of even the simplest conditions for items (*b*) to (*e*), i.e., uniform surface temperature and uniform current density, thermal conductivities and electrical resistivity independent of the temperature, the writer is unacquainted with any analysis, rigorous or approximate, wherein the actual toroidal geometry of the coil is taken into account. The mathematical analysis prerequisite to such a solution is so severe that, perforce, those who have worked on the problem have limited themselves to analyzing coils of simple geometry in which the temperature distribution approximates (so it is thought) that in certain special shapes of toroidal coils.

For example, a rectangular cylinder (i.e., an infinite cylinder of rectangular cross section) can be considered as a torus of the same cross section and of infinite inner radius. If, then, a toroidal coil of rectangular cross section is such that the inner radius is large compared with either dimension of its cross section, the temperature distribution in it tends to that in a rectangular cylindrical coil of the same cross section.

If, further, the length of the radial side (width) of the rectangular cross section of the toroidal coil is small, compared with that of the axial side (height), the temperature distribution in both the toroidal and the cylindrical coils tends to that in the geometrically simpler plate coil (i.e., a cylindrical coil with cross section an infinite strip of definite width). Accordingly, if the toroidal coil tends to either of these two limiting shapes, it can be conjectured that an approximate formula for the temperature distribution in the toroidal coil is afforded by the formula for the corresponding rectangular cylindrical or plate coil.

However, for these simpler coils, as for the more complex toroidal coil, the author is unacquainted with any rigorous solution for the temperature distribution. Neither is it to be expected that any exists. For even if temperature variations in the thermal conductivities and electrical resistivity be neglected, in a rigorous solution (as a problem in the mathematical theory of heat), account must yet be taken (*a*) of the discrete distribution of current density over the cross section, zero in the insulation, practically uniform in the metal, and (*b*) of the boundary conditions at the interfaces between the insulation and the metal, continuity of the temperature and of the normal component of heat-flux density. While it is not difficult to take the first factor into account, the second renders a rigorous solution mathematically intractable.

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[1] Associate Professor of Electrical Engineering, Illinois Institute of Technology.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., Nov. 29–Dec. 3, 1943, of The American Society of Mechanical Engineers.

Aware, explicitly or intuitively, of this intractableness, yet desirous of obtaining some approximation to the rigorous formulas for temperature distribution in these simpler coils, those who have worked at this problem have resorted, perforce, to various artifices. These latter comprise two categories, as follows:

1 Physical considerations serve as bases for the deduction of empirical formulas for the maximum and the average temperatures.

2 Formulas for the temperature distribution, maximum temperature, and average temperature in a homogeneous infinite plate or a homogeneous rectangular cylinder, subject to certain conditions of surface temperature and internally generated heat, are affected rigorously by application of the mathematical theory of heat. It is then implied that these formulas can be utilized either directly, in conjunction with certain measurements, or in combination with empirically calculated physical "constants," to calculate the corresponding temperatures in plate or in rectangular cylindrical coils.

In one or the other of these fashions formulas have been effected by Binder (2), Humburg (3), Jakob (1, 4), Ott (5), Punga (6), Rogowski (7), and Vidmar (8). The most recent, most inclusive, and most applicable work is that of Jakob (1). His analysis encompasses an important factor neglected by the others, i.e., the temperature variation of the resistivity of the conducting metal.[3] This factor has a decided effect upon the temperature distribution in the coil.

We epitomize Jakob's analysis:[4] If a toroidal coil is of such inner radius that it can be approximated by a cylindrical coil of the same cross section, and if the toroidal coil is composed of many turns of light wire, then:

1 We identify the temperature distribution T, the maximum temperature T_m, and the average temperature T_a for the coil with the corresponding quantities in a homogeneous cylinder of the following characteristics:

(a) Of the same cross section as the coil.

(b) Of "equivalent" thermal conductivity k (calculated as outlined in Section 4).

(c) Maintained at the same surface conditions as the coil.

(d) And wherein q, the heat generated per unit volume per unit time, is $q_0[1 + \alpha_0 T]$; T_s is the surface temperature; α_0 the temperature coefficient of resistivity of the conducting metal at 0 deg C; and $q_0 = I^2R_0/V$, I being the coil current, V the volume of the coil, and R_0 the coil resistance when the entire coil is at temperature 0 deg C.

2 And proceed to determine T, T_m, and T_a for the homogeneous cylinder as a problem in the mathematical theory of heat.

For the boundary condition of uniform surface temperature, Jakob obtains formulas for T, T_m, and T_a for the one-dimensional problems of the circular cylinder, and the infinite plate of definite thickness (and also the sphere). These formulas obtained, Jakob suggests that the formulas for T, T_m, and T_a for the circular cylinder can be used to approximate the corresponding quantities in the toroidal coil of square cross section, and that those for the infinite plate can be used similarly for toroidal coils of quite elongated rectangular cross section. In support of this suggestion and illustrative of his analysis in general, the formulas for T_m and T_a for the circular cylinder are used to calculate T_m and T_a in the toroidal coil of square cross section utilized in the experiments of Rogowski and Vieweg (11). The values thus calculated are found to be in substantial agreement with the known measured values.

Now most rectangular toroidal coils are neither of square nor of quite elongated rectangular cross section. In consequence, it is most desirable to have formulas for T, T_m, and T_a that are applicable for the general rectangular cross section. Such formulas are derived in Sections 2 to 4. Illustrative of their use and substantiative of their correctness, the formulas for T_m and T_a are then utilized to calculate these temperatures in the same coil as that used by Jakob in his numerical example. Seemingly, no other experimental data are available. Excellent agreement is obtained between the calculated and the measured values.

2 Solution of the Associated Heat Problem

The axis of a homogeneous infinite rectangular cylinder of thermal conductivity k watts cm^{-1} deg C^{-1} is coincident with the z axis of a Cartesian system of co-ordinates, the sides of the right cross section are parallel to the x and y axes (Fig. 1). The surface of the cylinder is maintained at uniform temperature T_s deg C. Heat per unit volume per unit time is developed in the cylinder at the rate of $q_0(1 + \alpha_0 T)$ watts cm^{-3}, q_0 and α_0 being constant and T a function of x and y alone. Required: The expressions for the temperature $T(x, y)$ at any point in the cylinder, the maximum temperature T_m, and the average temperature T_a.

From the mathematical theory of heat, the expression for T is that solution of the partial-differential equation

$$\partial^2T/\partial x^2 + \partial^2T/\partial y^2 = -q_0(1 + \alpha_0 T)/k \qquad [1]$$

which satisfies the given boundary condition

$$T = T_s \qquad [2]$$

at

$$x = \pm a, \quad y = \pm b$$

and the condition stemming from the symmetry of the problem

$$T(x, y) = T(\pm x, \pm y) \qquad [3]$$

By virtue of Equation [3], the temperature T is an even function of x and of y. Accordingly, T can be expressed by the double Fourier cosine series

$$T = A_0 + \sum_{i=1}^{\infty} \sum_{j=1}^{\infty} A_{ij} \cos M_i x \cos N_j y \qquad [4]$$

in which A_0, A_{ij}, M_i, and N_j are constants to be determined from Equations [1] and [2]. Substituting Equation [4] in Equation [2], we find Equation [4] satisfies Equation [2] providing

$$M_i = \pi(2i - 1)/2a \qquad i = 1, 2, 3 \qquad [5]$$

$$N_j = \pi(2j - 1)/2b \qquad j = 1, 2, 3 \qquad [6]$$

and

$$A_0 = T_s \qquad [7]$$

We determine A_{ij} from Equation [1] in the usual fashion. Substituting Equation [4] in [1]

[3] Jakob (1) remarks, and it is also so mentioned in the discussions of Davis and Tramontini appended to his paper, that his analysis encompasses the effects of the temperature variation of the thermal conductivity. In the opinion of the author, this is not so. For if the thermal conductivity k were a function of the temperature θ, then the correct form of Equation [5] of Jakob's paper would be $\partial(k\, \partial\theta/\partial x)/\partial x = -q'''$, whence assumption of some analytical expression of k as a function of θ gives rise to a more or less complicated differential equation. The correct interpretation of the constants in Equations [5] and [6] of Jakob's paper is that k is constant and q''' is a linear function of temperature $q_s(1 + \epsilon_s\theta)$, q_s a constant, and ϵ_s a temperature coefficient of resistivity at s deg C. Hence Equation [6] becomes $q'''/k = m(1 + \epsilon\theta) = (q_s/k)(1 + \epsilon_s\theta)$. Note that in his numerical example Jakob himself identifies ϵ with the temperature coefficient of electrical resistivity. For further discussion of the temperature variation of thermal conductivity see Section 4 of the present paper.

[4] This is the writer's interpretation as deduced from study of the numerical example in Jakob's paper. For a qualitative discussion of the correctness of this procedure see Section 4 of the present paper.

$$\sum_{i=1}^{\infty} \sum_{j=1}^{\infty} A_{ij}(M_i^2 + N_j^2 - C^2) \cos M_i x \cos N_j y = C' \quad [8]$$

in which

$$C^2 = q_0\alpha_0/k;\ C' = q_0(1 + \alpha_0 T_s)/k;\ C'/C^2 = T_s + \alpha_0^{-1} \quad [9]$$

multiplying each member of Equation [8] by $\cos M_p x \cos N_q y$, p, and q arbitrary positive integers; integrating over a quadrant of the cross section shown in Fig. 1

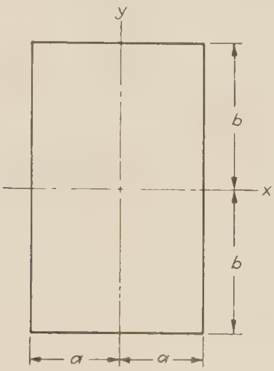

FIG. 1

$$\sum_{i=1}^{\infty} \sum_{j=1}^{\infty} A_{ij}(M_i^2 + N_j^2 - C^2) \int_0^a \int_0^b \cos M_i x \cos M_p x$$

$$\cos N_j y \cos N_q y \, dy\, dx = 4C' \int_0^a \int_0^b \cos M_p x \cos N_q y \, dy\, dx \quad [10]$$

evaluating the double integrals (all integrals on the left-hand side vanish except those for which $p = i$, $q = j$); simplifying by use of Equations [5] and [6]; and solving for A_{ij} yields

$$A_{ij} = (4C'/ab)(-1)^{i+j}/M_i N_j (M_i^2 + N_j^2 - C^2) \quad [11]$$

Substituting Equations [7] and [11] in Equation [4], we have

$$T = T_s + (4C'/ab) \sum_{i=1}^{\infty} \sum_{j=1}^{\infty} (-1)^{i+j} \cos M_i x \cos N_j y / M_i N_j (M_i^2 + N_j^2 - C^2) \quad [12]$$

We can transform Equation [12] into forms more convenient for numerical computation by use of the identities[5]

$$(2/b) \sum_{j=1}^{\infty} (-1)^{j-1} \cos N_j y / N_j (N_j^2 + K_i^2) = (1/K_i^2)(1 - \cosh K_i y \operatorname{sech} K_i b) \quad [13]$$

in which $K_i^2 = M_i^2 - C^2$ and K_i is the positive root; and

$$(2/a) \sum_{i=1}^{\infty} (-1)^{i-1} \cos M_i x / M_i K_i^2 \left.\begin{aligned} &= (1/C^2)(1 - \cosh Cx \operatorname{sech} Ca),\ C^2 < 0 \\ &= (-1/C^2)(1 - \cos Cx \sec Ca),\ C^2 > 0 \end{aligned}\right\} \quad [14]$$

[5] "Sur une méthode de sommation de certaines series de Fourier," by E. Roth and G. Kouskoff, *Revue Gènèrale de l'Electricité*, vol. 23, 1928, pp. 1061–1073.

Substituting Equation [13] in Equation [12] yields

$$T = T_s + (2C'/a) \sum_{i=1}^{\infty} (-1)^{i-1} \cos M_i x (1 - \cosh K_i y \operatorname{sech} K_i b)/K_i^2 M_i \quad [15]$$

Simplifying Equation [15] by use of the second form of Equation [14], we have

$$T = T_s - (C'/C^2)(1 - \cos Cx \sec Ca) - (2C'/a) \sum_{i=1}^{\infty} (-1)^{i-1} \cos M_i x \cosh K_i y / K_i^2 M_i \cosh K_i b \quad [16]$$

in which $C^2 > 0$.

By virtue of the symmetry of Equation [12], an alternative form of Equation [16] can be obtained by interchanging a and b, x and y, M_i and N_j, K_i and $K_j (= N_j^2 - C^2)$. As this is easily done, we shall not write down this alternative form.

The maximum temperature T_m occurs at the center of the cross section. Substituting $x = 0$, $y = 0$ in Equation [16], we have

$$T_m = T_s - (C'/C^2)(1 - \sec Ca) - (2C'/a) \sum_{i=1}^{\infty} (-1)^{i-1}/K_i^2 M_i \cosh K_i b \quad [17]$$

The average temperature T_a over the cross section is given by

$$\bar{T}_a = (1/4ab) \int_{-a}^{a} \int_{-b}^{b} T\, dy\, dx \quad [18]$$

Substituting Equation [16] in Equation [18], and integrating, yields

$$T_a = T_s - (C'/C^2)(1 - \tan Ca/Ca) - (2C'/a^2 b) \sum_{i=1}^{\infty} (-1)^{i-1} \sin M_i a \tanh K_i b / K_i^3 M_i^2 \quad [19]$$

3 Special Cases of the General Heat Problem

Certain special cases of the general problem are of particular interest.

Rectangular Cylinder and Uniform Heat Density. If $\alpha_0 = 0$, whence $C^2 = 0$, q is uniform throughout the cylinder. Taking the limit as $C^2 \to 0$ of the right-hand members of Equations [16], [17], and [19] (the details are contained in the Appendix), yields

$$T = T_s + C'(a^2 - x^2)/2 - (2C'/a) \sum_{i=1}^{\infty} (-1)^{i-1} \cos M_i x \cosh M_i y / M_i^3 \cosh M_i b \quad [20]$$

$$T_m = T_s + C'a^2/2 - (2C'/a) \sum_{i=1}^{\infty} (-1)^{i-1}/M_i^3 \cosh M_i b \quad [21]$$

$$T_a = T_s + C'a^2/3 - (2C'/a^2 b) \sum_{i=1}^{\infty} (-1)^{i-1} \sin M_i a \tanh M_i b / M_i^5 \quad [22]$$

This case was studied by Humburg (3), and later by Jakob (4). The latter's formulas for T, T_m, and T_a are less compact than the author's; they are expressed in terms of double-infinite

series whereas the present ones are expressed in terms of single-infinite series. The two forms are equivalent, however, in that Equations [20], [21], and [22] can be transformed into the corresponding Equations [17], [19], and [21] of Jakob's paper (1).

Infinite Plate and Nonuniform Heat Density. As $b \to \infty$, the rectangular cross section tends to an infinite strip of thickness $2a$, whence the rectangular cylinder tends to an infinite plate of thickness $2a$. Substituting $b = \infty$ in the right-hand members of Equations [16], [17], and [19], yields

$$T = T_s - (C'/C^2)(1 - \cos Cx \sec Ca) \qquad [23]$$

$$T_m = T_s - (C'/C^2)(1 - \sec Ca) \qquad [24]$$

$$T_a = T_s - (C'/C^2)(1 - \tan Ca/Ca) \qquad [25]$$

Jakob (1) has recently considered this case in detail. Corroborative of the present analysis, Equations [23], [24], and [25] are identical (aside from notation) with the corresponding Equations [14], [15], and [16] of Jakob's paper (1).

Infinite Plate and Uniform Heat Density. Substituting $b = \infty$ in Equations [20], [21], and [22], yields

$$T = T_s + C'(a^2 - x^2)/2 \qquad [26]$$

$$T_m = T_s + C'a^2/2 \qquad [27]$$

$$T_a = T_s + C'a^2/3 \qquad [28]$$

This case was studied first by Rogowski (7). The present equations for T, T_m, and T_a are identical with his.

4 Application of the Heat Problem to Electrical Coils

The formulas of Sections 2 and 3 can be utilized to calculate temperatures in electrical coils, provided we interpret q_0, α_0, and k as deduced in this section.

Inasmuch as the wire comprising an electrical coil is insulated, the same current I flows through any cross section of the wire. If R_0 is the resistance of the coil when its temperature is uniform at 0 deg C, the Joulean power developed in the coil at this temperature is I^2R_0; whence the average power per unit volume is I^2R_0/V. Now if the coil were composed of wire of zero thickness of insulation, I^2R_0/V would be identical with $q_0(= i^2\rho_0)$(i being the current density, and ρ_0 the resistivity at 0 deg C of the metal), the power per unit volume developed at any point in the coil. If the coil were composed of thinly insulated wire, the average of q_0 over the sectioned metal and insulation of a cross section of the wire would closely approximate I^2R_0/V. Accordingly, if we restrict ourselves to coils composed of thinly insulated wire (and commonly the wire is thus insulated), it is rational to identify q_0 in the equations of Sections 2 and 3 with I^2R_0/V.

In use, both the temperature T and the power density q vary from point to point in the coil. If, in addition, we assume that the coil is composed of wire of cross section small in comparison with that of the coil,[6] which is the case if (as is usual) the coil is composed of many turns of light wire, the current density i over any cross section of the (metal of the) wire is substantially uniform; whence variation in $q(= i^2\rho)$ from point to point in the coil must be attributed to variation in the resistivity of the metal from point to point. As a function of temperature $\rho = \rho_0(1 + \alpha_0 T)$; from which we derive $q = q_0(1 + \alpha_0 T)$. Accordingly, we identify α_0 in the equations of Sections 2 and 3 with the temperature coefficient of resistivity at 0 deg C of the metal of the coil.

The thermal conductivities k_m and k_i of the metal and of the insulation of a coil are quite different. Commonly, k_m is several thousand times greater than k_i. Accordingly, k in the formulas of Sections 2 and 3 is to be identified with the "equivalent thermal conductivity" of the coil.[7] The equivalent thermal conductivity of an array of parallel circular cylindrical metallic conductors embedded in a homogeneous insulation can be determined from curves due to Richter (9). These curves are reproduced in a text on electrical design by Moore (10).

If, as is frequently the case, the wire is of square or of not too elongated rectangular cross section, Richter's curves can yet be used through the device of replacing the rectangular cross section by an "equivalent" circular cross section of the same area. Alternatively, the areas of sectioned metal and insulation, comprising the cross section of a coil, can be taken as comprising a network of thermal conductances; these conductances are combined into a single equivalent thermal conductance in the same fashion as a series-parallel network of electrical conductances is reduced to a single equivalent conductance; and the equivalent conductivity k determined therefrom. This method of calculating k is made use of in Jakob's paper (1).

In general, despite the large ratio of k_m/k_i, k is only several times larger than k_i. Accordingly, if we were to attempt to take the variation of the thermal conductivities of k_i and k_m into account, it would be rational to do so by identifying the variation in k with that in k_i. Few quantitative data are available, however, on the temperature variations of the thermal conductivities of insulating materials. Nor, for that matter, are the k_i values known accurately at any particular temperature. In consequence although it is possible to effect a solution for the heat problem of Section 2, in which k as well as q is taken as a linear function of the temperature (in fact, the writer has done so), such a refinement as applied to calculation of the temperature in electrical coils would be meaningless.

In the light of the discussion to this point, we may expect that if a coil comprises many turns of thinly insulated wire of comparatively small cross section, and if the constants q_0, α_0, and k are interpreted as outlined in this section, the formulas of Sections 2 and 3 will yield fairly accurate values of temperature. Comparatively few experimental data are available for affording, through comparison of known test values with calculated values, either confirmation or disproval of this statement. Seemingly, the only available published data suitable to this purpose are those of Rogowski and Vieweg (11). Certain of their data were used by Jakob (1) to illustrate use of the formulas and procedure advanced in his paper. The author utilizes these same data in the numerical example of Section 5.

5 Numerical Example

We consider the following problem: To calculate the maximum temperature attained in the rectangular toroidal coil during Run 2 of the experiments of Rogowski and Vieweg (11). This particular run is selected in the thought that the surface condition apposite to the formulas derived in this paper, i.e., uniform surface tem-

[6] If the cross section of the wire is large, the current density will be nonuniform over the cross section. For, since the resistivity of different filaments of the wire will vary with the temperature variation over the cross section, and since all filaments are connected in parallel, whence the voltage gradient ϵ along each filament must be the same, the current density $i = \epsilon/\rho$ will necessarily vary over the cross section. Thus in the case of a single heavy circular cylindrical conductor, the current density is a minimum at the center of the cross section, at which point the temperature is greatest.

[7] Our reason for doing so is as follows: In various problems having to do with the flow of heat in materials which are not homogeneous from point to point but which are homogeneous in a "fine-grained" sense (a nonhomogeneous pattern of small area is regularly ranked to form a comparatively large area, thus the cross section of the coil is composed of a regular array of identical nonhomogeneous cross sections of the wire), experience is to the effect that utilization of such an equivalent thermal conductivity permits satisfactory analytical formulation of an otherwise intractable problem. Thus, heat-flow problems in cellular materials are handled in precisely this fashion.

perature, is more closely approximated in Run 2 than in the other runs. In Run 2 the coil is cooled on all sides by an air blast.

Given Data: Toroidal coil with square cross section; inner radius 4 cm; winding composed of 1416 turns of 0.1-cm-diam copper wire insulated with lacquer-soaked silk; these turns are arranged in coaxial layers, each layer composed of 40 turns; surface temperature 76.1 C; maximum temperature 122.4 C; mean temperature calculated from the "hot" or operating resistance 96.7 C; cold resistance ≈ 11 ohms; coil current 3 amp.

Derived Data: Using the given data, we find the volume V of the coil to be 754 cu cm. The percentage conductivity of the copper wire is not stated. Taking it to be 100 per cent, we find (as can be found in any electrical handbook) that $\alpha_0 \approx 0.00423$ deg C^{-1} and $\alpha_{20} = 0.00393$ deg C^{-1}. The temperature at which the "cold" resistance ≈ 11 ohms was measured is not stated. We assume 20 deg C (68 F). Then $R_0 = 11[1 - 0.00393 \times 20]$ 10.13 ohms; whence $q_0 = I^2R_0/V = 0.1207$ cm^{-3}. Jakob (1) has calculated the equivalent conductivity of the coil to be k = 0.00531 watt cm^{-1} deg C^{-1}.

Using these derived values of α_0, q_0, and k, we find by direct computation

$C^2 = 0.0963$; $C' = 29.9$; $C'/C^2 = 310.6$; $Ca = 0.683$
$M_1 = 0.714$; $M_1{}^2 = 0.5098$; $K_1{}^2 = 0.4135$; $K_1 = 0.643$; $K_1b = 1.415$
$M_2 = 2.142$; $M_2{}^2 = 4.588$; $K_2{}^2 = 4.492$; $K_2 = 2.119$; $K_2b = 4.651$

These values were calculated on a 20-in. slide rule. Substituting appropriately in Equations [17] and [19], we have

$$T_m = 76.1 + 89.8 - (42.2 - 0.11) = 123.8 \text{ deg C}$$

$$T_a = 76.1 + 58.8 - (136.7 - 0.13) = 98.3 \text{ deg C}$$

The terms within the parentheses are those contributed by the infinite series in Equations [17] and [19]. We note that these series converge so rapidly that generally it is necessary to compute only the first term; the second term is certainly much smaller than the error inherent in the general procedure.

Comparing these calculated values with the corresponding measured values,[8] T_m = 122.4 deg C and T_a = 96.7 deg C, we find differences of 1.4 deg C and 1.6 deg C. These differences are 1.15 per cent and 1.65 per cent, respectively, of T_m and T_a.

Considering the various assumptions made in Section 4, and the uncertainties in the given data, this substantial agreement between calculated and measured values is most gratifying. For, although success in this one numerical example hardly constitutes a general validification of the formulas given in this paper and although it is most desirable to obtain additional and more precise experimental data for use in effecting similar comparisons, it is substantiative of the author's opinion that the formulas of this paper will yield satisfactory numerical values of the temperature in rectangular toroidal coils, providing the coil geometry and surface conditions are such that application of these formulas is justifiable.

ACKNOWLEDGMENT

The author is indebted to Prof. M. Jakob for calling attention to the subject of this paper, and for personal discussion of certain points of his previous paper (1).

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[8] Jakob (1) states that the measured value was 100 deg C (= 76.1 + 23.9), but Rogowski and Vieweg definitely state 96.7 deg C for T_a.

Appendix

To obtain the limit as $C^2 \to 0$ of the right-hand member of Equation [16], we note that

$$\lim_{C^2 \to 0} K_i = M_i$$

and that

$$\lim_{C^2 \to 0} (C'/C^2)(1 - \cos Cx \sec Ca)$$

$$= \lim_{C^2 \to 0} (C'/C^2)[1 - (1 - C^2x^2/2! + C^4x^4/4! - \ldots)(1 - C^2a^2/2! + C^4a^4/4! - \ldots)^{-1}]$$

$$= \lim_{C^2 \to 0} (C'/C^2)[1 - 1 + C^2(x^2 + a^2)/2! - C^4(a^4/4! + x^4/4! + a^2x^2/2\cdot 2!) + \ldots]$$

$$= C'(x^2 + a^2)/2!$$

Taking the limit as $C^2 \to 0$ of the right-hand member of Equation [16] and making use of the foregoing evaluations yields Equation [20].

In similar fashion, Equation [22] follows from Equation [19] by making use of the identity

$$\lim_{C^2 \to 0} (C'/C^2)(1 - \tan Ca/Ca)$$
$$= \lim_{C^2 \to 0} (C'/C^2)[1 - (Ca + C^3a^3/3 + 2C^5a^5/15 + \ldots)/Ca]$$
$$= C'a^2/3$$

Discussion

Victor Paschkis.[9] Electrical coils are usually operated at constant voltage. Thus, if for reasons of poor design the temperature and therewith the electrical resistance would increase considerably, the current would drop.

It would be of considerable interest if the author's investigation could be extended to include the case of constant voltage, and a current initially not known but dependent upon the resulting temperatures.

In discussing the paper orally, Prof. L. M. Boelter mentioned some desirable refinements which may make the problem mathematically unsolvable. If this should be the case, investigation on

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the "heat and mass-flow analyzer" might offer a solution. The desirability was mentioned of determining the apparent thermal conductivity of laminated materials, such as cores of transformers, etc. Such determination would also be highly desirable in connection with heat flow in strip coils and staples of metal sheets. For this purpose a temperature range up to 1600 F or 1800 F should be investigated.

Knowledge of the thermal conductivity under these conditions would make possible a rational analysis of uniformity in heat-treating practice.

Friction Factors for Pipe Flow

By LEWIS F. MOODY,[1] PRINCETON, N. J.

The object of this paper is to furnish the engineer with a simple means of estimating the friction factors to be used in computing the loss of head in clean new pipes and in closed conduits running full with steady flow. The modern developments in the application of theoretical hydrodynamics to the fluid-friction problem are impressive and scattered through an extensive literature. This paper is not intended as a critical survey of this wide field. For a concise review, Professor Bakhmeteff's (1)[2] small book on the mechanics of fluid flow is an excellent reference. Prandtl and Tietjens (2) and Rouse (3) have also made notable contributions to the subject. The author does not claim to offer anything particularly new or original, his aim merely being to embody the now accepted conclusions in convenient form for engineering use.

In the present pipe-flow study, the friction factor, denoted by f in the accompanying charts, is the coefficient in the Darcy formula

$$h_f = f \frac{L}{D} \frac{V^2}{2g}$$

in which h_f is the loss of head in friction, in feet of fluid column of the fluid flowing; L and D the length and internal diameter of the pipe in feet; V the mean velocity of flow in feet per second; and g the acceleration of gravity in feet per second per second (mean value taken as 32.16). The factor f is a dimensionless quantity, and at ordinary velocities is a function of two, and only two, other dimensionless quantities, the relative roughness of the surface, $\frac{\epsilon}{D}$ (ϵ being a linear quantity in feet representative of the absolute roughness), and the Reynolds number $\mathbf{R} = \frac{VD}{\nu}$ (ν being the coefficient of kinematic viscosity of the fluid in square feet per second). Fig. 1 gives numerical values of f as a function of $\frac{\epsilon}{D}$ and $\mathbf{R}$.

Ten years ago R. J. S. Pigott (4) published a chart for the same friction factor, using the same co-ordinates as in Fig. 1 of this paper. His chart has proved to be most useful and practical and has been reproduced in a number of texts (5). The Pigott chart was based upon an analysis of some 10,000 experiments from various sources (6), but did not have the benefit, in plotting or fairing the curves, of later developments in functional forms of the curves.

In the same year Nikuradse (7) published his experiments on artificially roughened pipes. Based upon the tests of Nikuradse and others, von Kármán (8) and Prandtl (9) developed their theoretical analyses of pipe flow and gave us suitable formulas with numerical constants for the case of perfectly smooth pipes or those in which the irregularities are small compared to the thickness of the laminar boundary layer, and for the case of rough pipes where the roughnesses protrude sufficiently to break up the laminar layer, and the flow becomes completely turbulent.

The analysis did not, however, cover the entire field but left a gap, namely, the transition zone between smooth and rough pipes, the region of incomplete turbulence. Attempts to fill this gap by the use of Nikuradse's results for artificial roughness produced by closely packed sand grains, were not adequate, since the results were clearly at variance from actual experience for ordinary surfaces encountered in practice. Nikuradse's curves showed a sharp drop followed by a peculiar reverse curve,[3] not observed with commercial surfaces, and nowhere suggested by the Pigott chart based on many tests.

Recently Colebrook (11), in collaboration with C. M. White, developed a function which gives a practical form of transition curve to bridge the gap. This function agrees with the two extremes of roughness and gives values in very satisfactory agreement with actual measurements on most forms of commercial piping and usual pipe surfaces. Rouse (12) has shown that it is a reasonable and practically adequate solution and has plotted a chart based upon it. In order to simplify the plotting, Rouse adopted co-ordinates inconvenient for ordinary engineering use, since f is implicit in both co-ordinates, and $\mathbf{R}$ values are represented by curved co-ordinates, so that interpolation is troublesome.

The author has drawn up a new chart, Fig. 1, in the more conventional form used by Pigott, taking advantage of the functional relationships established in recent years. Curves of f versus $\mathbf{R}$ are plotted to logarithmic scales for various constant values of relative roughness $\frac{\epsilon}{D}$; and to permit easy selection of $\frac{\epsilon}{D}$, an accompanying chart, Fig. 2, is given from which $\frac{\epsilon}{D}$ can be read for any size of pipe of a given type of surface.

In order to find the friction loss in a pipe, the procedure is as follows: Find the appropriate $\frac{\epsilon}{D}$ from Fig. 2, then follow the corresponding line, thus identified, in Fig. 1, to the value of the Reynolds number $\mathbf{R}$ corresponding to the velocity of flow. The factor f is thus found, for use in the Darcy formula

$$h_f = f \frac{L}{D} \frac{V^2}{2g}$$

In Fig. 2, the scales at the top and bottom give values of the diameter in both feet and inches. Fig. 1 involves only dimensionless quantities and is applicable in any system of units.

To facilitate the calculation of $\mathbf{R}$, auxiliary scales are shown at the top of Fig. 1, giving values of the product (VD'') for two fluids, i.e., water and atmospheric air, at 60 F. (D'' is the inside diameter in inches.) As a further auxiliary, Fig. 3 is given, from which $\mathbf{R}$ can be quickly found for water at ordinary temperatures, for any size of pipe and mean velocity V. Dashed lines on this chart have been added giving values of the discharge or quantity of fluid flowing, $Q = AV$, expressed in both cubic feet per second and in U. S. gallons per minute.

[1] Professor, Hydraulic Engineering, Princeton University. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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[3] Rouse, reference (3), p. 250; and Powell, reference (10), p. 174.

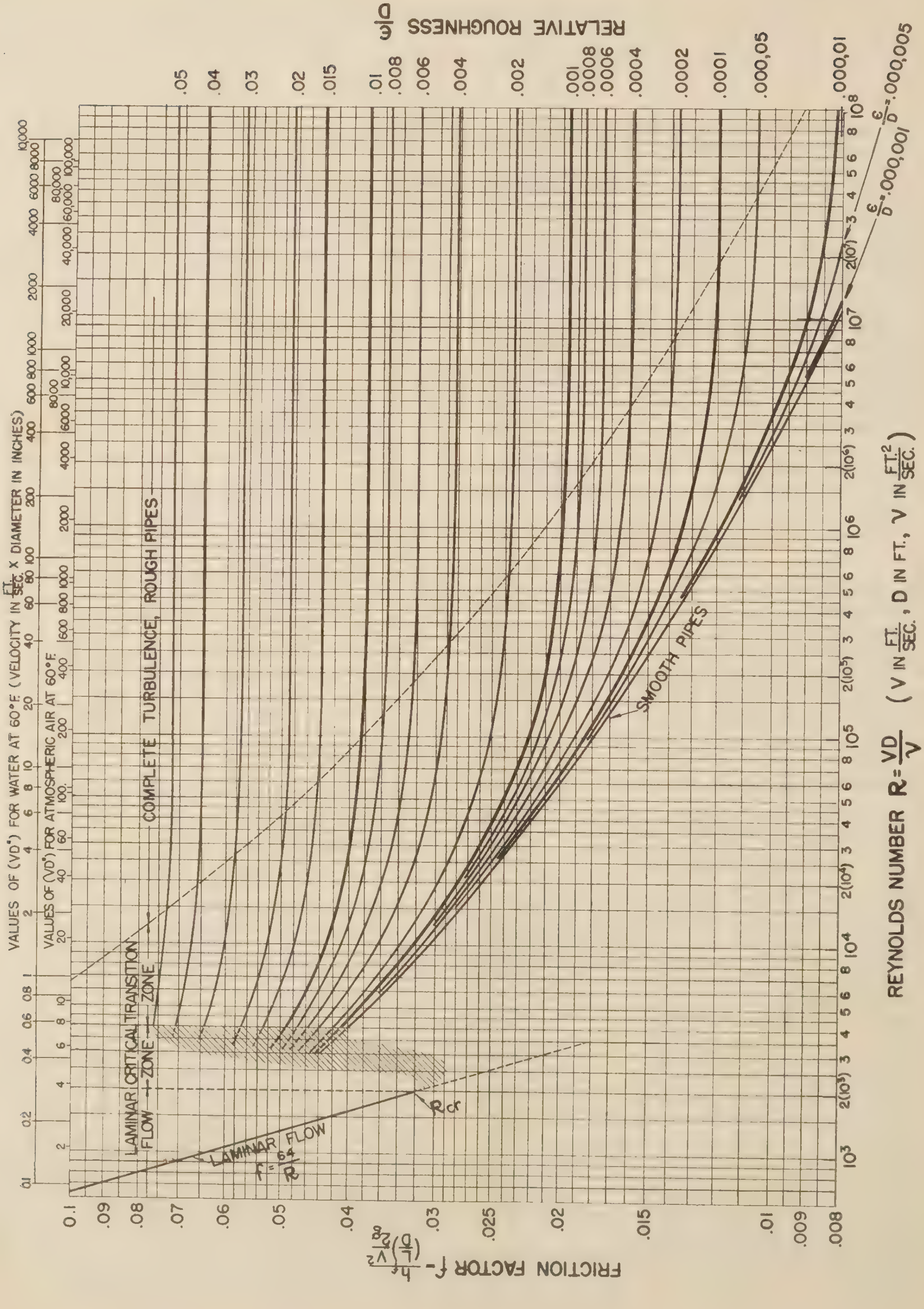
RELATIVE ROUGHNESS $\frac{\epsilon}{D}$
VALUES OF (VD") FOR WATER AT 60°F. (VELOCITY IN $\frac{FT.}{SEC.}$ X DIAMETER IN INCHES)
VALUES OF (VD") FOR ATMOSPHERIC AIR AT 60°F.
COMPLETE TURBULENCE, ROUGH PIPES
LAMINAR FLOW
CRITICAL ZONE
TRANSITION ZONE
LAMINAR FLOW
$f = \frac{64}{R}$
R_{cr}
SMOOTH PIPES
$\frac{\epsilon}{D} = .000,001$
$\frac{\epsilon}{D} = .000,005$
FRICTION FACTOR $f = \frac{h_f}{\left(\frac{L}{D}\right)\frac{V^2}{2g}}$
REYNOLDS NUMBER $R = \frac{VD}{\nu}$ (V IN $\frac{FT.}{SEC.}$, D IN FT., ν IN $\frac{FT.^2}{SEC.}$)

Fig. 1

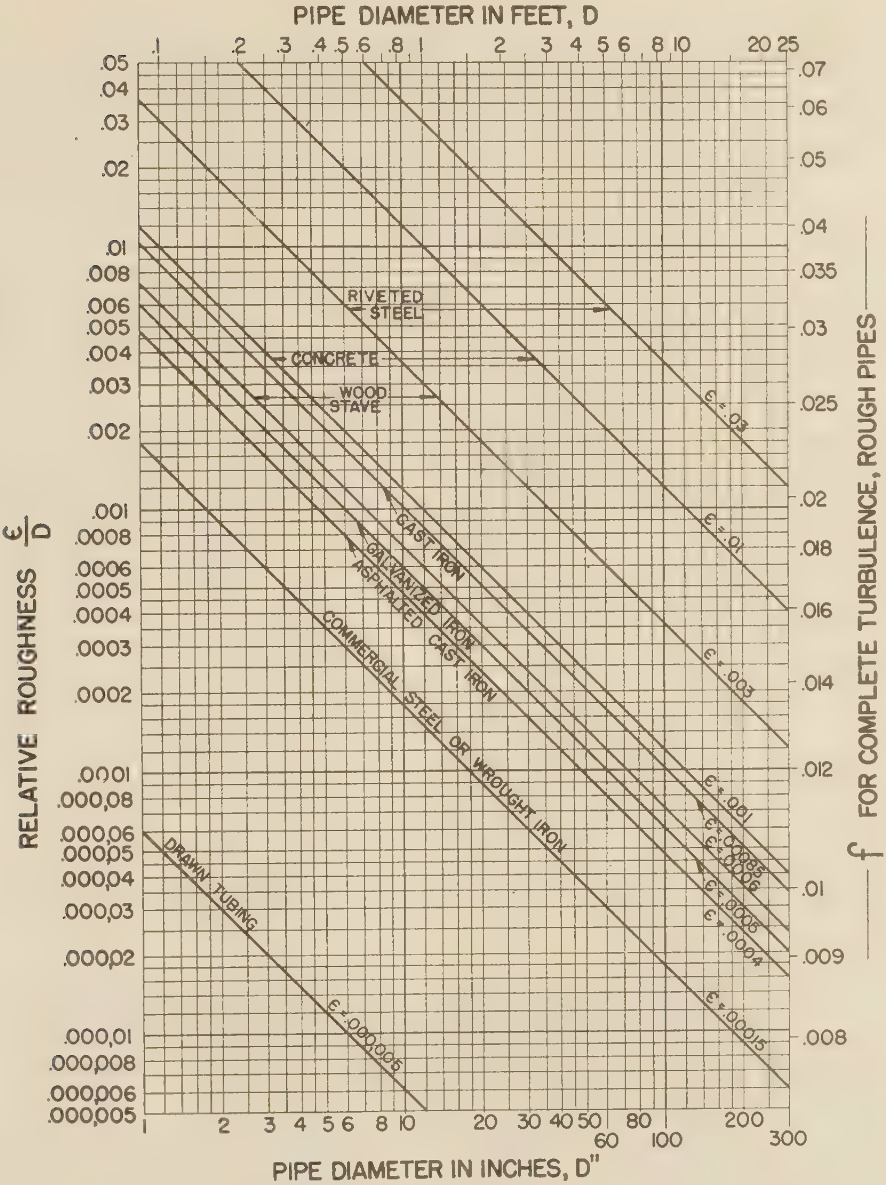

Fig. 2

For other fluids, the kinematic viscosity ν may be found from Fig. 4, which with Prof. R. L. Daugherty's kind permission has been reproduced.[4] To enable **R** to be quickly found for various fluids, Fig. 4 includes an auxiliary diagram constructed by Dr. G. F. Wislicenus, which gives **R** for various values of the product VD'' shown by the diagonal lines. For any value of ν in the left-hand diagram, by following a horizontal line to the appropriate diagonal at the right, the corresponding **R** may be read at the top of the auxiliary graph.

Over a large part of Fig. 1, an approximate figure for **R** is sufficient, since f varies only slowly with changes in **R**; and in the rough-pipe zone f is independent of **R**. From the last consideration, it becomes possible to show, in the right-hand margin of Fig. 2, values of f for rough pipes and complete turbulence. If it is seen that the conditions of any problem clearly fall in the zone of complete turbulence above and to the right of the dashed line in Fig. 1, then Fig. 2 will give the value of f directly without further reference to the other charts.

[4] Reference (13) and reference (5).

Illustration of Use of Charts

Example 1: To estimate the loss of head in 200 ft of 6-in. asphalted cast-iron pipe carrying water with a mean velocity of 6 fps: In Fig. 2, for 6 in. diam (bottom scale), the diagonal for "asphalted cast iron" gives $\frac{\epsilon}{D} = 0.0008$ (left-hand margin). In Fig. 3, for 6 in. diam (left-hand margin), the diagonal for $V = 6$ fps gives $\mathbf{R} = 2.5\ (10^5)$ (bottom scale) (or, instead of using Fig. 3, compute $VD'' = 6 \times 6 = 36$). In Fig. 1, locate from the right-

hand margin the curve for $\frac{\epsilon}{D} = 0.0008$ and follow this curve to a point above $\mathbf{R} = 2.5\ (10^5)$ on the bottom scale (or below $VD'' = 36$ on the top scale). This point gives $f = 0.02$ (left-hand margin); then

$$h_f = f\frac{L}{D}\frac{V^2}{2g} = 0.02\frac{(200)}{(0.5)}\frac{(6)^2}{64.3} = 4.5 \text{ ft friction loss}$$

Example 2: To estimate the loss of head per 100 ft in a 15-in. new cast-iron pipe, carrying water with a mean velocity of 20 fps: In Fig. 2, for 15 in. diam (bottom scale), the diagonal for "cast iron" gives $\frac{\epsilon}{D} = 0.0007$ (left-hand margin). In Fig. 3, for 15 in. diam (left-hand margin), the diagonal for $V = 20$ fps gives $\mathbf{R} = 2\ (10^6)$ (or, instead of using Fig. 3, compute $VD'' = 20 \times 15 = 300$). In Fig. 1, the curve for $\frac{\epsilon}{D} = 0.0007$ (interpolating between 0.0006 and 0.0008, right-hand margin), at a point above $\mathbf{R} = 2\ (10^6)$ (bottom scale) (or below $VD'' = 300$, top scale) gives $f = 0.018$ (left-hand margin). In this case the point on Fig. 1 falls just on the boundary of the region of "complete turbulence, rough pipes." Here $\mathbf{R}$ or VD'' need only be approximated sufficiently to see that the point falls in the complete turbulence region, and f can then be found directly from the right-hand margin in Fig. 2 without further reference to Fig. 1; then

$$h_f = f\frac{L}{D}\frac{V^2}{2g} = 0.018\frac{(100)}{(1.25)}\frac{(20)^2}{64.3}$$
$$= 8.95 \text{ or, say, } 9 \text{ ft friction loss}$$

It must be recognized that any high degree of accuracy in determining f is not to be expected. With smooth tubing, it is true, good degrees of accuracy are obtainable; a probable variation in f within about ± 5 per cent (14), and for commercial steel and wrought-iron piping, a variation within about ± 10 per cent. But, in the transition and rough-pipe regions, we lack the primary and obvious essential, a technique for measuring the roughness of a pipe mechanically. Until such a technique is developed, we have to get along with descriptive terms to specify the roughness; and naturally this leaves much latitude. The lines in Fig. 2 might be more graphically represented by broad bands rather than single lines, but this is not practical due to overlapping.

Even with this handicap, however, fairly reasonable estimates of friction loss can be made, and, fortunately, engineering problems rarely require more than this. It will be noted from the charts that a wide variation in estimating the roughness affects f to a much smaller degree. In the rough-pipe region, for the reasons just explained, the values of f cannot be relied upon within a range of the order of at least 10 per cent.

The charts apply only to new and clean piping, since the rapidity of deterioration with age, dependent upon the quality of the water or fluid and that of the pipe material, can only be guessed in most cases; and in addition to the variation in roughness there may be, in old piping, an appreciable reduction in effective diameter, making an estimate of performance speculative.

Although we have no accepted method of direct measurement of the roughness, in any case where we have a sample of pipe of the same surface texture available for test in the laboratory or in the field, then from a test of such a pipe in any size we can, by aid of the charts, find the absolute roughness corresponding to its performance. Thus we have a means for measuring the roughness hydraulically. The scale of the absolute roughness ϵ used in plotting the charts is arbitrary, based upon the sand-grain diameters of Nikuradse's experiments.

The field covered by Fig. 1 divides itself into four areas representing distinct flow characteristics. The first is the region of laminar flow, up to the critical Reynolds number of 2000. Here the flow is fully stabilized under the control of viscous forces which damp out turbulence, permitting a completely rational solution. The values of f are here given by a single curve, $f = \frac{64}{\mathbf{R}}$, independent of roughness, representing the Hagen-Poiseuille law.

Between Reynolds numbers of 2000 and 3000 or 4000, the conditions depend upon the initial turbulence due to such extraneous factors as sudden changes in section, obstructions, or a sharp-edged entrance corner prior to the reach of pipe considered;

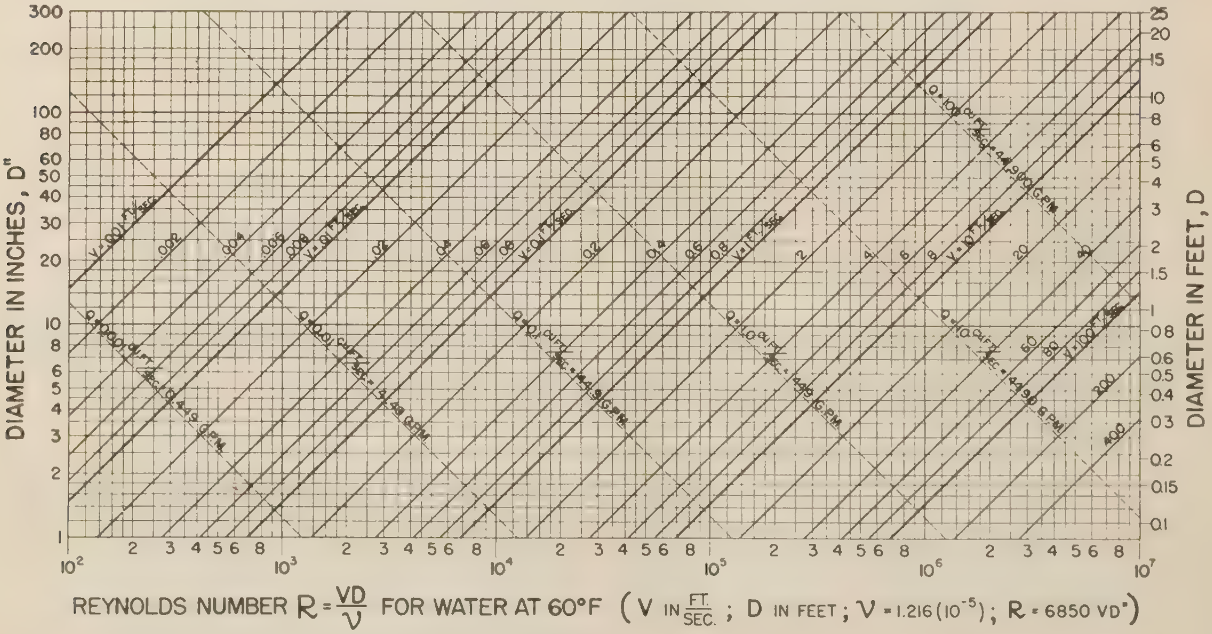

Fig. 3

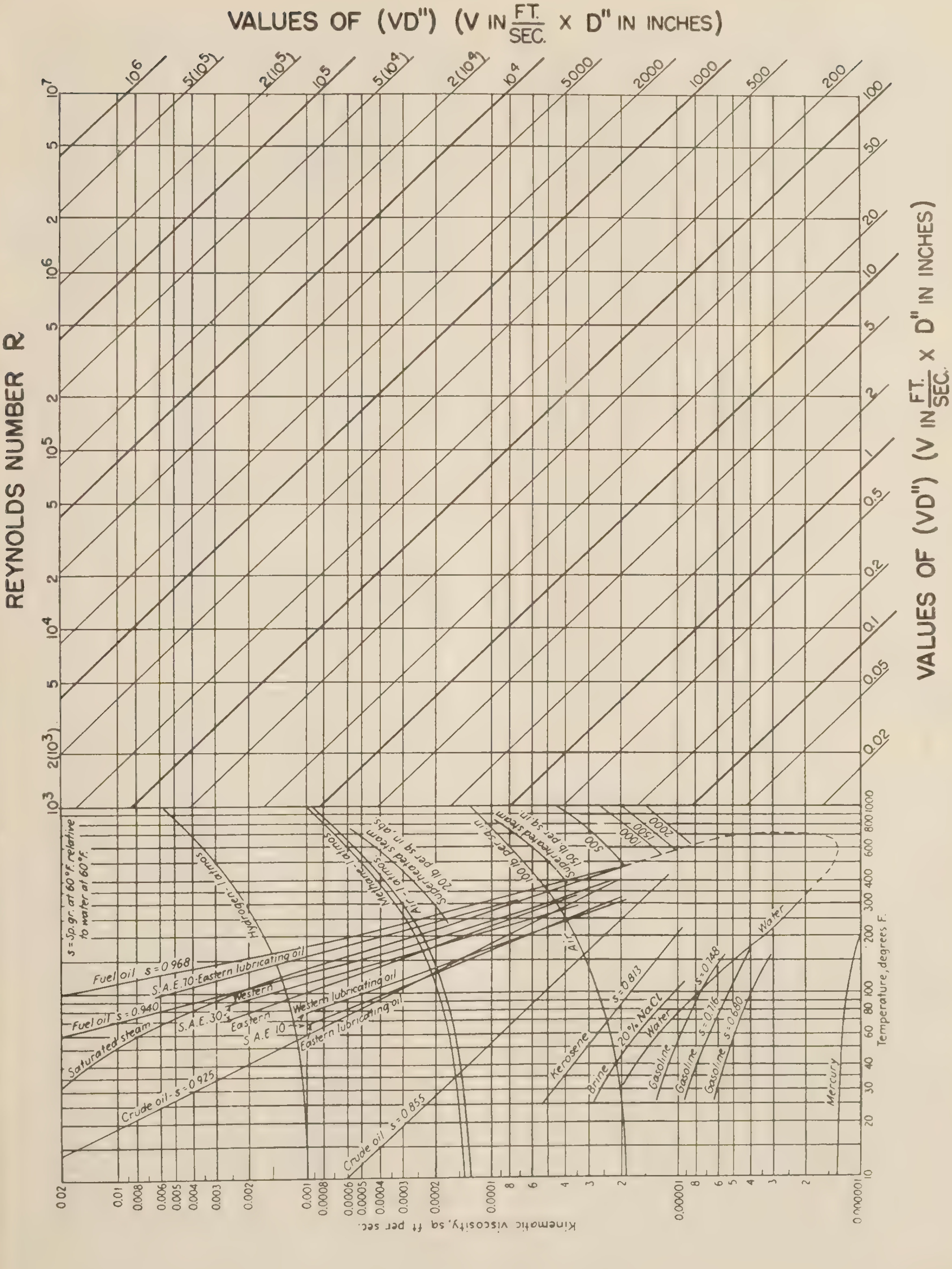
VALUES OF (VD") (V IN FT./SEC. X D" IN INCHES)
REYNOLDS NUMBER R
VALUES OF (VD") (V IN FT./SEC. X D" IN INCHES)
Kinematic viscosity, sq ft per sec.
Temperature, degrees F.
s = Sp. gr. at 60°F. relative to water at 60°F.
Fuel oil s = 0.968
S.A.E. 70-Eastern lubricating oil
Fuel oil s = 0.940
Saturated steam
S.A.E. 30
Western
Eastern
S.A.E. 10
Western lubricating oil
Eastern lubricating oil
Crude oil - s = 0.925
Crude oil s = 0.855
Hydrogen - 1 atmos.
Methane - 1 atmos.
Air - 1 atmos.
Superheated steam 20 lb per sq in. abs.
100 lb per sq in
Superheated steam 150 lb per sq in
500
1000
1500
2000
Air
Kerosene
Brine
20% NaCl
s = 0.813
Water
Gasoline
s = 0.748
Gasoline
s = 0.716
Gasoline
s = 0.680
Water
Mercury

FIG. 4

and the conditions are probably also affected by pressure waves initiating instability. This region has been called a critical zone, and the indefiniteness of behavior in this region has been indicated by a hatched area without definite f lines. The minimum limit for f values is the dotted continuation of the laminar-flow line, corresponding to very smooth and steady initial flow. When there is distinct turbulence in the entering fluid, the flow in the critical zone is likely to be pulsating (2) rather than steady. The effects of strong initial turbulence may even extend into the laminar-flow zone, raising the f values somewhat, as far as to a Reynolds number of about 1200.

Above a Reynolds number of 3000 or 4000, conditions again become reasonably determinate. Here we find two regions, namely, the transition zone and the rough-pipe zone. The transition zone extends upward from the line for perfectly smooth pipes, for which the equation is

$$1/\sqrt{f} = 2\log_{10}\frac{\mathbf{R}\sqrt{f}}{2.51};\ \text{or } 1/\sqrt{f} = 2\log\mathbf{R}\sqrt{f} - 0.8$$

(Kármán, Prandtl, Nikuradse) to the dashed line indicating its upper limit, plotted from the relation

$$\frac{1}{\sqrt{f}} = \frac{\mathbf{R}}{200}\,\frac{\epsilon}{D}$$

(following the corresponding line in Rouse's chart, reference 12). In the transition zone the curves follow the Colebrook function

$$1/\sqrt{f} = -2\log\left(\frac{\epsilon/D}{3.7} + \frac{2.51}{\mathbf{R}\sqrt{f}}\right)$$

These curves are asymptotic at one end to the smooth pipe line and at the other to the horizontal lines of the rough-pipe zone. Actually, the curves converge rapidly to these limits, merging with the smooth pipe line at the left, and at the right, beyond the dashed line, becoming indistinguishable from the constant f lines for rough pipe.

The Colebrook Function

The basis of the Colebrook function may be briefly outlined. Von Kármán had shown that, for completely turbulent flow in rough pipes, the expression $1/\sqrt{f} - 2\log(D/2\epsilon)$ is equal to a constant (1.74), or, as expressed by Colebrook, $2\log\dfrac{3.7D}{\epsilon} - 1/\sqrt{f}$ is equal to zero. In the transition region of incomplete turbulence von Kármán's expression is not equal to a constant but to some function of the ratio of the size of the roughnesses to the thickness of the laminar boundary layer. Accordingly, Nikuradse had represented his experimental results on artificially roughened pipes by plotting $1/\sqrt{f} - 2\log(D/2\epsilon)$ versus $\epsilon/\eth$, in which $\eth$ is the laminar layer thickness. By this method of plotting, the results for all types of flow and degrees of roughness were shown to fall on a single curve. Using logarithmic scales, the smooth-pipe curve becomes an inclined straight line, and the rough-pipe curves merge in a single horizontal line.

Colebrook (11), using equivalent co-ordinates,[5] plotted in his Fig. 1, here reproduced as Fig. 5, the results of many groups of tests on various types of commercial pipe surfaces. He found that each class of commercial pipe gave a curve of the same form, and while these curves are quite different from Nikuradse's sand-grain results, they agree closely with each other and with a curve representing his transition function.

[5] $\epsilon/\eth$ may be expressed in alternative forms as proportional to $\dfrac{\mathbf{R}\sqrt{f}}{r/k}$ in which $r = D/2$, $k = \epsilon$; or to $\rho\,\dfrac{V_*k}{\mu}$ in which $V_* = \sqrt{\dfrac{\tau_0}{\rho}}$. τ_0 being the shearing stress at the pipe wall, ρ the mass density of the fluid and μ its absolute or dynamic viscosity.

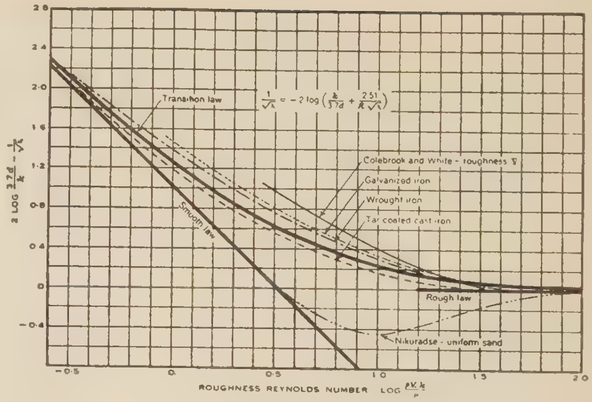

FIG. 5

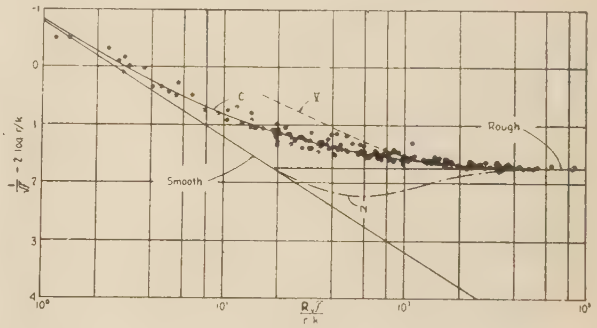

FIG. 6

Rouse (12), also using equivalent co-ordinates, has plotted in his Fig. 6, here reproduced as Fig. 6, a large number of points each of which represents a series of tests on a given size of commercial pipe, together with the Colebrook curve. As he points out, the deviation of the points from the Colebrook curve "is evidently not much greater than the experimental scatter of the individual measurements in any one series."

When the thickness of the laminar layer, which decreases with increasing Reynolds number, becomes so small, compared to the surface irregularities, that the laminar flow is broken up into turbulence, the flow conditions pass over into the zone of "rough pipes," with complete turbulence established practically throughout the flow. Viscous forces then become negligible compared to inertia forces, and f ceases to be a function of the Reynolds number and depends only upon the relative roughness, giving horizontal lines of constant f in the chart. These lines agree with the von Kármán rough-pipe formula

$$1/\sqrt{f} = 2\log\left(\frac{3.7}{\epsilon/D}\right),\quad \text{or}\quad 1/\sqrt{f} = 1.74 - 2\log\left(\frac{2\epsilon}{D}\right)$$

Since f depends upon the relative roughness, the ratio of the absolute roughness to the pipe diameter, even a fairly rough surface in a very large pipe gives a small relative roughness. Thus Colebrook plots the results obtained on the penstocks of the Ontario Power Company, where metal forms and specially laid concrete produced a very smooth example of concrete surface. This in combination with the large diameter gave a relative roughness comparable to drawn brass tubing, with f values falling practically on the "smooth pipe" line of Fig. 1. Such specially fabricated welded-steel pipe lines as those of the Colorado aqueduct system would probably give values along the same curve.

On the other hand, at very high velocities in drawn tubing of

small diameter, even the small absolute roughness is sufficient to break up the laminar boundary layer, and the tubing becomes in effect a "rough pipe." Very few experiments have carried the velocities and Reynolds numbers high enough to permit a close estimate of ϵ for drawn brass, copper, or similar tubing; but by applying the Colebrook function to the available data (14, 15), for the smoothest surfaces reported upon, ϵ was estimated as of the order of 0.000005; and a line corresponding to this value has been drawn in Fig. 2, serving as a minimum limit for surfaces likely to be encountered in practice.

Pipe Friction Factors Applied to Open-Channel Flow

Pipe friction factors have sometimes been applied to open-channel flow, and more commonly the friction losses in large pipes and other closed conduits have been computed from open-channel formulas. The Chezy formula for open channels is $V = C\sqrt{mS}$ in which V is the mean velocity; m the hydraulic mean depth or "hydraulic radius," the sectional area divided by the wetted perimeter; S the slope, the loss of head divided by length of channel, and C a coefficient. The Chezy formula is equivalent to the Darcy formula for pipes, the Chezy coefficient C being convertible into f by the relation $f = \frac{8g}{C^2}$. It should be considered, however, that the Chezy coefficients have been derived principally from observations on relatively wide and shallow channels of large area and rough bottoms, far from circular in shape, and that they involve a free water surface not present in closed conduits, so that, even when the flow is uniform, the problem is highly complex. Consequently, such formulas as Manning's are recommended for open channels in preference to the use of values of C derived from the pipe friction factors.

Open channels dealt with in engineering practice are usually rough-surfaced and of large cross section, corresponding to large Reynolds numbers and falling in the zone of complete turbulence, so that the friction factors are practically independent of Reynolds number. The presence of a free surface, however, with surface waves or disturbances, introduces a consideration not involved in closed-conduit flow. It is therefore the author's view that while, for open channels, we can drop the Reynolds number as an index of performance, we should replace it by a new criterion, the Froude number relating the velocity head and depth, which can be expressed as $\frac{V^2}{2gm}$ or $\frac{V}{\sqrt{gm}}$; or more strictly $\frac{V}{\sqrt{g\partial}}$, in which ∂ denotes the average depth or sectional area divided by the surface breadth; the latter form representing the ratio of mean velocity to the gravitational critical velocity or velocity of propagation of surface waves.

This proposed criterion defines whether the flow falls in the "tranquil," "shooting," or critical state. The neglect of this factor may at least partially account for inconsistencies between various open-channel formulas, and between open-channel and pipe-friction formulas, and casts particular doubt on accepted formulas for open-channel friction in the critical or shooting-flow regions. These considerations suggest the plotting of open-channel friction factors as a function of the relative roughness and the Froude number, in similar manner to the plotting of f as a function of the relative roughness and the Reynolds number for closed conduits.

For the foregoing reasons, Fig. 1 is not recommended with much confidence for general application to open channels, for which a formula such as Manning's better represents the available information. The charts can however be applied, at least as an approximation, to noncircular closed conduits of not too eccentric a form or not too different from a circular section, by using an equivalent diameter

$$D = 4m = 4\left(\frac{\text{Sectional area}}{\text{Length of perimeter}}\right)$$

Since civil engineers usually classify surface roughness by the Kutter and Manning roughness factor n, it would be helpful in selecting a value of ϵ for such variable surfaces as concrete, if we could correlate ϵ and n. P. Panagos[6] has applied the Colebrook function to the test data collected by Scobey (16) and finds the following values of ϵ corresponding to the Kutter n ratings given by Scobey, which may be at least tentatively utilized:

Kutter n.......	0.0105	0.011	0.012	0.013	0.014	0.015	0.016
Absolute roughness ϵ........	0.00015	0.0005	0.002	0.005	0.011	0.02	0.03

Accordingly, on the basis of Scobey's data the lines given in Fig. 2 for concrete may be somewhat more definitely described as follows:

ϵ	n	
0.001	0.0115	Highest practical grade of concrete. Surface and joints smooth
0.003	0.0125	
0.01	0.014	Concrete surface with slight form marks, fairly smooth joints or roughly troweled
0.03	0.016	Prominent form marks or deposits of stones on bottom

Although the curves in Fig. 1 are plotted from definite functional forms which can be accepted with some confidence within the degree of accuracy required in engineering use, further information will be welcomed which would improve the location and definition of the lines in Fig. 2, or which would add new lines for other materials. Any tests of friction head in pipe of any material can be applied to Fig. 1, and corresponding points can be readily located in Fig. 2. A 45-deg line through a point so located can then be added to Fig. 2, to represent a particular kind of pipe surface.

Acknowledgment

For helpful suggestions and assistance, the author is particularly indebted to Prof. B. A. Bakhmeteff, Mr. Ralph Watson, Dr. G. F. Wislicenus, Dr. A. T. Ippen; and to Mr. P. Panagos for collecting data and numerical checking.

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Discussion

R. L. Daugherty.[7] The writer has nothing but commendation for this excellent paper. The author has presented the latest theory combined with the available experimental data in a manner which makes it more convenient for use than has been the case heretofore. His evaluation of relative roughness for different types and sizes of pipes is a step forward.

While this paper deals primarily with pipe friction it is interesting to note the suggestions made concerning the treatment of the flow in open channels. The latter has not been given the attention from the standpoint of rational analysis that has been devoted in the past to pipes. It is to be hoped that developments in this field may be made along the lines suggested by the author.

The author calls attention to the well-known fact that in the transition zone the Nikuradse curves for his artificial sand-grain roughness are quite different from those obtained with commercial pipes. The writer would like to know if the author has any explanation to offer for this marked difference.

C. W. Hubbard.[8] This paper is of interest to engineers who must estimate fluid-friction loss closely for certain types of problems. Ordinarily the Manning type of formula is preferred, since the roughness value may be determined from the type of surface of the wall as contrasted to the Darcy formula where the roughness coefficient varies with the size of pipe and is difficult to estimate. The author's Fig. 2 allows a quantitative wall roughness estimated from the type of wall to be used.

During some recent tests made to select a protective paint for steel which would also have a low friction loss, it was found that several coatings, particularly those consisting of certain bitumastic constituents which required them to be applied thickly to the wall, gave low flow-resistance values. The tests, made in 3-in. pipes, which were split longitudinally to allow proper application of the coating, showed roughness values of the order of those obtained with drawn-brass tubing. However, the appearance of the coating was not as smooth as drawn tubing. The writer had previously experienced this effect with similar coatings. There seems to be little published material on the friction loss produced by various protective paints and coatings on pipe walls, particularly on small pipes when the flow is likely to occur in the transition range where the friction loss is dependent upon Reynolds number. Apparently the roughness of such surfaces is of the wavy type which cannot be evaluated on the same basis as the same magnitude of roughness which is of the granular type.

A. T. Ippen.[9] The author has ably satisfied the object of his paper stated in the beginning with an extremely timely and practical summary of the latest information available on pipe friction. Academic research in this field over the last 30 years conducted on a fundamental basis has finally yielded a satisfactory explanation of the nature of the laws of pipe friction and has cleared up the concepts of energy dissipation in conduits and channels.

The evidence for the adoption of the methods for determining the pipe friction factor as presented by Colebrook is rather astonishing. Some experiences in this connection may be contributed here. The writer has computed two comprehensive sets of data on pipe friction, one by John R. Freeman and another by L. H. Kessler. The former completed his experiments during the years 1889 to 1893 and his data were published by this Society in a special volume (15)[10] in 1941. The second set of data was obtained from pipe-friction experiments at the Wisconsin Experimental Station, the results of which were published in 1935. Both experimenters performed tests on $^1/_4$ in- to 8 in-diam wrought-iron pipes in new condition covering the maximum range in Reynolds numbers possible under their experimental conditions. After plotting these results every one of their runs shows essentially the f versus $\mathbf{R}$ curve indicated by Colebrook and ϵ values calculated for all the various sizes come out very close to the average value stated for wrought-iron pipe in the present paper. It must be remembered that Kessler's data were obtained 40 years after those of Freeman and that it can hardly be assumed that manufacturing processes remained identical during that period.

Another fact of importance to the practical engineer from this analysis of Freeman's and Kessler's data is worth mentioning. Rouse and Moody in their f versus $\mathbf{R}$ curves terminate the transition range from smooth- to rough-pipe flow along a line corresponding to a ratio of absolute roughness ϵ to the laminar boundary-layer thickness δ of 6.08. Kessler's and Freeman's data do not give a single value that high in all their runs; their highest values obtained were about $\frac{\epsilon}{\delta} = 2.5$. Under practical conditions of use therefore the flow of water in pipes occurs well in the transition range from smooth- to rough-pipe flow.

This fact easily explains why a final solution of the pipe-friction problem was possible only after the concepts of "smooth-pipe" and "rough-pipe" flow had been established separately. While Nikuradse's results on uniformly sand-coated pipe were helpful in this respect, they also resulted in more complicated transition curves than are obtained from tests with the statistical roughness patterns encountered on most commercial pipe surfaces. The Colebrook universal function seems to fit the better in this transition range; the more the roughness irregularities are statistically distributed as far as size and shape are concerned and vice versa, the more regular the size and pattern of the irregularities the closer Nikuradse's transition curves are approached, where finally the critical velocity for all roughness bodies is the same in the ideal case of completely uniform size.

The familiar functions for the pipe friction factor f may be written in the following form

$$\frac{1}{\sqrt{f}} = 1.74 - 2\log\frac{18.6}{\mathbf{R}\sqrt{f}} \qquad [1]$$

for smooth-pipe flow

$$\frac{1}{\sqrt{f}} = 1.74 - 2\log\frac{\epsilon}{r} \qquad [2]$$

for rough-pipe flow

$$\frac{18.6}{\mathbf{R}\sqrt{f}} = 0.282\frac{\delta}{r} \qquad [3]$$

for laminar boundary-layer thickness.

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[8] Lieutenant Commander, U.S.N.R. Mem. A.S.M.E.

[9] Assistant Professor, Hydraulic Laboratory, Lehigh University, Bethlehem, Pa.

[10] Numbers in parentheses throughout the discussion refer to the Bibliography at the end of the author's paper.

According to Colebrook, Equations, [1] and [2] are combined into the following universal function

$$\frac{1}{\sqrt{f}} = 1.74 - 2 \log \left(\frac{18.6}{\mathbf{R}\sqrt{f}} + \frac{\epsilon}{r} \right) \quad \ldots\ldots\ldots\ldots [4]$$

This function reverts to either Equation [1] or Equation [2], if either the influence of the relative roughness disappears or when the viscous influence becomes insignificant. By use of Equation [3], the Colebrook function may be written in the alternative form

$$\frac{1}{\sqrt{f}} = 1.74 - 2 \log \frac{\epsilon}{r} \left(1 + 0.282 \frac{\delta}{\epsilon} \right) \quad \ldots\ldots\ldots [5]$$

This equation clearly brings out the dependence of the pipe friction phenomena upon the thickness of the laminar boundary layer, i.e., on the viscosity of the fluid. It will be found in practical calculations that this influence is very seldom absent. The proposed ultimate value of $\frac{\mathbf{R}\sqrt{f}}{r/\epsilon} = 400$ is equivalent to a value of $\frac{\epsilon}{\delta}$ of 6.08.

It is evident that aging of pipes under varying conditions of use will result in new values of absolute roughness which at present are not easily predicted. From experiments on galvanized steel pipe of 4 in. diam at the Hydraulic Laboratory of Lehigh University, an initial average value of $\epsilon = 0.00045$ ft was obtained. This value of ϵ was doubled within 3 years as a result of the change in surface conditions with aging under moderate conditions of use. It must be remembered here that this change in ϵ represents only about a 20 per cent increase in the Darcy-Weisbach factor f, since the ϵ value is a much more sensitive indicator of pipe roughness than the factor f.

In conclusion, it may be hoped that this paper will bring the general adoption of this relatively easy and reliable method of determining pipe friction and thereby establish a standard procedure in practice which is based upon sound analytical and experimental evidence.

W. S. Pardoe.[11] In the following tests on pipes of various diameters and materials, the exponent n in the exponential formula

$$V = Kd^mS^n$$

varied from 0.535 to 0.546, thus checking Williams' and Hazen's formula

$$V = 1.318\, C_w r^{0.63} S^{0.54}$$

very closely. The maximum value of $\mathbf{R}$ was about 1,250,000 for 8-in. Neoprene dock-loading hose (very smooth) which is much below the "complete turbulence zone." The tests included:

- 6-in. Italite cement-asbestos pipe (predecessor of Transite)
- 4-in. Ruberoid cement-asbestos extruded pipe
- 4-in. fiber conduit
- 6-in. and 8-in. Neoprene dock-loading hose for E. I. du Pont de Nemours
- 2-in. to 12-in. steel pipe
- 8-in. rubber dock-loading hose with 1-in. $\times$ $^1/_8$-in. helical metal band on inside

In no case except the last did the exponent n show even a tendency of decreasing, let alone approaching a value of 0.5 or complete turbulence. This must be due to the smoothness of the materials and the low value of Reynolds number. In the last case, the values of f did show a tendency to become constant, the value of ϵ/D being quite large.

The writer has not conducted a sufficient number of tests on pipes and is far from a pundit on this subject. At some time in the future, he will attempt to work into the "complete turbulence zone," if such there is, even if he must use a bit of 4-in. turberculated cast-iron pipe.

Mr. Pigott in his discussion has mentioned my insistence on the fact that the coefficients of Venturi meters become constant. This coefficient may be approximated by the formula

$$C = \sqrt{\frac{1 - \beta^4}{1 - \beta^4 + K}}$$

in which β is the diameter ratio d_2/d_1 and K is the coefficient of loss in

$$h_f = K \frac{V_2^2}{2g}$$

The value of K on the flat part of tests of 85 cast-iron Venturi meters approximates

$$K = \frac{0.0435}{(d_2'')^{0.25}}$$

As the absolute roughness is constant, the proportional roughness varies inversely as the diameter or the coefficient increases with the diameter. The tests ran to quite high values of Reynolds number in terms of $\frac{V_2 d_2 \rho}{\mu}$, thus indicating there is such a thing as complete turbulence. Solving the foregoing expression

$$K = \frac{1 - \beta^4}{c^2} - (1 - \beta^4)$$

Hence a constant value of c gives a constant K, or h_f varies as V^2.

This is of course arguing from the writer's experience with Venturi meters to make up for his lack of adequate experimental knowledge of the subject under discussion.

Professor Moody says f is a function of "two and only two" dimensionless quantities ϵ/D and $\frac{VD\rho}{\mu}$. The writer has found in his work on Venturi meters a variation of over $^1/_2$ per cent, due to the effect of the ambient temperature.[12]

As a variation of $^1/_2$ per cent in c requires a variation of 25 per cent in k it seems to the writer the effect of a difference of temperature of 20 deg F on f at low value of $\mathbf{R}$ might be considerable. This effect is brought about by a change in the boundary shear; thus

$$\tau_0 = \frac{f\rho V^2}{8} = \mu \frac{dv}{dy}$$

If Q is kept constant dv/dy will also be constant, and μ corresponds to the temperature of the inside wall of the pipe, which will lie between the ambient temperature and that of the water. It will decrease as the velocity increases as a result of the heat being conducted away more rapidly. This the writer will check in future experiments; it may throw some light on the upper limit of the critical or unstable zone. The effect is a function of Reynolds and Prandtl's or Nusselt's numbers, and the writer is not certain "what the price of cheese in Denmark does to effect f."

[11] Department of Civil Engineering, University of Pennsylvania, Philadelphia, Pa.

[12] "Effect of High Temperatures and Pressures on Cast-Steel Venturi Tubes," by W. S. Pardoe, Trans. A.S.M.E., vol. 61, 1939, p. 247.

Professor Moody is to be congratulated on producing a very usable plot of friction factors which in due time may replace the Pigott and Kemler curves which have to date been extensively quoted and used by engineers. Thus do we progress.

R. J. S. Pigott.[13] This study of friction factor in pipes is particularly interesting to the writer, as it is a valuable further rationalization of a situation which has been unsatisfactorily empirical.

At the time the writer's own correlation (4) was presented (1933) there was almost complete lack of uniformity between various formulations in general use, wandering all the way from Kutter, Hazen, and Williams tables, to Aisenstein's averaged values.

There was great need to prepare a formulation that would work satisfactorily for all kinds of conduit, from brass tubing to brick ducts, and for all fluids.

Dr. Kemler, then on the writer's staff, did the laborious part of the job, in correlating the results of all the experiments published up to that time, culling all those with incomplete data (6). The writer summarized this work, in form for direct application generally, introducing the roughness effect by rather strong-arm empirics, but at any rate the resulting chart worked well and has been growing in use.

The great value of the author's study is that it puts the roughness effect, at last, on a much better justified basis. For example, Buckingham (Fig. 1, reference 5) drew the lines for different sizes of steel pipe curved as they approached the viscous region, the same as the author now shows them; Stodola (Fig. 1, ref. 5) shows them straight and intercepting.

The later material used by the author shows that they are curved. Another important point settled by the author is that the lines for all roughnesses finally reach a constant value. The point at which this condition obtains is plainly shown to be a function of relative roughness, and so solves a difficulty Dr. Kemler and the writer had, in correlating some of the test material. Some of the experimental results showed rather flat coefficients that were unexpected in regions of moderate roughness. But this constant value of f is confirmed by Professor Pardoe's findings on Venturi discharge coefficients. He has been pointing out for years that the coefficient reaches a constant value at some Reynolds number that increases with size. Since most Venturis above rather small sizes are made with cast approach cones and the losses are substantially represented by pipe friction, this situation corresponds to flat final value of f at complete turbulence, and a decrease of the value of f with decrease of roughness.

Some engineers may be interested in the flow of queer materials like greases, muds, cement slurries, etc., that have thixotropic properties (quoted from the rheologists), that is, they have plasticity mixed in with viscosity. All these materials have apparent viscosities which decrease with increase of shear rate, but, when they finally reach turbulent flow, behave like true liquids of rather low viscosity. Such activities as oil-well drilling, cement-gun and grouting operations, automotive greasing equipment, and ball bearings involve such materials. In food industries, one gets tomato ketchup and pea soup; glue and soap solutions, paint and varnish operations, and various queer materials in the rayon and plastics industries. For those interested, a paper[14] by the writer presents more or less a rational solution that has been quite satisfactorily supported by tests.

In Fig. 1, the author has drawn his dotted line of complete turbulence somewhat in advance of the Reynolds number values at which the friction factor becomes a constant quantity. The writer finds that the expression

$$\frac{3500}{\frac{\epsilon}{D}} = \mathbf{R}_t$$

represents, as closely as can be determined from the small-scale diagram, the point at which the friction factor becomes absolutely constant. It is curious and probably only accidental that the value 3500 corresponds about to the upper limit of the critical zone.

Hunter Rouse.[15] Important results of laboratory research frequently do not reach the hands of practicing engineers until many years after the original papers have been published. A salient case in point is the discovery by Blasius in 1913 of the dependence of the Darcy-Weisbach resistance coefficient f upon the Reynolds number $\mathbf{R}$, which did not come into general engineering use until perhaps a decade ago. It often happens, however, that once engineers have accepted a new idea they are loath to modify it in any way. The paper under discussion is a very commendable endeavor to make recent experimental findings immediately useful to the engineer, but the writer feels that it still caters to a regrettable degree to the engineer's innate conservatism.

If the writer's belief is correct, this paper is intended to fulfill the same purpose as that which prompted the writer to present a somewhat similar paper (12) and resistance chart at the Second Hydraulics Conference in 1942. The author claims that in this chart, which is reproduced herewith in slightly modified form,[16] the writer adopted co-ordinates inconvenient for ordinary engineering use. Such criticism resulted from the writer's deliberate advancement beyond the now familiar Blasius f-versus-$\mathbf{R}$ notation in the belief that both greater convenience and greater significance would be attained thereby. Since these two papers of identical purpose thus differ in their basic method of approach, a criticism of the one point of view must necessarily involve a defense of the other.

Although Blasius' original dimensional analysis of the variables involved led to his adoption of the form VD/ν as the most significant grouping of terms upon which f should depend, it must be realized that the following three different combinations of the same variables are all equally valid for the basic case of a smooth pipe:[17]

$$f = \frac{2gh_fD}{LV^2} = \varphi_1\left(\frac{VD}{\nu}\right) = \varphi_1(\mathbf{R})$$

$$f = \frac{2gh_fD}{LV^2} = \varphi_2\left(\frac{2gh_fD^3}{L\nu^2}\right) = \varphi_2'(\mathbf{R}\sqrt{f})$$

$$f = \frac{2gh_fD}{LV^2} = \varphi_3\left(\frac{2gh_fQ^3}{L\nu^5}\right) = \varphi_3'(\mathbf{R}f^{1/5})$$

The combination now most familiar to the engineer, of course, is the first, although it has long since been proved that it will yield a linear plot on logarithmic paper for only the laminar zone. The second, on the other hand, is the basis of the Kármán-Prandtl analysis of the turbulent zone, the general functional relationship simply being written in the specific form

[13] Chief Engineer, Gulf Research and Development Company, Pittsburgh 30, Pa. Fellow A.S.M.E.

[14] "Mud Flow in Drilling," by R. J. S. Pigott, Drilling and Production Practic A.P.I., 1941, pp. 91–103.

[15] Director, Iowa Institute of Hydraulic Research, University of Iowa, Iowa City, Iowa.

[16] "Elementary Mechanics of Fluids," by Hunter Rouse; John Wiley & Sons, Inc., New York, N. Y. (in Press).

[17] "Solving Pipe Flow Problem With Dimensionless Numbers," by A. A. Kalinski, *Engineering News-Record*, vol. 123, 1939, p. 23.

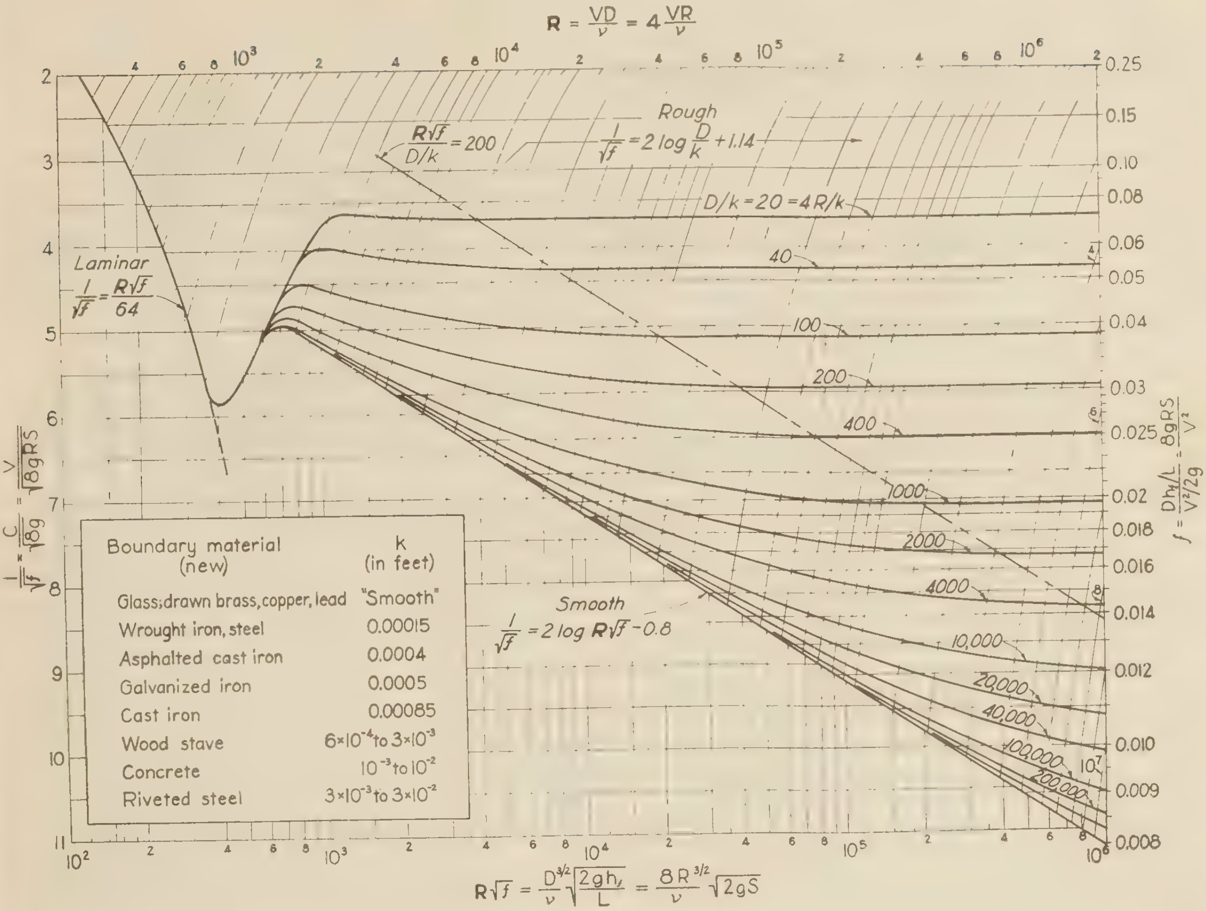

Fig. 7

$$1/\sqrt{f} = A + B \log (\mathbf{R}\sqrt{f})$$

Despite the author's indication to the contrary, f is not inextricably embodied in the second term of this relationship, as may be seen from the identity $\mathbf{R}\sqrt{f} = \sqrt{(2gh_fD^3/L\nu^2)}$. If the Kármán-Prandtl parameters are chosen as the basis of a semilogarithmic chart, as in the accompanying figure, not only will the smooth-pipe relationship plot as a straight line, but all transition curves from the smooth to the rough relationship will be geometrically similar. It would appear to the writer that this combines ease in interpolation (and hence convenience) with greater significance than the Blasius plot will permit. This, therefore, is one of the writer's two reasons for continuing to recommend the newer type of chart in preference to that of the author.

The writer's second reason will be evident after further inspection of the foregoing functional relationships. The first relationship will be directly useful in graph form only if the velocity or rate of flow is known; otherwise the desired coefficient may be evaluated from the graph only through the inconvenient process of trial and error. If the velocity or rate of flow is not known, on the other hand, a graph of the second functional relationship will make the desired coefficient immediately available. In order to provide a single chart which would satisfy both sets of requirements, the writer supplied ordinate scales of both f and $1\sqrt{f}$ (the latter being proportional to the Chezy C) and abscissa scales of both $\mathbf{R} = VD/\nu$ and $\mathbf{R}\sqrt{f} = \sqrt{2g\,h_f/L}\;D^{3/2}\nu$. Since the parameters $1/\sqrt{f}$ and $\log (\mathbf{R}\sqrt{f})$ were selected by the writer for the primary ordinate and abscissa scales, the alternative abscissa parameter $\log \mathbf{R}$ is necessarily represented by curved lines over a portion of the writer's chart. Had $\log f$ and $\log \mathbf{R}$ been chosen as the primary parameters, $\log (\mathbf{R}\sqrt{f})$ would still have required sloping lines in the grid; such choice therefore involves no particular advantage over the writer's but rather defeats the writer's purpose owing to the accompanying distortion of the entire system of transition curves. The author's graph, of course, contains no secondary grid system simply because it permits direct solution for only one of the several variables.

Brief mention might be made of the third possible combination of variables, which is evidently applicable to problems in which the diameter is the unknown quantity. So long as the pipe is smooth, such a plot will be of use, but the adoption of a similar function for the case of rough surfaces will still require a trial-and-error solution, unless the graph is made hopelessly complex, owing to the fact that for a given boundary material the pipe diameter must be known before the relative roughness may be evaluated. Solution by trial might therefore proceed just as well from either of the other two functional relationships contained in the writer's diagram.

The writer commends the author's presentation in graph form of the values of absolute roughness given in the writer's paper, but notes with interest that this plot is consistent with the writer's rather than the author's choice of basic parameters. Such a graph would therefore have its greatest value when prepared as

a marginal extension of the writer's resistance chart, for then no relative-roughness computations would have to be made.

So far as the author's discussion of open-channel resistance is concerned, the writer takes exception to two points of fundamental importance: First, the author states that such relationships as the Manning formula should be used in open-channel computations in preference to values derived from pipe tests, implying that the familiar empirical open-channel formulas are inherently more valid. It is known, however, that the Manning formula (not to mention those of Bazin and Kutter) is not in accordance with the logarithmic law of relative roughness upon which the author's paper is based. So far as the writer can ascertain, the only reason pipe tests could not generally be used in evaluating open-channel resistance lies in the fact that few open-channel boundary surfaces are suitable to testing in pipe form. Aside from the moot question of the effect of cross-sectional shape (which the empirical open-channel formulas in no way answer), it would appear to the writer that a general resistance graph for uniform open channels should differ little from that for pipes, except in that the familiar parameters C and S might conveniently be included in the co-ordinate scales; this has been done in the present form of the writer's chart.

The writer's second objection to the author's closing section is in regard to his implication that the Froude number should replace the Reynolds number as the fundamental resistance parameter for open-channel flow. So far as boundary resistance is concerned, the writer can see no possibility of the Froude number playing a comparable role. It is true that viscous shear is of little significance in comparison with boundary roughness in most open-channel problems, but it is also true that the effect of surface waves upon the internal resistance to flow has not yet been ascertained. The open-channel problem is, in fact, quite analogous to that of ship resistance, in which the matter of surface drag is considered wholly independent of the Froude number. If, to be sure, the phenomena of slug flow, atmospheric drag, and air entrainment prove to govern the resistance in the comparatively infrequent case of supercritical flow in open channels, then the Froude number may well become an appropriate resistance criterion, as it already is for cases of channel transition. But to imply that it should replace the Reynolds number as a resistance parameter whenever a free surface exists seems rather untimely to the writer, in that it could lead to serious misinterpretation of those few principles of boundary resistance which have been definitely established.

P. H. Schweitzer.[18] Lest the author's charts, presented in delightfully handy forms, be used indiscriminately, it is perhaps in order to add one note of caution. Most of the statements, formulas, and charts are valid only for "long" pipes. For short pipes, the rules controlling turbulence are different, and Reynolds number is not the sole or deciding criterion for the state of flow.

If the velocity of flow in a long tube is decreased below the "critical" value, a change from turbulent to laminar flow takes place rather abruptly. The author sets the indeterminate region between 2000 and 4000 Reynolds number. Even that represents a rather narrow strip in the total range covered by the flow of such liquids as water or light oil. Outside of this indeterminate region, the flow is either completely laminar or decidedly turbulent, ignoring the rather thin laminar-boundary layer.

While this is true of relatively long tubes, for short tubes or nozzles it is not. In a short tube, as was shown by the writer,[19] the normal state may be described as "semiturbulent flow," which may be visualized as a turbulent core in the center and a laminar envelope near the periphery. The thickness of the laminar envelope may vary between wide limits. The change from turbulent to laminar flow or the reverse takes place in a short tube so gradually that the intermediate stage usually covers the whole practical region.

Of course, in both long and short tubes, turbulent flow is promoted by high flow velocity, large tube size, curvature of the tube, divergence of the tube, rapid changes in direction and cross-sectional area of the tube. Laminar flow is promoted by high liquid viscosity, laminar approach, rounded entrance to the tube, slight convergence of the tube, absence of curvature and disturbances.

Irrespective of the length of the tube an originally turbulent flow will remain turbulent, if its Reynolds number $\mathbf{R} = vd/\nu$ is greater than the critical Reynolds number. Conversely, an originally laminar flow will remain laminar if $\mathbf{R}$ is lower than the number.

If the flow at the entrance is turbulent but its Reynolds number in the tube is lower than the critical, the flow will turn purely laminar if the tube is straight, reasonably smooth, and sufficiently long. If the flow at the entry of the tube is laminar but its Reynolds number is above the critical, it is hard to predict the character of the ensuing flow. If the entry is smooth and rounded and the tube free from disturbances and irregularities, the flow will remain laminar even at Reynolds numbers[20] as high as 15,000.

In a complete absence of all disturbances, a laminar flow probably never turns turbulent, no matter how high its Reynolds number, but the slightest disturbance will ultimately cause turbulence if the Reynolds number is above the critical. The higher the Reynolds number the greater the disturbance, the shorter the tube travel necessary for turbulence to set in.

In a short tube the critical Reynolds number is not the one above which the flow generally or in a particular case is turbulent. The flow is frequently laminar at Reynolds numbers above the critical and it may be turbulent or semiturbulent at Reynolds number below the critical.

The critical Reynolds number is the one below which, in a straight long cylindrical tube, disturbances in the flow will damp out. Above the critical Reynolds number disturbances (approach, entry, etc.) never damp out, no matter how long the tube is. The critical Reynolds number so defined was found by Schiller[21] to be approximately 2320.

In short tubes, or nozzles, the length is not nearly enough for the flow to assume a stable condition. Under the circumstances, a Reynolds number higher than critical will have a tendency toward turbulence and vice versa, but it may take a tube travel of 60 times diameter before a stable velocity distribution is developed. The actual flow in the nozzle will be influenced considerably by the state of flow before the orifice and the disturbances in the approach and within the nozzle. The combination of these factors in addition to the Reynolds number will determine the state of turbulence at the exit of the short tube. For a given short tube or nozzle, the influence of the nozzle factors can be considered the same; therefore the Reynolds number alone will determine the character of the flow.

With decreasing Reynolds number, the thickness of the laminar layer increases and the turbulent inner portion decreases until it finally disappears. It is peculiar to nozzles or short tubes that the change from turbulent to laminar flow (or vice versa)

[18] Professor of Engineering Research, The Pennsylvania State College, School of Engineering, State College, Pa. Mem. A.S.M.E.

[19] "Mechanism of Disintegration of Liquid Jets," by P. H. Schweitzer, *Journal of Applied Physics*, vol. 8, 1937, pp. 513–521.

[20] With a convergent tube of 10-deg cone angle Gibson (Proceedings of the Royal Society of London, vol. A83, 1910, p. 376), observed laminar flow at $\mathbf{R} = 97{,}000$.

[21] "Untersuchungen über laminare und turbulente Strömung," by L. Schiller, *V.D.I. Forschungsarbeiten*, vol. 248, 1922.

takes place gradually rather than abruptly. The semiturbulent state extends over a wide range of Reynolds numbers, differing only in the relative thickness of the turbulent core and laminar envelope.

Author's Closure

The paper was intended for application to normal conditions of engineering practice and specifies a number of qualifications limiting the scope of the charts, such as their restriction to round (straight) new and clean pipes, running full, and with steady flow. Under such conditions it was stated, as noted by Professor Pardoe, that the friction factor f "is a dimensionless quantity, and at ordinary velocities is a function of two, and only two, other dimensionless quantities,—the relative roughness of the surface and the Reynolds number."

Under abnormal conditions f could of course be affected by other dimensionless criteria. In closed conduits at very high velocities or with rapidly varying pressures it depends on the Mach or Cauchy number introducing the acoustic velocity. In open channels, as pointed out, free surface phenomena, gravity waves, make it logically dependent on Froude's number. At very low velocities in shallow open troughs it would conceivably be controlled also by the Weber number for surface tension and capillary waves. Capillary forces while important to insects, as to a fly on flypaper, are negligible to us in problems of engineering magnitude. Under usual conditions of pipe flow only the two dimensionless ratios mentioned need be considered, and it is possible to present the relations between the factors in a chart such as Fig. 1.

The discussions have brought out a number of other departures from normal conditions and further limitations to the scope of the charts. Professor Pardoe reminds us that a considerable temperature difference between the fluid and pipe wall may have a measurable effect on the shear stresses, due to ambient currents which would increase the momentum transfer in similar manner to turbulent mixing. This effect would probably be of importance only at the lower Reynolds numbers and with material temperature differences.

Mr. Pigott reminds us that the scope is limited to simple fluids and does not cover "queer materials like greases, muds, cement slurries" and mixtures with suspended solids. Professor (now Commander) Hubbard and Professor Pardoe mention some unusual forms of pipe surfaces. The author thinks that most of these, including paint coatings, will follow the lines of the charts closely enough for practical purposes if the proper roughness figures are determined; but the rubber dock-loading hose with helical internal band will probably follow a curve similar to curve V in Fig. 6, which Colebrook and White obtained for spiral-riveted pipe.

Dr. Ippen mentions the rate of increase of roughness from corrosion and gives some useful test information. Colebrook found that corrosion usually increases the value of ϵ at substantially a uniform rate with respect to time. Professor Schweitzer calls attention to the point that the pipe must be long, with an established regime of flow, and that the charts do not apply to the entrance or "smoothing section" which require separate allowances. Fortunately we are seldom concerned with close estimates of friction loss in short tubes, where friction is a minor element in the total loss of head.

Dr. Ippen's discussion admirably summarizes the basic structure of the charts and gives supporting evidence. His own studies of the problem had, the author believes, led him independently to conclusions similar to Colebrook's.

The Colebrook function has given us a practically satisfactory formulation bridging the previous gap in our theoretic structure, a region in which the majority of engineering problems fall. It has the further useful property of covering in a single formula the whole field of pipe flow above the laminar and critical zones; and throughout the field agrees with observations as closely as can be reasonably demanded within the range of accuracy available in the measurements, particularly in the evaluation of the boundary roughness.

Referring to a question raised by Professor Daugherty, the inconsistency between Nikuradse's tests in the transition zone and those from commercial pipe is usually attributed either to the close spacing of the artificially applied sand grains, such that one particle may lie in the wake behind another, or to the uniformity of Nikuradse's particles in contrast to the usual commercial surface, which is probably a mixture of large and small roughnesses distributed at random. The latter explanation seems particularly plausible, since a few large protuberances mixed with smaller ones could project far enough into the laminar boundary layer to break it up, while a uniform layer of projections of average size would all remain well within the same thickness of layer. Thus Nikuradse's curve clings closely to the smooth pipe line much farther than the curve for commercial surfaces. At any rate the artificial character of Nikuradse's surfaces weighs against the use of his values in the region where the discrepancies appear.

Mr. Pigott reviews the progress in charting friction factors and gives evidence supporting the laws adopted. At the end of his discussion he brings up an interesting question, the form and location of the dashed line in Fig. 1 marking the boundary of the rough pipe zone for complete turbulence, beyond which the friction factors become practically horizontal. With his gift for detecting relationships he arrives at a modified equation for this curve.

Referring to Figs. 5 and 6 it will be noted that Nikuradse's experiments on artificial roughness gave a curve which dropped below the "rough pipe" line and then approached it from below, while ordinary commercial pipes give points which approach the rough pipe line from above, and that both sets of points seem to merge with the rough pipe line at about $\dfrac{\mathbf{R}\sqrt{f}}{D/2\epsilon} = 400$, which Rouse accordingly adopted as the equation of the boundary of the rough pipe zone, the dashed line shown in Fig. 1. If, however, we adopt the Colebrook function for the transition region to the left of this boundary curve, strictly speaking the Colebrook curves never completely merge with the constant f lines but are asymptotic to them; so that on the basis of the Colebrook function there is no definite boundary to the rough pipe region.

Practically however the Colebrook function converges so rapidly to the horizontal f lines that beyond Rouse's dashed curve the differences are insignificant. Considering the practical difficulties of measurement and consequent scatter of the test points, and the fact that the Colebrook function is partly empirical and merely a satisfactory approximation, it seems hardly justifiable to draw fine distinctions from an extrapolation of this function. If the function could be accepted as completely rational it would be more logical to locate the boundary curve so that it would correspond to some fixed percentage of excess in f over the f for complete turbulence.

Prompted by Mr. Pigott's suggestion, the author has analyzed the Colebrook equation from this point of view. Calling f the value of the friction factor according to Colebrook, and f_k the value for complete turbulence according to von Kármán, the Colebrook equation can be expressed

$$1/\sqrt{f_k} - 1/\sqrt{f} = 2\log\left(1 + \frac{3.7 \times 2.51}{\epsilon/D\,\mathbf{R}\sqrt{f}}\right)$$

Calling $\dfrac{3.7 \times 2.51}{\epsilon\ D\ \mathbf{R}\sqrt{f}} = x$, a small quantity compared to 1 (of the order of 0.05 or less in the region of the boundary curve) then

log $(1 + x)$ can be expanded in a series giving

$0.4343\left(x - \dfrac{x^2}{2} + \dfrac{x^3}{3} - \ldots\right)$; and neglecting x^2 and higher powers

$$1/\sqrt{f_k} - 1/\sqrt{f} = 0.8686x.$$

If now we denote by s the *proportional* change in f, that is $s = \dfrac{f - f_k}{f_k} = \dfrac{f}{f_k} - 1$, s being small compared to 1, then

$$1/\sqrt{f_k} - 1/\sqrt{f} = 1/\sqrt{f}(\sqrt{f/f_k} - 1)$$
$$= 1/\sqrt{f}(\sqrt{1+s} - 1)$$

which, expanding by the binomial formula, is very nearly

$$1/\sqrt{f}\ (1 + s/2 - 1) = s/2\sqrt{f}$$

Hence

$$s/2\sqrt{f} = 0.8686x = \frac{0.8686 \times 3.7 \times 2.51}{\epsilon/D\ \mathbf{R}\sqrt{f}};\ \text{and } s = \frac{16.1}{\epsilon/D\mathbf{R}}$$

is the proportional change in f caused by the Colebrook function.

In plotting Fig. 1, the author, instead of continuing indefinitely with the insignificant effect of the small term, and favoring the view that f should become substantially constant in the rough pipe zone, adopted the compromise of ignoring the variation when it fell below about one half of one per cent; and beyond this point the f lines were drawn horizontally at the Kármán value. That is, the chart applied the Colebrook formula only to the transition zone.

Putting $s = .005 = \dfrac{16}{\epsilon/D\mathbf{R}}$, we have $\mathbf{R} = \dfrac{3200}{\epsilon/D}$, practically confirming Mr. Pigott's deduction. If we adopt a one per cent variation of f as a reasonable allowance, the boundary curve could be plotted from $\mathbf{R} = \dfrac{1600}{\epsilon/D}$. It might be more logical, to be consistent with the Colebrook function, to use this formula for the boundary curve instead of Rouse's form. The two curves differ but little, and the choice seems more a matter of academic preference than practical importance; the scatter of test observations obscures a final answer.

As noted in numerous references in the paper the author has been indebted to Professor Rouse for his contributions to the subject, particularly his valuable paper at the Iowa Hydraulic Conference. Professor Rouse's inclusion in his discussion of his chart, Fig. 7, from the latter paper, is a useful addition to the material here collected. The co-ordinates selected for this chart bring out the functional relationships in a simple manner; and those who prefer to adopt this form of chart now have it at hand.

The author still considers it less convenient for usual engineering problems than his Fig. 1. While the horizontal scale of Fig. 7 can be expressed in terms of the frictional loss of head in place of f, this is of no help where the velocity is given and the friction loss is to be found, nor is it of much help in usual engineering problems where the total head is given and the velocity is to be found. The total head almost always includes not only the friction loss in a pipe system, but also the exit loss, and the losses at entrance and in fittings, bends, and changes in section; and we can seldom assign in advance the value of the friction head or slope of the hydraulic gradient; so that successive approximations or trial-and-error solutions are still required. While Rouse's chart is easier to construct, for the reasons explained the author adopted the form of Fig. 1 as easier to use.

Regarding the author's suggestions as to open channels, the questions raised by Professor Rouse are probably due mainly to the omission of fuller explanation in the paper. It was not the intention to imply that at low velocities in relatively smooth open channels the friction loss would be independent of Reynolds number, and it may well be found that in this region the logarithmic laws may continue to apply, at least in modified form, and that, as Professor Rouse states, "a general resistance graph for uniform open channels should differ little from that for pipes."

The author was speaking of another region, "open channels dealt with in engineering practice...usually rough-surfaced and of large cross section, corresponding to large Reynolds numbers and falling in the zone of complete turbulence." With fairly high velocities, corresponding to large Reynolds numbers, in the presence of a free surface, dimensional considerations require us to include the Froude number as a criterion; and in the region of complete turbulence we can fortunately afford to omit the Reynolds number as a controlling factor so that we do not have too many variables to handle. The author did not intend to imply that the Froude number "should replace the Reynolds number as a resistance factor *whenever a free surface exists*" but only in the region described, which however is within the range of ordinary practice.

Professor Rouse recognizes that free-surface phenomena comprise a factor in the problem; his objection to including the Froude number is merely that "the effect of surface waves upon the internal resistance to flow has not yet been ascertained—" which calls on us to investigate the effect rather than to ignore it. Certainly wave-making resistance is a very real factor both in ship resistance, and in open channel flow in the region of the gravitational critical velocity. Even in tranquil flow it still may have a measurable effect; the location of the maximum velocity point below rather than at the surface suggests an influence of this factor.

The author is confident that Professor Rouse will agree with his belief that further research on open channel friction is much needed; and he commends such a project particularly to the civil engineers. Neither the f versus $\mathbf{R}$ charts nor such formulas as Manning's, Kutter's or Bazin's are believed to take into account all of the major controlling factors, and a statistical analysis of available data along the lines suggested, supplemented by further experiments, may yield working charts or formulas of great value to engineers.

It is regretted that Professor (now Major) Colebrook, who has been serving in the British Army since 1939, was unable to submit a discussion. The author wishes to thank all of the discussers for their useful contributions, and also to thank Mr. Richard B. Willi for his able presentation of the paper at the Pittsburgh meeting on behalf of the author.

Factors Affecting the Thickness of Coal-Ash Slag on Furnace-Wall Tubes[1]

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Modern central-station power plants generating steam in slag-tap pulverized-coal-fired furnaces often report difficulties in operation as the result of excessive accumulations of slag on the heat-absorbing surfaces of the furnace. Such deposits of slag interfere with the expected transfer of heat from the furnace to the tube metal, and not only may cause trouble in maintaining superheat temperatures at proper levels, but also may be responsible for troubles experienced with circulation. Few means have been available in the past to evaluate the tendency of slags to form objectionable deposits. This paper describes methods of predicting the relative thickness of slag deposits on heat-absorbing surfaces and is based upon the application of data obtained in the laboratory on the flow properties of coal-ash slags at high temperatures. Although based on idealized furnace conditions, it should serve as a guide in comparing the behavior of different coal ashes in the same furnace under similar operating conditions, or in predicting the effect of change in these conditions on the action of a given coal ash. Also, it should serve as a guide for field tests of the effect of slag on furnace performance.

As a result of increased interest in the effect of slag on the performance of modern central-station boiler furnaces burning pulverized coal, recent studies in this laboratory have been directed largely toward determining the fundamental properties of coal-ash slags at high temperatures. Such data are useful in studying the accumulation of slag on heat-absorbing surfaces, such as furnace-wall tubes. One of the problems facing both designers and operators of these furnaces is the effect of deposits of slag in decreasing the normal transfer of heat from the furnace cavity to the tube metal, resulting in unpredictable and variable fluctuations in the temperature of the furnace gases. This usually causes difficulty in maintaining a desired superheat temperature, while troubles experienced with circulation often have been attributed to unexpected variations in the rate of heat transfer to certain sections of waterwalls caused by unforeseen slag accumulations.

Design rates of heat transfer based upon previous experience with a certain coal may not be attainable when other coals are burned because of differences in the composition of the ash, which may result in thicker slag deposits and, consequently, greater insulating effects. A proper knowledge of slag characteristics and of the behavior of slag in the furnace should permit the design of furnaces having predictable heat-absorbing characteristics when fired with coal having ash of known composition and would make it possible to select coals upon the basis of slag performance rather than upon the now generally used guide of the cone-softening temperature.

Only recently has that fundamental information been obtained which makes possible a critical analysis of the action of slag on heat-absorbing surfaces. The development in 1940 by the Bureau of Mines of a high-temperature viscometer[4] for measuring the flow properties of coal-ash slag at temperatures where the slag behaved as a true liquid and, in 1943, of a modification,[5] permitting study of the region where the slag behaved as a plastic rather than a viscous material, furnished methods of obtaining the required data. A preliminary discussion of a means of calculating the thickness of slag deposits on heat-absorbing surfaces based on the flow properties of the slag was given in the second report,[6] but its importance seemed to warrant additional explanation. Consequently, the study was carried further

The present paper reviews the information included in a forthcoming Technical Paper of the Bureau of Mines,[6] in which the subject is treated in considerable detail and in which the mathematical derivations of the complex relationships are given. The present paper describes the principal elements included in that publication to show the factors which are of importance in fixing the thickness of slag deposits on heat-absorbing surfaces but does not include specific details as to the means of arriving at the relationships obtained. It is expected that much of the information included will be used in a field study of the action of coal ash in boiler furnaces now being conducted in co-operation with the Special Research Committee on Furnace Performance Factors; thus the present paper may be considered as a preliminary report to that committee. Others interested in the effect of slag on heat transfer in boiler furnaces may find the method advantageous over purely speculative schemes based upon empirical considerations.

The factors involved in this study can be divided into two classes, one fixed by the properties of the slag and the other dependent on furnace conditions; one example of this is the rate of supply of slag to the wall from the flame. Factors depending on the flow properties of the slag will be discussed first.

Properties of Slag Affecting Its Behavior on Furnace Walls

This report refers only to the deposition of slag on the walls of that part of the furnace where the slag surface is molten and a continuous surface flow of slag occurs. On other sections of the furnace, such as on generating tubes, where adherent but nonflowing deposits are formed, or where the deposits break off after reaching considerable thickness, the conditions are less susceptible to analysis and will not be considered here. Because of the

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[1] Published by permission of the Director, Bureau of Mines, United States Department of the Interior.

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[4] "Viscosity of Coal-Ash Slags," by P. Nicholls and W. T. Reid, Trans. A.S.M.E., vol. 62, 1940, pp. 141–153.

[5] "The Flow Characteristics of Coal-Ash Slags in the Solidification Range," by W. T. Reid and P. Cohen. Published in pamphlet, "Furnace Performance Factors," The American Society of Mcehanical Engineers, May, 1944.

[6] "The Flow of Coal-Ash Slag on Furnace Walls," by P. Cohen and W. T. Reid, Tech. Paper 663, Bureau of Mines, in press.

high rate of heat transfer in the slagging zone, an analysis confined to that part of the furnace is most important.

Physical Structure of Slag Deposits. Examination of a section of slag, taken from the wall of a slag-tap furnace in the slagging zone, often gives a visual indication of the flow process. Fig. 1 shows a representation of a typical deposit. Immediately adjacent to the surface of the tube is a thin, semiporous, irregular layer of partly melted ash, through which the slag adheres to the tube. Next, is a dense, crystalline layer of slag, which usually makes up the largest part of the deposit. Finally, there is a layer of glassy slag which extends to the surface, the boundary between it and the crystalline layer usually being sharply defined. The arrangement of these layers results from the temperature gradient through the slag which existed during operation of the furnace, and does not result from events subsequent

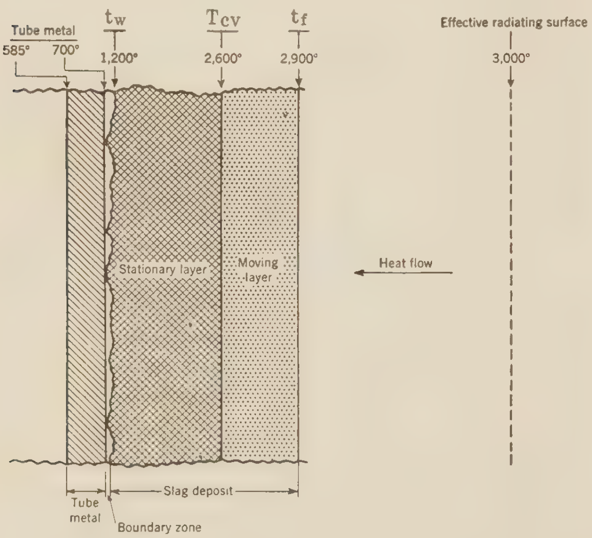

Fig. 1 Temperature Distribution in Slag Deposit

to taking the furnace off the line, in which case the glassy layer would have been next to the tube surface, having there been cooled most rapidly. It is evident, then, that when a furnace is operating, the part of the slag deposit nearest the tube is maintained at a sufficiently low temperature for the slag to crystallize, whereas on the furnace side the slag is molten.

Flow Properties of Coal-Ash Slags. Laboratory studies have shown that coal-ash slags are not true liquids over a wide range of temperatures. Most slags on being cooled slowly from high temperatures where they exist as true liquids, at some point, depending upon the composition of the slag and the state of oxidation of its iron content, and somewhat on the thermal history of the melt, begin to separate solid material from the otherwise liquid slag. This solid matter, usually present as very small crystals, causes the slag to become plastic rather than viscous. The essential characteristics of this plastic state are that a definite shear stress (the internal yield stress) is necessary to produce flow and that the rate of flow is proportional to the excess of the applied shear stress over this minimum value. The temperature at which this transition occurs is called the "temperature of critical viscosity," T_{cv}; values of this constant have been determined for a wide range of composition of coal-ash slags, including variation in the state of oxidation of the iron in the slag that is significant. Because both the internal yield stress and the plastic viscosity increase rapidly with decrease in temperature below T_{cv} and, furthermore, at low applied shear stress the effect of even slight internal yield stress is such as to make the apparent viscosity very high, the important assumption is made that the flow of slag at temperatures below T_{cv} is insignificant as compared with flow at higher temperatures. In other words, for all practical purposes where the applied stress is low (as by the action of gravity), the point at which the temperature of critical viscosity occurs in a slag deposit represents an abrupt change in physical properties; slag at a lower temperature behaves as a solid, while hotter slag flows like any liquid; the slightest application of force producing a corresponding flow. Thus it is reasonable to believe that the demarcation between the crystalline stationary layer and the glassy flowing layer of slag on furnace walls occurs at the temperature of critical viscosity.

In the flowing liquid layer of slag, the low-temperature boundary will be at the temperature of critical viscosity, while the high-temperature boundary will be at the furnace temperature; thus usually there will be a temperature difference of several hundred degrees between the two sides of the layer. The viscosity of the liquid slag and its variation with temperature will determine the thickness of the liquid layer required to carry off the slag being supplied. Data obtained from a great many determinations of the viscosity of coal-ash slags in the liquid region, that is, at temperatures higher than T_{cv}, show that the variation of viscosity with temperature can be expressed in the form $\eta^{-z} \doteq At - B$ where η is the viscosity in poises; Z and A are constants for the entire field of composition normally found and are equal numerically to 0.1614 and 0.000452, respectively; t is the temperature in degrees F; and B depends on the composition of the slag. Knowing the viscosity of any slag at a particular temperature, B can be computed from the known values of Z and A.

The flow properties important in fixing the thickness of a slag deposit on a heat-absorbing surface are, therefore, (1) the temperature of critical viscosity, because it fixes the lower temperature of the flowing layer and determines the fraction of the total deposit that effectively will be solid and thus can be considered to be fixed in position; (2) the relationship between temperature and viscosity of the liquid layer, which will control the rate at which the slag can flow to cause a decrease in the over-all thickness. Obviously, as the temperature of critical viscosity increases, the fraction of the total deposit of slag which is stagnant will also increase, and the thickness of the layer will be greater. Likewise, as the viscosity of the liquid layer of slag increases at any fixed temperature, the rate of flow of the slag will be decreased, and the slag thickness will be greater. Thus if these properties of the slag are known from measurements made in the viscometer and correlated with composition, it is possible to predict actions of the slag in the furnace.

Effect of Furnace Conditions

Factors other than the fundamental flow properties of coal-ash slags are important in fixing the thickness of slag on heat-absorbing surfaces. These include (1) temperatures in the furnace and of the slag deposit; (2) the quantity of slag being supplied to the surface from the furnace gases per unit of time; (3) the height in the furnace at which flow begins; (4) the height at which the thickness of slag is to be considered; (5) the angle of inclination of the flowing slag (essentially that of the furnace wall); and (6) the rate of heat transfer through the layer of slag. Additional factors dependent upon slag properties other than flow include (1) the thermal conductivity of the slag, and (2) the density of the slag.

The importance of temperature is obvious, because the flow properties of any given slag depend principally on that factor. Further, assuming that the temperature gradient through the slag deposit is linear, the thickness of the flowing layer of slag

will depend on the temperature of its boundaries, this layer then being a definite fraction of the total thickness of the deposit. For instance, if the furnace side of the slag deposit is at a temperature of 2900 F, the tube side is at 1200 F, and the temperature of critical viscosity is 2600 F, the flowing layer is $\frac{2900 - 2600}{2900 - 1200} =$ 0.18 of the total thickness of the deposit, and the stationary layer is 0.82 of the total thickness. In this case the total thickness of the deposit will be 5.6 times that of the flowing layer.

If the furnace is operating under steady conditions, so that equilibrium occurs and no change takes place in any factor, the volume of slag flowing past a given position in the furnace must equal the total volume of slag deposited on the furnace wall in the slagging zone above that position. Therefore, the quantity of slag being deposited from the flame is important. It is also evident that the height of the slagging zone in the furnace, and the position of the area being considered, will affect the quantity of slag flowing past any given point. Because the quantity of slag reaching the wall is difficult to determine, comparisons in this report are based upon an arbitrary quantity of slag flow, and two different slags are compared by considering the ratio of the thicknesses that would be obtained for the same flow of slag in each case. By taking this ratio for all slags relative to a "standard" slag and fixed furnace conditions, a set of relative thicknesses is obtained.

The angle of the flowing layer of slag will affect its thickness in a simple manner. Although wall tubes are usually vertical, this factor has been included to permit consideration of special cases.

The point of major interest is of course the rate of heat transfer from the furnace through the slag to the tubes. For a given temperature drop through the deposit, the rate of heat transfer is proportional to the reciprocal of the total thickness of slag, the proportionality factor being the thermal conductivity. Lacking any information at present on the thermal conductivity of coal-ash slags, this factor can be assumed to be independent both of composition and temperature, and the relative rates of heat transfer of the deposits can be obtained from their relative thickness. Lack of information on absolute values of rate of slag deposition, thickness of the deposit, or the rate of heat transfer need not detract from the usefulness of this study; when the data are available, the absolute values can be computed by these same methods.

Equation for Computing Thickness of Slag Deposits

The assumptions made in deriving the equation for the thickness of slag on heat-absorbing surfaces are (1) the flow of slag at temperatures below that of the critical viscosity is insignificant; (2) the thermal gradient through the slag deposit is constant; (3) the slag density is independent of composition and temperature; and (4) the slag exists on a plane surface and heat transfer occurs normal to that surface.

The equation, the derivation of which is given in another publication,[6] is long and involved, including, as it does, approximately 30 parameters. Those interested in the mathematical method may wish to follow the derivation step by step; others interested solely in the action of slag in furnaces can better understand its usefulness by referring to the figures given later. The equation is

$$D = \frac{\dfrac{0.00115\ \Delta t}{(\rho\ \sin\alpha)^{1/3}}\left(\displaystyle\int_h^{h_m} V_S\, dh\right)^{1/3}}{[\quad]^{1/3}}$$

where

$$[\quad] = \left[\left(\frac{\omega + A'\Delta t}{7.20}\right)\left\{\frac{(\omega + A'\Delta t)^{8.20} - (\omega + A'\beta\Delta t)^{8.20}}{8.20} - (\omega + A'\beta\Delta t)^{7.20}(1 - \beta)(A'\Delta t)\right\} - 0.122\left\{\frac{(\omega + A'\Delta t)^{9.20} - (\omega + A'\beta\Delta t)^{9.20}}{9.20} - (\omega + A'\beta\Delta t)^{8.20}(1 - \beta)(A'\Delta t)\right\}\right]$$

The nomenclature used is as follows:

t_f = temperature, deg F, of furnace side of slag deposit
t_w = temperature, deg F, of wall side of slag deposit
T_{cv} = temperature of critical viscosity, deg F, below which no flow is assumed to occur
Δt = $t_f - t_w$, temperature drop across slag deposit, deg F
D = total thickness of slag deposit, in.
ρ = density of slag, assumed constant, lb mass per cu ft
h_m = height in furnace at which flow begins, above datum line, ft
h = height above datum line in furnace, ft
V_s = volume of slag supplied to wall at height h, ft³ per ft² per sec
V = volume of slag flow at height h, ft³ per ft² per sec
α = angle of inclination of wall from horizontal, deg
β = $\frac{T_{cv} - tw}{\Delta t}$ = fraction of deposit which does not flow
A' = 1.546A = 0.000699
B' = 1.546B
ω = $A't_w - B'$
t_g = temperature of gas, deg F
T_g = temperature of gas, deg R
t_a = adiabatic combustion temperature, deg F
T_f = temperature of furnace side of slag deposit, deg R

Application of Equation

Effect of Change in Viscosity and Temperature of Critical Viscosity. The effect on the thickness of slag deposits caused by change in the flow properties of the slag is shown in Fig. 2; the data were calculated by the equation just described. This and

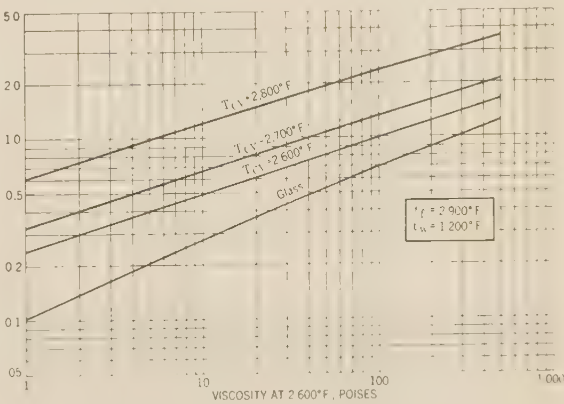

Fig. 2 Relative Thickness of Slag Deposits for Temperature of Furnace Side of Slag Deposit of 2800 F

the succeeding plots express the thickness of slag in terms of the relative thickness compared with a "standard" slag having a viscosity of 100 poises at 2600 F, and a temperature of critical viscosity of 2600 F. They show that the effect of change in viscosity is only slightly greater than change in the temperature of critical viscosity, T_{cv}. For instance, with a slag for which T_{cv} is 2600 F, decreasing the viscosity from 100 to 10 poises, as measured at 2600 F, decreases the slag thickness to 48 per cent of the

original thickness and, if the viscosity were reduced to 1 poise, to 23 per cent. However, if T_{cv} is changed from 2600 F to its minimum value, which would be the case for a slag which acted as a glass, the thickness would be 55 per cent of the original if the viscosity were maintained at 100 poises and 31 per cent if the comparison were made at a viscosity of 1 poise for both slags. It should be noted that as T_{cv} approaches the temperature of the furnace side of the slag deposit, there is a sharp increase in relative thickness at all viscosities, whereas when T_{cv} decreases to approximately 2000 F, the properties approach that of a glass.

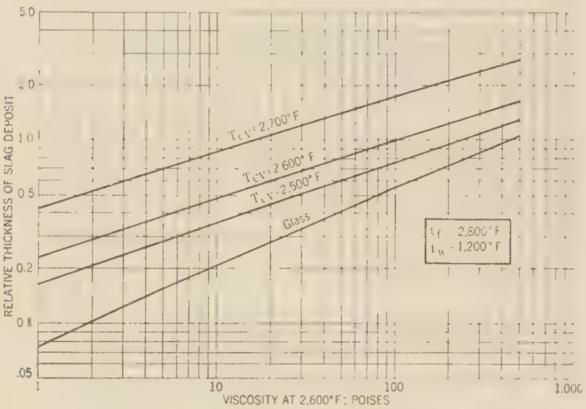

Fig. 3 Relative Thickness of Slag Deposits for Temperature of Furnace Side of Slag Deposit of 2900 F

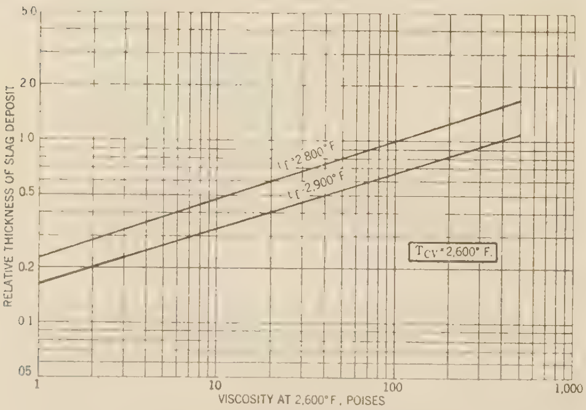

Fig. 4 Relative Thickness of Slag Deposits for Temperature of Critical Viscosity of 2600 F

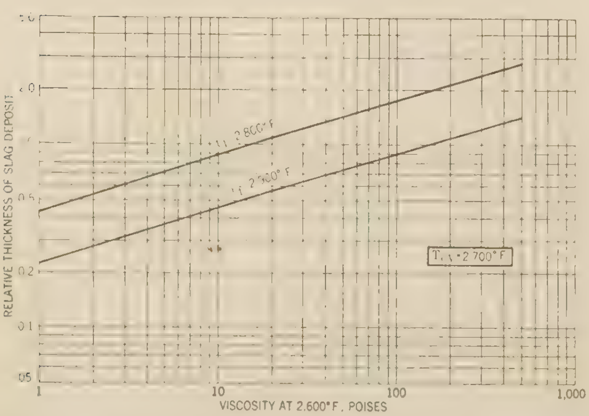

Fig. 5 Relative Thickness of Slag Deposits for Temperature of Critical Viscosity of 2700 F

Effect of Change in Furnace Conditions. Changing the rate of heat release in a furnace will affect the temperature of the furnace side of the slag deposit, t_f. Fig. 3 shows the result of increasing t_f to 2900 F, the principal change being a slight decrease in the spacing of the curves. Viscosity and temperature of critical viscosity are again approximately of equal significance in affecting the thickness of the slag deposit.

A different method of expressing the same data is shown in Fig. 4, where the effect of change in furnace temperature on thickness is shown for slags of varying viscosity but with a fixed temperature of critical viscosity of 2600 F. It is apparent that increasing t_f from 2800 to 2900 F decreased the thickness at 100 poises to 66 per cent of the original thickness, and that the percentage decrease of thickness is approximately the same for a given change in t_f independent of the viscosity of the slag. In Fig. 5, where the same type of plot is used, except that T_{cv} is increased to 2700 F, similar effects are observed, except that increasing t_f from 2800 to 2900 F decreased the relative thickness at 100 poises from 1.8 to 0.9, or to 50 per cent of the original thickness. Again, because the isotherms are parallel lines, this percentage change will be constant over the range of viscosities studied.

As noted previously, both the viscosity and the temperature of critical viscosity of coal-ash slags have been determined experimentally for a wide range of compositions. These data have been correlated so that knowledge of the chemical analysis of the slag permits prediction of these properties. Such correlations have been used in Fig. 6, in expressing the relative thickness of slag deposits as a function of composition. Thus knowing the composition of two slags being deposited on the wall tubes of a furnace under identical conditions, a prediction can be made of the relative thickness of the two deposits.

Fig. 6 shows that increase in the equivalent Fe_2O_3 content of the slag results in decreased thickness and that decreasing the ferric percentage, that is, increasing the state of reduction of the iron oxides, decreases the thickness of the deposit. The action of CaO is also to decrease the thickness of the slag as its content increases, the effect being greatest with slags containing less than 20 per cent equivalent Fe_2O_3. For instance, with a slag containing 15 per cent equivalent Fe_2O_3, at a ferric percentage of 20, the thickness with 10 per cent CaO is 44 per cent of that for a slag having 5 per cent CaO, while at 30 per cent equivalent Fe_2O_3 the thickness would be 73 per cent that for a slag containing 5 per cent CaO. Although these relationships may appear complex, by reference to this figure they can be made relatively simple. It is important to note that the use of this figure is limited to slags having a silica : alumina ratio of 2; experimental work now in progress will extend it later to cover a wider range of compositions, including most commonly occurring slags.

Relative Rates of Heat Transfer Through Slag Deposits. The relative rate of transfer of heat through two slags is shown in Fig. 7, as a function of the temperature of the furnace side of the slag deposit; the slags had viscosities of 5 and 100 poises, respectively, and a temperature of critical viscosity of 2600 F. It is obvious that normal change in the viscosity of the slag is more important than change in furnace temperature in affecting the rate of heat transfer. Also, as the furnace temperature approaches the temperature of critical viscosity, the rate of heat transfer will approach zero. This is true of course because under these conditions the thickness of slag increases markedly; because such slag deposits probably would not be physically strong enough to build up indefinitely, this region of the curve has theoretical significance only.

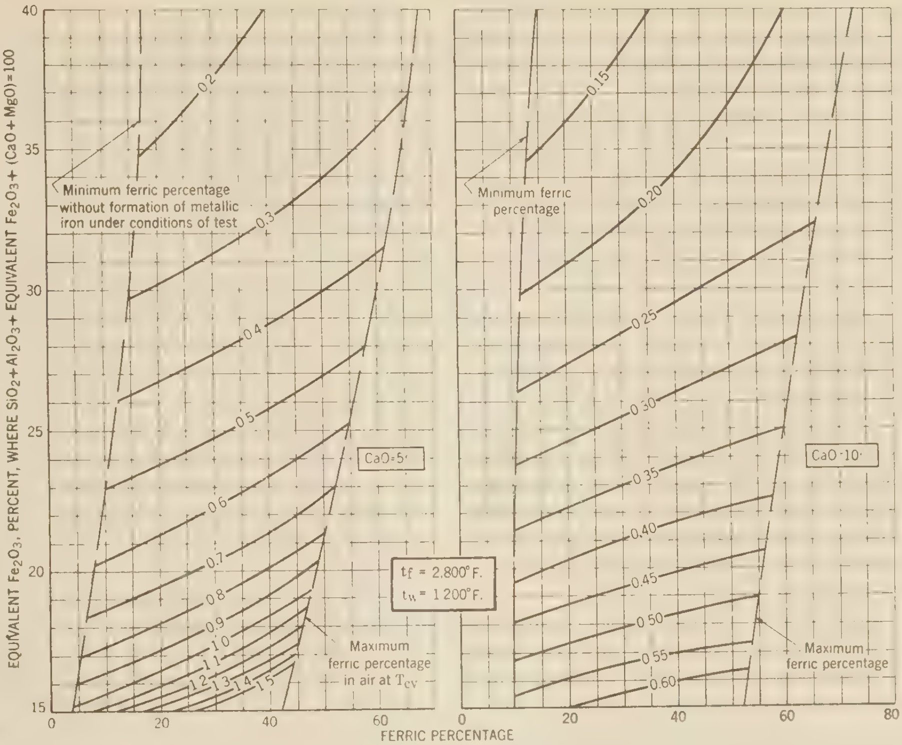

Fig. 6 Relative Thickness of Slag Deposits as Function of Composition and Ferric Percentage for a Silica: Alumina Ratio of 2

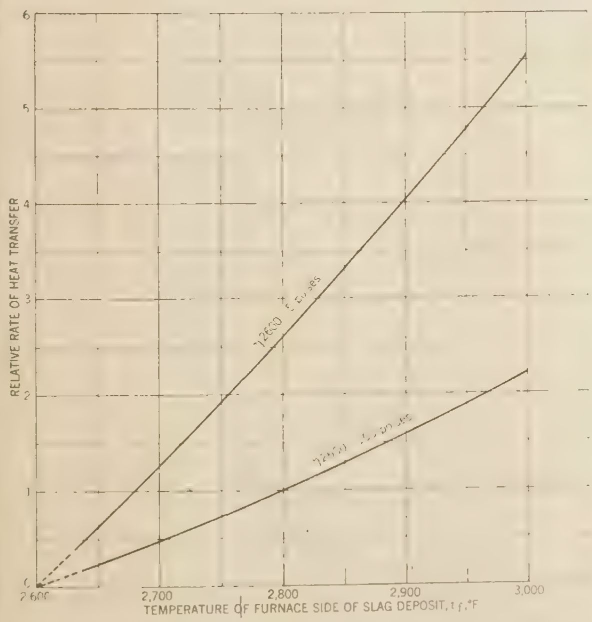

Fig. 7 Relative Rates of Heat Transfer for Two Slags Having Temperature of Critical Viscosity of 2600 F

Effect of Slag on Rate of Heat Transfer Under Furnace Conditions

The principal interest in the thickness of slag on furnace-wall tubes, of course, is based upon its effect on the rate of transfer of heat. This has already been described briefly, but for a model furnace where conditions are assumed constant over the entire area under study; actually, in an operating furnace they would vary from point to point in an unknown manner. The problem of taking these variations into account would be enormous, even if the distribution of thermal energy were known or if the rate of supply of slag to the walls could be accurately determined. Also, if the conditions in the furnace were uniform throughout, comparison between different slags would be complicated by the fact that any change would result in a different rate of heat transfer, and the temperatures in the furnace would be different unless there had been some accidentally correct change in the rate of firing. However, a means of making the necessary corrections is available and will be described briefly.

A method of determining the heat absorbed by furnace walls has been described by Wohlenberg and others[7] who measured the temperature of furnace-exit gases with high-velocity thermocouples. Mullikin,[8] using the Stefan-Boltzmann law, determined the fraction of the total heat supplied which was absorbed in the furnace and estimated the effects of the slag accumulation. More recently Bailey[9] studied the effect of slag on the heat absorption in steam-generating units, using the cone-fusion temperature as a measure of the flow properties of the slag. He obtained a relationship, called the "gas-ash temperature graph," between the properties of the slag and the temperature measured in several typical furnaces.

[7] "An Experimental Investigation of Heat Absorption in Boiler Furnaces," by W. J. Wohlenberg, H. F. Mullikin, W. H. Armacost, and C. W. Gordon, Trans. A.S.M.E., vol. 57, 1935, paper RP-57-4, pp. 541–553.

[8] "Evaluation of Effective Radiant Heating Surface and Application of the Stefan-Boltzmann Law to Heat Absorption in Boiler Furnaces," by H. F. Mullikin, Trans. A.S.M.E., vol. 57, 1935, paper RP-57-2, pp. 517–529.

[9] "Modern Boiler Furnaces," by E. G. Bailey, Trans. A.S.M.E., vol. 61, 1939, pp. 561–576.

The method used in the present investigation is described in detail in Appendix II of reference 6. Briefly, it may be described as follows: If all the heat transfer in the slagging zone is assumed to occur by radiation, then the sensible-heat loss of the furnace gases between the adiabatic temperature t_a, and the furnace gas temperature t_g, must equal the heat transfer by radiation from the furnace gases at temperature t_g to the furnace wall at temperature t_f. Further, this must equal the heat conducted through the slag layer from the furnace side at temperature t_f to the tube side at temperature t_w. If the sensible-heat loss is proportional to $t_a - t_g$, the heat transfer by radiation to $T_g^4 - T_f^4$, the capitals denoting absolute temperatures, and the heat transfer by conduction to $\frac{t_f - t_w}{D}$, then these three quantities are mutually proportional, and a change in any one factor will result in changes in the others, which can be calculated. Thus in a given furnace, if a change is made in the coal fired so that the thickness of slag deposited on the wall tubes is decreased, other things being equal, the heat transfer through the slag will be increased, and, as a result, the furnace temperature will be lowered. Knowing the expected change in slag thickness resulting from a change in composition of the ash in the coal, therefore, allows prediction of the effect on furnace performance.

Summary

The increasing importance of slag deposits to the designer and operator of slag-tap pulverized-coal boiler furnaces has created a demand for more quantitative knowledge of the behavior of slag under furnace conditions. Recent developments in the study of these slags have furnished data from which the thickness of slag deposits can be calculated for idealized conditions.

An equation is given relating the thickness of the slag deposit to the flow properties of the slag, the temperature of the hot and cold sides of the deposit, and the volume of slag flow. The relative thickness of slag deposits for typical conditions can be calculated from this equation, thus providing a method of comparing the behavior of different coal ashes on furnace walls.

It is shown that for fixed furnace conditions, increase in the temperature of critical viscosity, that is, the temperature at which the slowly cooled slag changes from the liquid to the plastic state, and increase in the viscosity of the slag cause approximately equal increases in the thickness of the slag deposit. As the temperature of critical viscosity approaches the temperature of the furnace side of the slag deposit, there is a large increase in the thickness for all viscosities. It is shown that, for a fixed temperature of critical viscosity, there is an equal percentage decrease in the thickness of the deposit for all viscosities as the temperature of the furnace side of the slag deposit is increased.

The relationship between slag composition and relative thickness of deposit is shown for part of the field of composition of coal-ash slags. Increase in equivalent Fe_2O_3 content and decrease in ferric percentage decrease the thickness of the deposit; CaO also decreases the deposit thickness as its content increases, the effect being greatest with slags containing less than 20 per cent equivalent Fe_2O_3.

Change in viscosity of slag due to change in composition of ash is shown to be more important than normal change in furnace temperature in affecting the rate of heat transfer.

Reference is made to a method of determining the effect on furnace temperatures of changes in the rate of heat transfer through the slag deposit, thus permitting the correlation of all the factors involved. Although the method applies strictly only to a model furnace, its application to operating furnaces should assist in studying the part taken bv slag in affecting furnace performance.

Draft-Gear Action in Train Service

By O. R. Wikander,[1] Pittsburgh, Pa.

In order to investigate the most desirable characteristics of a draft gear by mathematical analysis, a study has been made of the mechanics of an elastic bar, which is subjected to external forces that correspond to those acting in trains under various service conditions. The results of this study, including the derivation of the various equations governing the behavior of an elastic bar, solid or with free slack, are given in another paper by the author.[2] The present paper contains substantially the same equations but transcribed so as to apply to railway trains. Numerical examples are given showing the applications of these equations to assumed test trains.

Definition of Terms

The "capacity" of a draft gear is the amount of energy required to close it solid in drop test under a 27,000-lb tup, ft-lb.

The "recoil energy" of a draft gear is the amount of work, expressed in ft-lb or as a fraction or percentage of the draft-gear capacity, which the gear performs when it recoils the full distance of its service travel.

The "free slack" per coupling is the maximum relative movement between two adjacent cars, which can be made without compressing the draft gears.

The "controlled slack" per coupling is the amount of movement between adjacent cars, which is resisted by draft-gear action.

The "residual slack" per coupling is the amount of controlled slack which is released between two adjacent cars, when, during a train operation, a previously compressed train is stretched, or a previously stretched train is compressed. (The effect is substantially that of increasing the free slack by the amount of the residual slack.)

The "propagation speed" is the speed at which a force or vehicle speed is transmitted through a train without free slack.

The "rate of progressive force application" is the speed at which a serial force application progresses through a long train.

The "propagation ratio" is the ratio of the propagation speed to the rate of progressive force application.

The "passing point," existing under certain conditions in elastic bars or long trains with free and residual slack, is the point at which the slack take-up passes the force application, and is defined by the number of the car at which it occurs, counted from the end of the bar or the train, at which the progressive external forces were originally applied.

The "critical train slack" is the amount of free plus residual slack per coupling which, in case of emergency-brake application, will place the passing point at the rear end of a train. A decrease below this amount of slack will move the passing point toward the front end of the train, and an increase will move it to an imaginary point beyond the rear end of the train.

[1] Mechanical Engineer, Ring Spring Department, Edgewater Steel Company. Mem. A.S.M.E.

[2] "Dynamics of an Elastic Bar," by O. R. Wikander. To be presented at the Annual Meeting, Nov. 27–Dec. 1, 1944, of the A.S.M.E.

Contributed by the Railroad Division and presented at the Semi-Annual Meeting, Pittsburgh, Pa., June 19–22, 1944, of The American Society of Mechanical Engineers.

Equations Governing Train Dynamics

The following nomenclature will be used:

C = capacity of one draft gear, ft-lb
e = base of natural logarithms
F = retarding force per car, lb
L_1 = length per car between buffing surfaces, ft
M = mass per car, slugs
M_1 = mass of locomotives and tender, slugs
n = number of cars or number of car in train
n_1 = number of car at passing point
n_2 = number of car of maximum coupler pressure in train without free slack
P = drawbar pull of a locomotive, lb
P_1 = tractive effort of a locomotive, lb
p = propagation ratio
q = full travel per draft gear, ft
s_1 = free plus residual slack per coupling, ft
s_2 = critical free plus residual slack per coupling, ft
t_1 = time interval between emergency brake or other force applications on adjacent cars, sec

Starting a Train

It will be assumed for the present that the train is started down a hill of such a grade that the frictional and gravitational forces balance. The entire power of the locomotives is thus applied to acceleration of the masses. Different equations apply, depending upon whether the train is started stretched or with free slack, and also whether the drawbar pull or the tractive effort of the locomotive is assumed to be constant during the starting period.

In any stretched train the drawbar pull P will be transmitted through the train at a constant propagation speed V which may be expressed as

$$V = (L_1/q)\sqrt{(C/M)} \quad [1]$$

Case 1. A locomotive of constant drawbar pull starts a train without free slack:

In this case the pickup speed v of the individual cars is also constant and may be expressed as

$$v = Pq/\sqrt{(CM)} \text{ or } P = (v/q)\sqrt{(MC)} \quad [2]$$

The time t required to set the train of n cars in motion, or the duration of the starting period, is

$$t = nq\sqrt{(M/C)} \quad [3]$$

The distance D traveled by the locomotive during the time t may be expressed as

$$D = nPq^2/C \quad [4]$$

After the time t, the train as a whole would, in this as well as in the following three cases, accelerate at the rate of $P/(M_1 + nM)$ if no external forces were impeding its motion.

Case 2. A locomotive of constant tractive effort starts a train without free slack:

During the starting time t, between the application of the tractive effort P_1 and until the last car begins to move, the speed of the locomotive increases according to the equation

$$v = [P_1q/\sqrt{(MC)}][1 - e^{-t\sqrt{(MC)}/(qM_1)}] \quad [5]$$

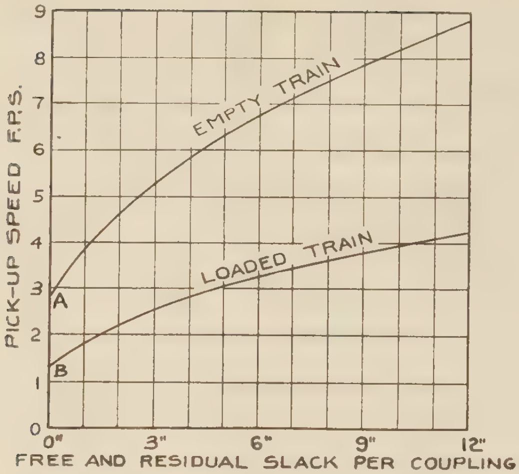

FIG. 1 PICK-UP SPEEDS OF EMPTY AND LOADED TRAINS AT CONSTANT DRAWBAR PULL

The duration t of this starting period is given in Equation [3].

Substituting this value in Equation [5], the latter takes the form

$$v = [P_1q/\sqrt{(MC)}][1 - e^{-nM/M_1}] \quad \text{[5a]}$$

This equation expresses the speed of the locomotive at the time when car n begins to move.

The drawbar pull at that time is

$$P = P_1(1 - e^{-nM/M_1}) \quad \text{[6]}$$

Case 3. A locomotive of constant drawbar pull starts a train with free slack:

The pick-up speed of the individual cars is constant and can be expressed as

$$v = [P_1q/\{2\sqrt{(CM)}\}][1 + \sqrt{\{1 + 4s_1C/(P_1q^2)\}}] \quad \text{[7]}$$

The duration of the starting period or the time t at which car n is set in motion is

$$t = [nq\sqrt{M}/(2\sqrt{c})][1 + \sqrt{\{1 + 4s_1C/(P_1q^2)\}}] \quad \text{[8]}$$

The distance D traveled by the locomotive during the starting time t or until the last car is picked up may be expressed as

$$D = n[s_1 + (P_1q^2/\{2c\})(1 + \sqrt{\{1 + 4s_1C/\{P_1q^2\}\}})] \quad \text{[9]}$$

Case 4. A locomotive of constant tractive effort starts a train with free slack:

The pickup speeds v of the individual cars increase with the number of the car n counted from the locomotive. The relationship between v and n under the assumption that all bunched cars have a uniform speed is expressed by the equation

$$n = (M_1/M)\{[\sqrt{(s_1C)}/\sqrt{(s_1C + P_1q^2\langle x - x^2\rangle)}]$$
$$[(a^2 + 2ax + 2x - 1)/(a^2 - 2ax + 2x - 1)]^{1/(2a)} - 1\} \quad \text{[10]}$$

where $x = v\sqrt{(CM)}/(P_1q)$ is a parameter and $a = \sqrt{(1 + 4s_1C/[P_1q^2])}$ is a constant.

This equation permits the computation of n for assumed values of v and makes it possible to plot the vn curve.

The duration of the starting period t for a train with n cars may be expressed as

$$t = (v/P_1)(M_1 + Mn) \quad \text{[10a]}$$

where v has the value on the vn curve corresponding to the number of the car n. For large values of n the value of v from Equation [7] gives a good approximation. Substituting this latter value in Equation [10a] gives

$$t = [(M_1 + Mn)/\{2\sqrt{(CM)}\}][1 + \sqrt{\{1 + 4s_1C/(P_1q^2)\}}] \quad \text{[11]}$$

The distance traveled by the locomotive during the starting period may be expressed as

$$D = s_1n + Aq\sqrt{(M/C)} \quad \text{[12]}$$

where A represents the area under the vn curve from the origin to the car number n.

BRAKING A TRAIN

The following equations, governing the braking of a train, apply also, if the proper constants are used, to its behavior during a change in the grade of the track:

A train constant, which is very important since it affects practically every characteristic of a train, is its "propagation ratio" p.

The value of p is computed from the equation

$$p = (t_1/q)\sqrt{(C/M)} \quad \text{[13]}$$

Braking a Train Without Free Slack. If the train is bunched when the brakes are applied from the front or stretched when they are applied from the rear, its behavior will correspond to that of a solid elastic bar to which uniformly distributed external forces are serially applied.

It will be different, depending upon whether the propagation ratio is higher than, equal to, or lower than unity and these three cases will therefore be treated separately.

Case 1. Propagation ratio is higher than unity: The relative impact speed between adjacent cars in such a train is zero, since the compression speed between adjacent cars gradually increases from zero.

The time required for the brake application to reach the first car in the train will be designated as t_1, which is also the time interval between the brake application on adjacent cars; hence, the brakes will be applied to car n at the time nt_1. Maximum gear compression and coupler pressure is reached at the time

$$t = 2nt_1/(p + 1) \quad \text{[14]}$$

This maximum coupler pressure may be expressed as

$$P_m = 2Fnp/(p + 1)^2 \quad \text{[15]}$$

It is located at car number n_2, counted from the locomotive; it is

$$n_2 = 2n/(p + 1) \quad \text{[16]}$$

The contraction of the train during the time t is

$$D = (Fq^2n^2/c)(p/[p + 1]^2) \quad \text{[17]}$$

Case 2. Propagation ratio is equal to unity:

The maximum coupler pressure amounts to

$$P_m = Fn/2 \quad \text{[18]}$$

It occurs at the car number $n_2 = n$, counted from the front end of the train.

The relative car impact speed, due to the instantaneous rise of pressure is

$$v = Fqn/[2\sqrt{(CM)}] \quad \text{[19]}$$

The time t required for the coupler pressure and for the force application to reach the end of a train of n cars is

$$t = nt_1 \quad \text{[20]}$$

The contraction of the train during the time t is

$$D = Fq^2n^2/(4C) \quad \text{[21]}$$

Case 3. Propagation ratio is lower than unity:

The maximum coupler pressure in such a train amounts to

$$P_m = Fnp/2 \quad [22]$$

and occurs at car number n_2 from the front end, expressed as

$$n_2 = n(p + 1)/2 \quad [23]$$

The relative impact speed between adjacent cars is zero. The time t required for the maximum coupler pressure to form may be expressed as

$$t = nt_1(p + 1)/(2p) \quad [24]$$

The contraction of the train at the time t may be expressed as

$$D = Fq^2n^2p/(4C) \quad [24a]$$

Braking a Train With Free Slack. If a train is stretched when the brakes are applied from the front, or bunched when they are applied from the rear, a certain amount of free slack will be taken up during the braking period.

In such a train the conditions will be different, depending upon whether the car under consideration is located:

1 Before the passing point or in a train which has no passing point.
2 At the passing point.
3 Beyond the passing point.

These three cases will be considered separately.

Case 1. Conditions in a train without a passing point or before that point is reached:

The relative car impact speed v at which the car $(n + 1)$ collides with a string of n bunched cars is

$$v = Fnt_1/(2M) \quad [25]$$

The time t' at which the car number $(n + 1)$ collides with the string of n bunched cars is

$$t' = s_1M/(Ft_1) + nt_1(2p + 1)/(3p) \quad [26]$$

According to this equation the time t', plotted against the car number n, is a straight line, beginning at the point $n = 0$, $t' = s_1M/Ft_1$ and rises from the n axis at a slope of $t_1(2p + 1)/(3p)$. For $p = 1$, nt_1 and t' are represented by parallel lines, while for $p < 1$ these lines are divergent and in neither case has the train a passing point.

For $p > 1$ and only in that case, the lines are convergent, and the train has a passing point.

The contraction of the train at the time t' is

$$D = s_1^2M/(2Ft_1^2) + s_1n(2p + 1)/(3p) + [Fn^2t_1^2/(2M)]\left(\frac{1}{9} + \frac{5}{18p} + \frac{1}{9p^2}\right) \quad [27]$$

Case 2. Conditions in a train at the passing point:

The location of the passing point at car number n_1 is expressed as

$$n_1 = [Ms_1/(Ft_1^2)][3p/(p - 1)] \quad [28]$$

The relative car impact speed v_1 of the number n_1 car is

$$v_1 = (s_1/t_1)[3p/\{2(p - 1)\}] \quad [29]$$

The time t_2 at which the passing point is reached is

$$t_2 = [s_1M/(Ft_1)][3p/(p - 1)] \quad [30]$$

The contraction of the train at the time t_2 is

$$D_1 = [s_1^2M(Ft_1^2)][3p(4p - 1)/\{4(p - 1)^2\}] \quad [31]$$

Case 3. Conditions in a train beyond the passing point:

In braked trains with a propagation ratio greater than unity and small or moderate amounts of free slack, and in trains with any reasonable amount of slack, subjected to a change in grade of the track, the car impact point will pass beyond the point of force application and from then on the relative car impact speed v will increase in a much slower tempo.

The relationship between the loss in speed of a group of n bunched cars, assumed to travel at a uniform speed, or their relative collision speed with the car number $(n + 1)$, as yet unretarded, is expressed by the equation

$$dv/dn = \sqrt{[2F/(Mt_1)]}[s_1/\sqrt{(vn)} + (t_1/p)\sqrt{(v/n)}] - v/n \quad [32]$$

The solution of this equation is as yet unknown, and it is therefore necessary to use approximative integration, using the Simpson rule or some equivalent method to construct the nv curve for cars beyond the passing point.

From the origin and up to the passing point n_1v_1 the nv curve is computed from Equation [25] and is thus a straight line. At that point it bends sharply to the slope dv/dn expressed by the equation

$$dv/dn = [Ft_1/M][(p + 2)/(6p)] \quad [33]$$

It can be seen from this equation that all the v-n curves have the same slope at the passing point, regardless of the location of the latter which is governed by the amount of train slack.

The time t' required to bunch a group of n cars may be expressed as

$$t' = nt_1 + s_1A_1/L_1 \quad [34]$$

where A_1 is the area in the $\frac{1}{v}$ n diagram, located between the $\frac{1}{v}$ curve and the n axis, and between the verticals at n_1 and n.

The amount of contraction of the train after the bunching of n cars is

$$D = 3Ms_1/t^2 + s_1n + A(t_1/p) \quad [35]$$

where A is the area in the n-v diagram, located between the v-curve and the n axis and between the verticals n_1 and n.

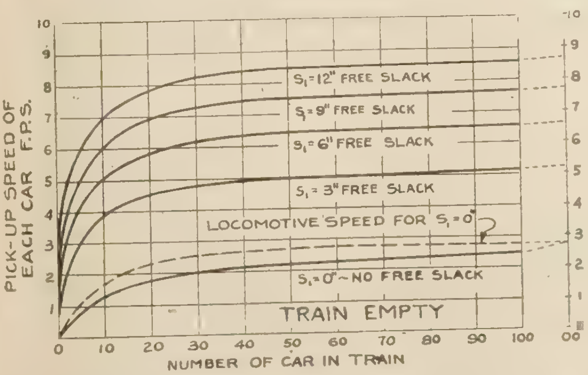

FIG. 2 STARTING EMPTY TRAIN AT CONSTANT TRACTIVE EFFORT

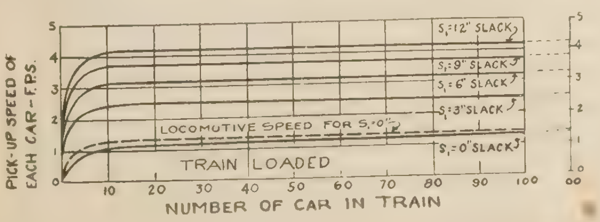

FIG. 3 STARTING LOADED TRAIN AT CONSTANT TRACTIVE EFFORT

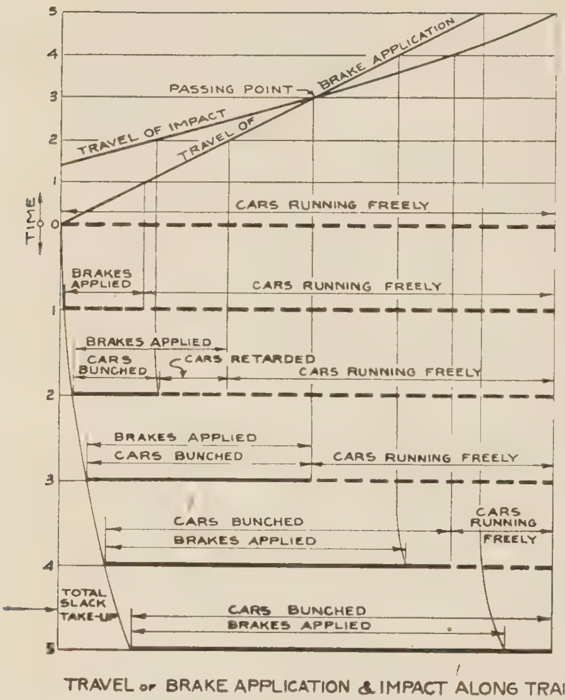

Fig. 4 Travel of Brake Application and Impact Along Train

Brake Action in a Long Train

Fig. 4 illustrates, in diagram form, the conditions under which the brake application and the resulting car impacts proceed in a train with free slack.

The freely running train is represented at the time zero by a horizontal broken line, in which each dash represents a car or a group of cars. Before the application of the brakes, the cars are assumed to be moving uniformly toward the left. The brakes are applied aerially from the front of the train.

The forward motion of the train as a whole is not shown.

The conditions of the train after the periods of time, represented by the horizontals 1 to 5 above the zero line, are shown on the horizontal train diagrams below the zero time line, designated by the same numbers.

Fig. 4 represents a case where the propagation ratio of the train is $p > 1$ and the two inclined diagram lines intersect at the "passing point."

If $p = 1$ these lines are parallel, if $p < 1$ they are divergent, and in neither case has the train a passing point.

Critical Train Slack

Beyond the passing point, the impact speed v increases at a slower rate and the critical slack, defined as the amount of free plus residual slack per coupling, which will place the passing point at the end of the train therefore produces the highest possible car impact speed.

In a train of n cars, the critical slack s_2 may be computed from Equation [28] by substituting $s_2 = s_1$ and $n = n_1$

$$s_2 = [Ft^2n/(3M)]/[(p-1)/p] \qquad [36]$$

In cases where p is equal to or smaller than unity, s_2 is infinite, and the car impact speed is entirely independent of the amount of free or residual slack in the train.

Application of Equations

The use of the equations will be illustrated by a few examples, in which they will be applied to assumed test trains made up of identical cars. The following values are given:

C = capacity of one draft gear, 26,300 ft-lb
g = total travel of each gear = 0.219 ft (= $2^5/_8$ in.)
F = external force acting on each car during emergency-brake application
= 7500 lb, when type K brake is used
= 3750 lb, when type AB brake is used as an initial application, which has the main purpose of bunching a stretched train before the full retarding force of 7500 pounds is applied
M = mass per car
= 1550 for empty car weighing 50,000 lb
= 6500 for loaded car weighing 210,000 lb
M_1 = mass of locomotives and tender
= 17,100 for a combined weight of 550,000 lb
P_1 = tractive effort of locomotive, = 80,000 lb
P = drawbar pull of locomotive, lb
t_1 = time between emergency-brake application on adjacent cars, sec
= 0.092 sec for type K brake
= 0.0612 sec for type AB brake

Example 1. A locomotive of constant drawbar pull starts a stretched train:

In this case the pickup speed v of the individual cars is constant during the whole starting period and can be computed from Equation [2].

For the empty test train, it is represented in Fig. 1, by the vertical OA. For the loaded one it is represented by the vertical OB in the same figure.

Example 2. A locomotive of constant tractive effort starts a stretched train:

In this case the speed v of the locomotive at the time of starting of each individual car is different, depending upon the car number

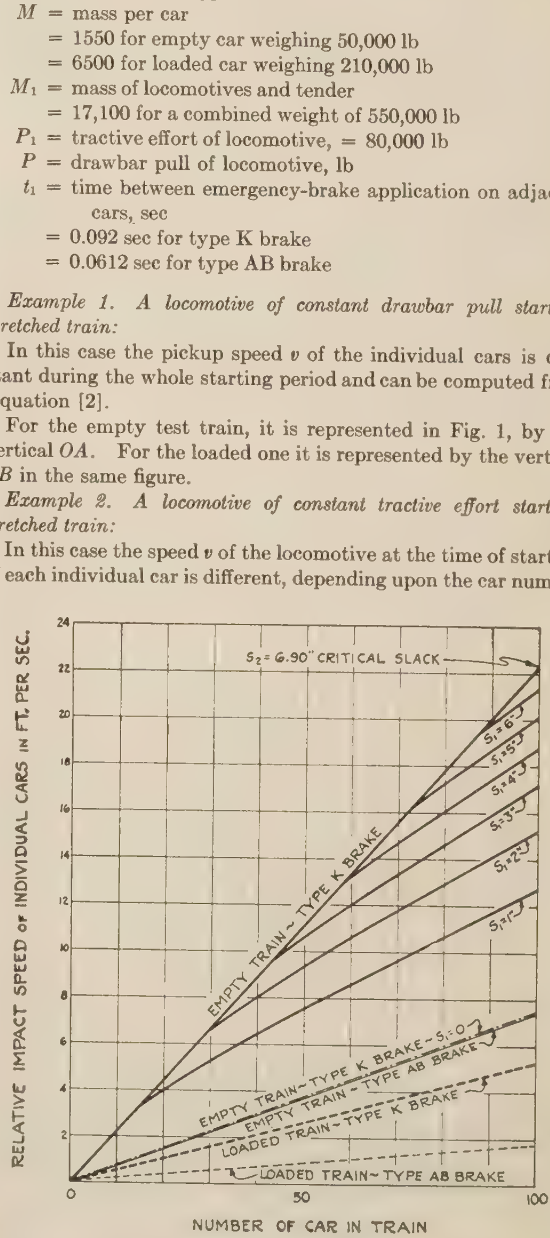

Fig. 5 Relative Car Impact Speeds in Emergency Braking of Long Trains

n counted from the locomotive, and may be computed from Equation [5*a*].

For the empty train, this speed is represented in Fig. 2 by the dotted curve marked "locomotive speed" for $s_1 = 0$. For the loaded train it is represented by the corresponding dotted curve in Fig. 3.

Example 3. A locomotive of constant drawbar pull starts a train with free slack:

For a given train slack, the pick-up speed of the individual car is constant during the entire starting period and can be computed from Equation [7].

The computed car impact speeds for the empty and loaded test trains are plotted in Fig. 1, against the amount of free plus residual slack per coupling. It will be seen from the curves that the constant pickup speed v of the individual cars in empty as well as in loaded trains increases sharply with the amount of free and residual slack between the individual cars.

Fig. 1 shows that the pickup speed of the loaded train is only about one half that of the empty one, but the mass of the individual car is about 4 times larger, and thus the impact energies of the individual cars are the same regardless of whether the cars are loaded or empty.

Example 4. A locomotive of constant tractive effort starts a train with free slack:

Referring to Fig. 2, the pickup speeds of the individual cars, from 1 to 100 in the empty train have been computed from Equation [10], and for various values of the free plus residual slack s_1 from 0 to 12 in. per coupling under the assumption that the tractive effort P_1 of the locomotive is constant.

It is seen that these curves, rising from zero at the first car, rapidly approach horizontal asymptotes, which latter represent the pickup speeds of all the cars in the train under the assumption of a constant drawbar pull P_1 equal to the tractive effort of the locomotive.

It will be seen that beyond car 50, the computed pickup speed would be practically the same regardless of whether the train were started with a constant drawbar pull or an equal constant tractive effort.

Example 5. Computation of the propagation ratios during emergency braking of the assumed test trains:

By substituting the proper constants in Equation [13], the values of p, listed in Table 1, are obtained.

TABLE 1 PROPAGATION RATIOS p FOR VARIOUS TEST TRAINS

	Type K brake	Type AB brake
Empty train, p =	1.73	1.155
Loaded train, p =	0.845	0.563

Example 6. Relative car impact speeds:

How to compute the relative car impact speeds of all the cars in a 100-car train in case of emergency braking from the front.

(*a*) *Empty Train, Type K Brakes.* In all the empty trains, the relative car impact speeds before the passing point are independent of the draft-gear characteristics and are located on the straight line through the origin defined by Equation [25].

The location of the passing point on this line at car n_1 and the corresponding relative car impact speed v_1 are determined by Equations [28] and [29], respectively.

The relative impact speeds of the cars beyond the passing point are governed by the differential Equation [32].

Fig. 5 shows in heavy full lines the nv curves of the empty train with type K brakes for various values of the free plus residual slack s_1 up to its critical value, $s_2 = 6.90$ in., the latter to be computed from Equation [36].

The curves beyond the passing point were constructed by approximative integration from Equation [32], using a method similar to the Simpson rule.

(*b*) *Loaded Trains, Type K Brakes.* For these trains the propagation ratio is less than unity, as shown in Table 1. The relative car impact speeds v are therefore independent of the amount of free plus residual slack per coupling s_1 and may be computed from Equation [25]. They are shown in heavy dotted lines in Fig. 5.

Example 7. Computation of the propagation ratios for a train passing through a change in the grade of slack:

Let us assume that the test trains coast at a speed of 40 mph and are subjected to a change in grade of 4 per cent.

The gravitational forces will produce a change in force per car equal to 4 per cent of its weight, and therefore

$$F = 0.04 \times 50{,}000 = 2000 \text{ lb for empty cars and}$$
$$= 0.04 \times 210{,}000 = 8400 \text{ lb for loaded cars}$$

The time t_1 between force applications on adjacent cars at a train speed of 40 mph (= 58.6 fps) and for a car length of 50 ft is

$$t_1 = 50/58.6 = 0.852 \text{ sec}$$

Substituting these constants in Equation [13], the propagation ratios are found to be recorded as follows:

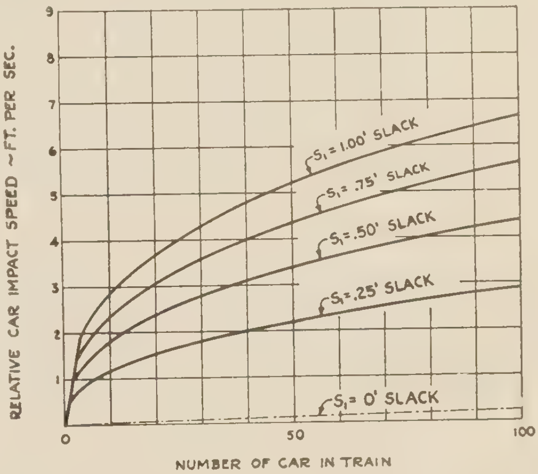

Fig. 6 Relative Car Impact Speeds in Empty Trains Running Through Dip in Track at 40 Mph, 4 Per Cent Change in Track Grade

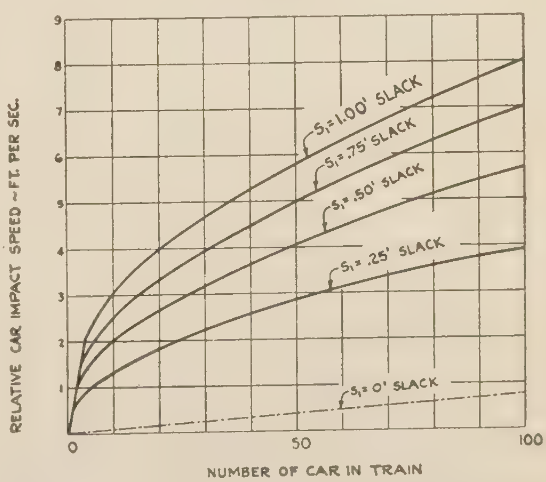

Fig. 7 Relative Car Impact Speeds in Loaded Trains Running Through Dip in Track at 40 Mph, 4 Per Cent Change in Track Grade

TABLE 2

In empty trains,	$p = 16.1$
In loaded trains,	$p = 7.83$

Example 8. Relative car impact speeds:

This covers the computation of the relative car impact speeds of all cars in the test trains when passing through a 4 per cent dip at a train speed of 40 mph for different values of the free plus residual slack per coupling. For all the cars before the passing point, these speeds are located on the straight line through the origin, defined by Equation [25]. The locations of the passing points on this line are computed from Equations [28] and [29] and are plotted in Fig. 6.

The nv curves for empty and loaded trains beyond the passing point were computed by approximate integration from Equation [33] and plotted in Fig. 6 for empty trains, and in Fig. 7 for loaded trains.

Energy Intake and Gear Reaction

The foregoing equations and examples refer mostly to gear reactions in trains without free or residual slack, and to relative car impact speeds in trains with free and residual slack. A few remarks regarding the relationship of car impact speeds to energy intake and gear reaction are therefore indicated.

If two cars, each of the mass M, collide at the relative impact speed v, their draft gears will be compressed until the car speeds are equalized. At that time one of the cars will have gained and the other one will have lost a speed, equal to $v/2$ and, disregarding the elasticity of the car bodies, the energy taken up by each of the two coacting gears is, therefore

$$(^1/_2)M(v/2)^2 = Mv^2/8$$

Likewise, if a single car strikes a group of cars and if the draft gears of the struck cars were fully expanded before the impact occurred, the same relative impact speed v will cause the same energy intake per gear as just given.

If, however, a car strikes a group of bunched cars with fully compressed draft gears, the relative kinetic energy of the moving car, amounting to $Mv^2/2$, will be taken up by the two impacting gears, each taking up the energy $Mv^2/4$.

In the latter case and if the draft gears have a capacity of C ft-lb and a total travel of q ft, the gear reaction P produced by the energy intake of $Mv^2/4$ would be the one given in Equation [2].

Instead of computing the gear reaction from this equation and since most draft gears do not have a straight compression line, it is recommended to use a curve, obtained by laboratory test, where the gear reaction, as established by drop test, is plotted against the energy intake of the gear.

Draft-Gear Recoil

The foregoing equations as well as the curves plotted from the computations in the various examples show, and practical experience confirms, that free and residual slack increases the relative car impact speeds, and causes undesirably high gear reactions in a train, during starting, braking, and passing over changes in the grade of the track.

The free slack is mainly composed of clearances in couplers and draft-gear attachments, and is therefore largely beyond the control of the draft-gear designer.

The residual slack, which is equally undesirable, is caused by sticking draft gears and by gears of insufficient recoil energy to expand a partly or fully compressed train or to contract a partly or fully stretched one.

It is therefore desirable to design draft gears with ample recoil energy so as to prevent sticking and reduce the residual slack.

It is also desirable to design them with high initial recoil force in order to reduce the residual slack, even after comparatively light train compressions.

Acknowledgment

The author is indebted to the Edgewater Steel Company for sponsoring this investigation; to Mr. L. H. Fry for deriving some of the more important equations by a different method, thus checking the validity of the elastic-bar method; and to Dr. W. M. Dudley for helpful suggestions.

Temperature Distribution Within Boiler Tubing Under Oblique Radiation

By W. S. KIMBALL,[1] ANNAPOLIS, MD.

The surface and interior temperature distribution in a section of boiler tubing under normal symmetric radiation from the furnace has been investigated by G. M. Dusinberre and others, and reported on in a discussion of data furnished by W. F. Davidson.[2] In the same issue such problems in heat conduction are also treated by Paschkis[3] using the method of electrical analogy, and by Emmons[4] using the numerical method of obtaining approximate solutions; and in the reported discussion of the latter Dusinberre points out that the numerical method, carried out to the needed accuracy, checks the analytical solution and results are obtained with greater ease. The analytical method of Fourier, previously confined to the symmetric case of normal radiation, is here extended to the unsymmetric case where radiation from the fire is received obliquely and with consequent shadowing effect from adjoining tubes. An advantage of this investigation is that it supplies a general mathematical expression for temperature distribution under parallel radiation received at any angle with the boiler wall normal, thus affording a check on results that sometimes may be obtained easier by other methods. A further advantage is that this general mathematical expression makes it fairly easy to compare temperature gradients and hence rates of the heat transfer as done herein for various obliquity angles of received radiation.

In the present study of temperature distribution in boiler tubing, it is assumed that interface resistance and external convection and conduction are negligible. These assumptions are in line with the conclusions of the authors mentioned, when conduction is ruled out by having insulated refractory backing at practically tube-surface temperature. Thus no heat transfer across the outer tube surface occurs except by radiation.

It is further assumed, as formerly, that a tube receives radiation uniformly per unit projected area as from a distant source. For symmetric radiation previously treated this leads to the temperature-slope condition at the outer tube surface

$$\frac{\partial T}{\partial r} = \frac{Q}{K} \cos \theta$$

where θ is the angle between the received radiation and the tube surface normal. For oblique radiation now treated, the foregoing condition is replaced by condition, Equation [3] with $\theta - \beta$ in place of θ, since β measures the obliquity of the radiation.

The general temperature expression here derived includes the special symmetric case where the obliquity angle is zero and the received radiation comes in normal to the boiler wall with formula given by Equation [13] of the Davidson paper[2] and graphed in his Fig. 48. In all cases herein parallel radiation is assumed,[5] making a given obliquity angle β with the boiler wall normal, and the wall is made up of cylindrical tubes in contact.

This new general temperature expression is here studied for a series of obliquity angles for the received radiation. With all of these, the surface exposed to radiation is reduced by shadowing of adjoining tubes below the 180 deg exposure, or semicircumference opposite the fire for normal radiation.

Special attention and graphs are given for four typical cases, (*a*) where 120 deg of tube surface is exposed to radiation of exactly three quarters of the amount of normal heat radiation, (*b*) where 90 deg of tube surface is exposed, receiving exactly one half the amount of normal radiation per tube, (*c*) where 60 deg of tube surface is exposed, receiving exactly one quarter of the normal radiation per tube, and (*d*) where 30 deg of tube surface is exposed, receiving one fifteenth of the normal radiation.

For the 120-deg exposure or two thirds the normal, the radiation comes in at an obliquity angle of 41°25′ with the boiler wall normal. For 90 deg or one half the normal exposure to radiation, the radiant heat comes in at a 60 deg angle of obliquity with the boiler wall normal, and for a 60-deg tube exposure or one third the normal, the radiation comes in at a 75°30′ angle with the boiler wall normal. When only 30 deg or one sixth of the tube surface is exposed, the radiation comes at an angle of 86 deg with the boiler wall normal, being only 4 deg short of the grazing angle of tangency with the wall, with the amount of received radiation one fifteenth that of the normal radiation per tube.

The temperature graphs show that the hot point of highest temperature is not where the oblique radiation comes in normal to the tube surface, but is shifted back toward the symmetric point opposite the fire cavity where the tube surface parallels the boiler wall, and where the hot point would be in case of radiation normal to the boiler wall. This hot-point shift occurs in the direction of the geometrical center of the received radiation, as is to be expected.

A most interesting result is the way the temperature gradient or difference between outer and inner tube-surface temperatures falls off with increased angle of obliquity, assuming the same density of radiation. Where the received radiation is three quarters the normal radiation, and comes at a 41-deg angle with 120 deg of tube surface exposed to it, there is only a 2 per cent drop in temperature gradient, the distribution being merely shifted around the tube in the direction of the received radiation.

When one half the normal radiation is received over 90 deg of exposed surface, coming in at 60 deg to boiler wall normal, there is a 14 per cent loss in the temperature gradient, and when only one quarter of normal radiation is received over one third as much tube surface exposed (60 deg), then the loss in temperature drop is 50 per cent, as compared with that for normal radiation.

[1] Lieutenant Commander, Department of Marine Engineering, U. S. Naval Academy.

[2] "Studies of Heat Transmission Through Boiler Tubing at Pressures From 500 to 3300 Pounds," by W. F. Davidson, P. H. Hardie, C. G. R. Humphreys, A. A. Markson, A. R. Mumford, and T. Ravese, Trans. A.S.M.E., vol. 65, 1943, pp. 553–579, and discussion, pp. 579–591.

[3] Ibid., p. 587, discussion by V. Paschkis.

[4] "The Numerical Solution of Heat-Conduction Problems," by H. W. Emmons, Trans. A.S.M.E., vol. 65, 1943, pp. 607–612.

Contributed by the Heat Transfer Division and presented at the Spring Meeting, Birmingham, Ala., April 3–5, 1944, of The American Society of Mechanical Engineers.

[5] See reference 2, Appendix, p. 581, Max Jakob.

And when about one fifteenth the normal radiation is received at only 4 deg short of grazing angle with 30 deg of exposed surface (one sixth of the normal), then the temperature drop is about 15 per cent of normal radiation, representing an 85 per cent reduction in gradient.

These results indicate that the ease or efficiency with which heat is disposed of decreases with increased obliquity angle, since the shortage of heat supplied from the fire falls off with increasing obliquity angle much faster than the corresponding temperature gradients.

A further point to be noted is that the sum of these temperature expressions as solutions of Laplace's equation is still such a solution and would represent a physical situation corresponding to a superposition of the conditions they separately represent, likewise, if different values of Q, the normal heat flux per unit area, were used for the various cases. Thus any condition equivalent to and separable into two or more cases of parallel radiation acting in unison could be represented by a temperature expression given by the sum of such series as here shown.

Nomenclature

The following nomenclature will be used in succeeding sections of the paper:

T = temperature
T_0 = temperature at inner tube radius r_0(= 0, for convenience)
r = radial distance from tube center
θ = angle measured from an axis normal to boiler wall and extending into fire
r_0 = inside radius of boiler tube
r_1 = outside radius of boiler tube
Q = maximum heat flux at r_1, i.e. the density of radiant heat
K = thermal conductivity of metal
β = angle between boiler wall normal and direction of oblique radiation
γ = shadow angle, measuring from boiler wall, the extent of tube surface that receives no radiation owing to shadow from the adjoining tube (see Fig. 1).
$\pi - \beta - \gamma = \pi - 2\alpha$ = angular range of received radiation on a tube
$\alpha = \dfrac{\beta + \gamma}{2}$ = auxiliary angle, measuring one half reduction in angular range from π, corresponding to normal radiation
$\phi = \theta - \dfrac{\beta - \gamma}{2} = \theta - (\beta - \alpha)$ = position angle measured from the mid-point, $\beta - \gamma/2$ of angular range exposed to oblique radiation (see Fig. 1)
$\rho = \dfrac{r}{r_0}$ = dimensionless ratio for radial distance r
$\rho_1 = \dfrac{r_1}{r_0}$ = dimensionless ratio at outer tube radius r_1 (= 1.5 throughout this paper)
A_m, B_m, A'_m, B'_m are Fourier constant coefficients evaluated later

Assumptions and Boundary Conditions

Three basic assumptions are made as follows

$$T_0 = \text{constant } (= 0) \quad \text{[1]}$$

at inner boundary where $r = r_0$ which implies negligible interface resistance at the inside of the tube in contact with the steam.

$$\left(\frac{\partial T}{\partial r}\right)_{r=r_1} = 0 \quad \text{[2]}$$

for
$$\left(\frac{\pi}{2} - \gamma\right) < \theta < \left(\frac{3\pi}{2} + \beta\right)$$

Condition Equation [2] says there is no heat transfer across the entire outer boundary where $r = r_1$ except where radiation is received between angles β and γ shown in Fig. 1. The back part is assumed to be insulated and the part of the circumference toward the fire that receives no radiation is assumed to be in equilibrium with the surroundings.

The third basic assumption gives the condition for absorption of radiant energy, as follows

$$\left(\frac{\partial T}{\partial r}\right)_{r=r_1} = \frac{Q}{K}\cos(\theta - \beta) \quad \text{[3]}$$

for
$$-\left(\frac{\pi}{2} - \beta\right) < \theta < \left(\frac{\pi}{2} - \gamma\right)$$

This condition refers to parallel radiation of uniform density received on the tube surface at angle $\theta - \beta$ with the tube surface

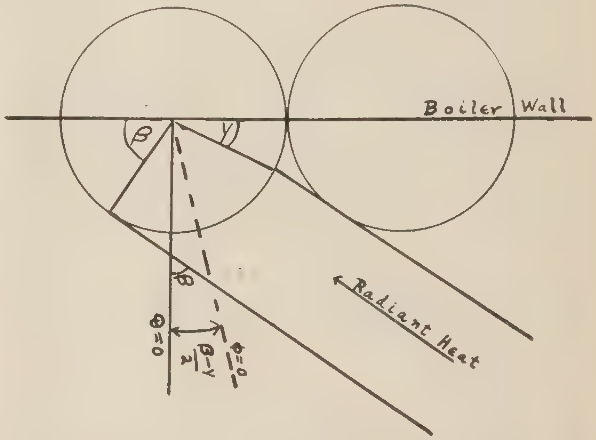

Fig. 1 Obliquity Angle β, Shadow Angle γ, and Mid-Point $\theta = \dfrac{\beta - \gamma}{2}$ of Surface Exposed to Oblique Radiation

normal. Unit cross section of radiation will be spread over an area equal to sec $(\theta - \beta)$ so that the fraction of Q, the radiation density that is conducted away through unit surface area, will be reduced by this same proportion according to Equation [3].

When conditions, Equations [2] and [3], are referred to the angle ϕ and constant angle α which are symmetric about the dotted center line $\phi = 0$ of exposed tube surface, shown in Fig. 1, Equations [2] and [3] take a symmetric form adapted to computation of the Fourier coefficients.

$$\left(\frac{\partial T}{\partial r}\right)_{r=r_1} = 0 \quad \text{[4]}$$

for
$$\left(\frac{\pi}{2} - \alpha\right) < \phi < \left(\frac{3\pi}{2} + \alpha\right)$$

$$\left(\frac{\partial T}{\partial r}\right)_{r=r_1} = \frac{Q}{K}\cos(\phi - \alpha) = \frac{Q}{K}(\cos\alpha\cos\phi + \sin\alpha\sin\phi) \quad \text{[5]}$$

for
$$-\left(\frac{\pi}{2} - \alpha\right) < \phi < \left(\frac{\pi}{2} - \alpha\right)$$

Relation Between Obliquity Angle β, and Shadow Angle γ

The relation between the obliquity of radiation β and the shadow angle γ is most easily obtained by applying the law of sines to the triangle made by extending the straight-line tangent to the right-hand circle in Fig. 1, until it hits the line of centers of the two circles. We then have

$$\cos(\beta + \gamma) = 2\cos\beta - 1 = \cos 2\alpha \quad \ldots\ldots [6]$$

where on the right the symmetric auxiliary angle α is introduced, defined according to the nomenclature. This relation is essential to obtaining the graphs which follow. Note $\cos^2\alpha = \cos\beta$.

Temperature Within a Boiler Tube Under Oblique and Normal Radiation

Expressions for the temperature within a boiler tube under radiation are obtained by solving Laplace's equation and computing the Fourier coefficients in the usual way. When the exposed surface includes the angular range $\pi - 2\alpha = \pi - \beta - \gamma$, according to the nomenclature, we have in terms of α and the ratios $\rho = \dfrac{r}{r_0}$ and $\rho_1 = \dfrac{r_1}{r_0}$ of the radii

$$
\left.
\begin{aligned}
T = \frac{Qr_1}{\pi K}\Bigg[& \cos^2\alpha \log\rho + \frac{\rho_1}{2\rho}\left(\frac{\rho^2 - 1}{\rho_1^2 + 1}\right)\Big\{(\pi - 2\alpha \\
& - \sin 2\alpha)\sin\alpha\sin\phi + (\pi - 2\alpha + \sin 2\alpha)\cos\alpha\cos\phi\Big\} \\
& + \frac{\rho_1^2}{6\rho^2}\left(\frac{\rho^4 - 1}{\rho_1^4 + 1}\right)\Big\{(3\cos\alpha + \cos 3\alpha)\sin\alpha\sin 2\phi \\
& + (3\cos\alpha - \cos 3\alpha)\cos\alpha\cos 2\phi\Big\} \\
& + \frac{\rho_1^3}{12\rho^3}\left(\frac{\rho^6 - 1}{\rho_1^6 + 1}\right)\Big\{(2\sin 2\alpha + \sin 4\alpha)\sin\alpha\sin 3\phi \\
& + (2\sin 2\alpha - \sin 4\alpha)\cos\alpha\cos 3\phi\Big\} \\
& - \frac{\rho_1^4}{60\rho^4}\left(\frac{\rho^8 - 1}{\rho_1^8 + 1}\right)\Big\{(5\cos 3\alpha + 3\cos 5\alpha)\sin\alpha\sin 4\phi \\
& + (5\cos 3\alpha - 3\cos 5\alpha)\cos\alpha\cos 4\phi\Big\} + \ldots\ldots
\end{aligned}
\right\} \ldots [7]
$$

The most important special case of Equation [7] occurs when the radiation is normal to the boiler wall and the angular range of exposed tube surface is 180°, with obliquity angle $\beta = 0$. We see from Equation [6] and Fig. 1, that in this case angles α and γ are also zero, hence $\sin\alpha = 0$ and $\cos\alpha = 1$, and likewise with the sines and cosines of all multiples of α. Hence Equation [7] reduces to

$$
\begin{aligned}
T = \frac{Qr_1}{\pi K}\Bigg[& \log\rho + \frac{\pi\rho_1}{2\rho}\left(\frac{\rho^2 - 1}{\rho_1^2 + 1}\right)\cos\theta + \frac{\rho_1^2}{3\rho^2}\left(\frac{\rho^4 - 1}{\rho_1^4 + 1}\right)\cos 2\theta \\
& - \frac{\rho_1^4}{30\rho^4}\left(\frac{\rho^8 - 1}{\rho_1^8 + 1}\right)\cos 4\theta + \frac{\rho_1^6}{105\rho^6}\left(\frac{\rho^{12} - 1}{\rho_1^{12} + 1}\right)\cos 6\theta + \ldots [8]
\end{aligned}
$$

This is the same temperature expression as given by Equation [13] in the appendix of the Davidson paper[2] and graphed there in Fig. 48. Equation [8] herewith is expressed in terms of the dimensionless variable $\rho = \dfrac{r}{r_0}$ and constant $\rho_1 = \dfrac{r_1}{r_0}$ instead of the r's, thus checking the previous result.

Temperature Distribution Under Radiation of 41°25′ Obliquity and 120° of Exposed Tube Surface With Three Quarters of the Heat Supply of Normal Radiation and 2 Per Cent Loss of Temperature Gradient

The angular range (see nomenclature) determines α

$$\pi - 2\alpha = \frac{2}{3}\pi, \text{ therefore } \alpha = \frac{\pi}{6} = 30 \text{ deg}$$

Hence from Equation [6]

$$\cos\beta = \frac{3}{4}; \quad \beta = 41°\,25'$$

$$\gamma = 18°35'$$

and the mid-point of the angular range of exposed radiation (nomenclature) is

$$\theta_0 = \beta - \alpha = 11°25'$$

where $\phi = 0$.

Since Q, the radiation density, is the same as for normal radiation, and since the entire beam of received radiation is for 120 deg of exposed surface, just $^3/_4$ the breadth of the normal radiation beam, we see that the total heat supplied is exactly $^3/_4$ of that supplied under normal radiation.

Using $\alpha = \dfrac{\pi}{6}$ in Equation [7] gives the formula for this distribution

$$
\left.
\begin{aligned}
(a)\quad T = \frac{Qr_1}{\pi K}\Bigg[& \frac{3}{4}\log\rho + \frac{\rho_1}{4\rho}\left(\frac{\rho^2 - 1}{\rho_1^2 + 1}\right)\Big\{\Big(\frac{2}{3}\pi \\
& - \frac{\sqrt{3}}{2}\Big)\sin\phi + \left(\frac{2}{3}\pi + \frac{\sqrt{3}}{2}\right)\sqrt{3}\cos\phi\Big\} \\
& + \frac{\rho_1^2}{8\rho^2}\left(\frac{\rho^4 - 1}{\rho_1^4 + 1}\right)\left(\sqrt{3}\sin 2\phi + 3\cos 2\phi\right) \\
& + \frac{\rho_1^3}{16\rho^3}\left(\frac{\rho^6 - 1}{\rho_1^6 + 1}\right)\left(\sqrt{3}\sin 3\phi + \cos 3\phi\right) \\
& + \frac{\rho_1^4}{80\rho^4}\left(\frac{\rho^8 - 1}{\rho_1^8 + 1}\right)\left(\sqrt{3}\sin 4\phi - 3\cos 4\phi\right) + \ldots\ldots
\end{aligned}
\right\} \ldots [9]
$$

Temperature Distribution Under Radiation of 60 Deg Obliquity and 90 Deg of Exposed Tube Surface With One Half the Heat Supply of Normal Radiation and 14 Per Cent Loss of Temperature Gradient

Angle α is determined by the exposed angular range

$$\pi - 2\alpha = \frac{\pi}{2}; \quad \alpha = \frac{\pi}{4} = 45°$$

Hence from Equation [6]

$$\cos\beta = \frac{1}{2}; \quad \beta = 60°$$

$$\gamma = 30°$$

and the mid-point of angular range exposed to radiation is

$$\theta_0 = \beta - \alpha = 15°$$

where $\phi = 0$.

With the same radiation density Q as before and one half the breadth of the beam, we have exactly one half the heat supply. We note that the hot point is now at only 86 per cent of the 100 deg taken as a comparison standard, being the temperature drop for the hot point under normal radiation. The hot point here

is no longer under the radiation that hits the tube normally at $\theta = 60$ deg because this is the outer edge of the radiation beam, but is shifted some 25 deg or 30 deg toward the center of the radiation beam.

The formula for this distribution is given by Equation [7] using $\alpha = \frac{\pi}{4}$

$$(b)\quad T = \frac{Qr_1}{\pi K}\left[\frac{1}{2}\log\rho + \frac{\rho_1\sqrt{2}}{4\rho}\left(\frac{\rho^2-1}{\rho_1^2+1}\right)\left\{\left(\frac{\pi}{2}-1\right)\sin\phi + \left(\frac{\pi}{2}+1\right)\cos\phi\right\} + \frac{\rho_1^2}{6\rho^2}\left(\frac{\rho^4-1}{\rho_1^4+1}\right)(\sin 2\phi + 2\cos 2\phi) + \frac{\sqrt{2}\rho_1^3}{12\rho^3}\left(\frac{\rho^6-1}{\rho_1^6+1}\right)(\sin 3\phi + \cos 3\phi) + \frac{\rho_1^4}{60\rho^4}\left(\frac{\rho^8-1}{\rho_1^8+1}\right)(4\sin 4\phi + \cos 4\phi) + \ldots\right. \quad ..[10]$$

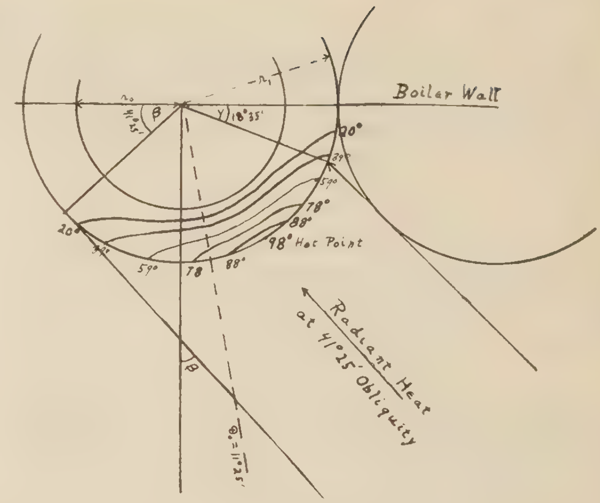

FIG. 2 RADIATION AT 41°25′ OBLIQUITY, OF THREE QUARTERS THE AMOUNT, AND RECEIVED BY TWO THIRDS THE SURFACE EXPOSED TO NORMAL RADIATION

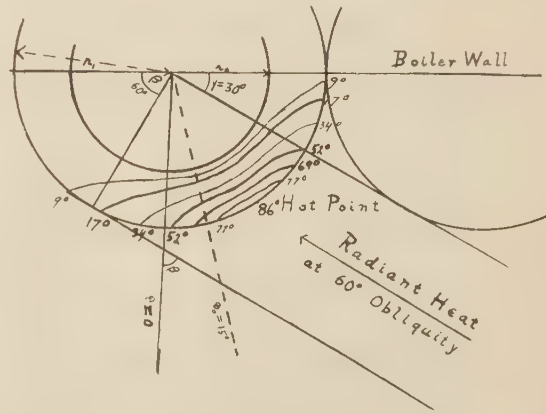

FIG. 3 RADIATION AT 60 DEG OBLIQUITY, OF ONE HALF THE AMOUNT, AND RECEIVED BY ONE HALF THE SURFACE EXPOSED TO NORMAL RADIATION

TEMPERATURE DISTRIBUTION UNDER RADIATION OF 75°30′ OBLIQUITY AND 60° OF EXPOSED TUBE SURFACE WITH ONE QUARTER OF THE HEAT SUPPLY OF NORMAL RADIATION AND 50 PER CENT LOSS OF TEMPERATURE GRADIENT

Angle α is determined by the exposed angular range

$$\pi - 2\alpha = \frac{\pi}{3} = 60° = \alpha$$

Hence from Equation [6]

$$\cos\beta = 1/4;\ \beta = 75°30'$$
$$\gamma = 44°30'$$

and the mid-point of the angular range exposed to radiation is

$$\theta_0 = \beta - \alpha = 15°30' \text{ where } \phi = 0$$

The temperature distribution of Fig. 4 is given by using $\alpha = \frac{\pi}{3}$ in Equation [7]

$$T = \frac{Qr_1}{\pi K}\left[\frac{1}{4}\log\rho + \frac{\rho_1}{4\rho}\left(\frac{\rho^2-1}{\rho_1^2+1}\right)\left\{\left(\frac{\pi}{\sqrt{3}}-\frac{3}{2}\right)\sin\phi + \left(\frac{\pi}{3}+\frac{\sqrt{3}}{2}\right)\cos\phi\right\} + \frac{\rho_1^2}{24\rho^2}\left(\frac{\rho^4-1}{\rho_1^4+1}\right)(\sqrt{3}\sin 2\phi + 5\cos 2\phi) + \frac{\rho_1^3}{16\rho^3}\left(\frac{\rho^6-1}{\rho_1^6+1}\right)(\sin 3\phi + \sqrt{3}\cos 3\phi) + \frac{\rho_1^4}{240\rho^4}\left(\frac{\rho^8-1}{\rho_1^8+1}\right)(7\sqrt{3}\sin 4\phi + 13\cos 4\phi) + \ldots\right. \quad ..[11]$$

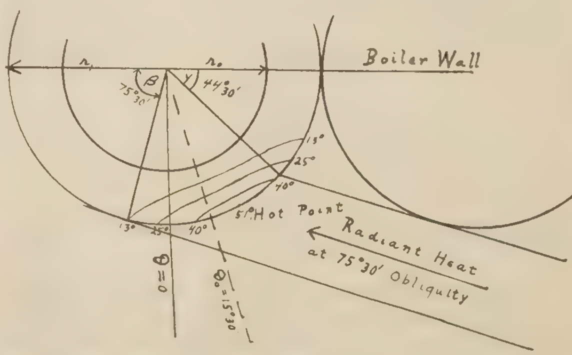

FIG. 4 RADIATION AT 75°30′ OBLIQUITY, OF ONE QUARTER THE AMOUNT, AND RECEIVED BY ONE THIRD THE SURFACE EXPOSED TO NORMAL RADIATION

TEMPERATURE DISTRIBUTION UNDER RADIATION OF 86°10′ OBLIQUITY, RECEIVED OVER 30° OF EXPOSED TUBE SURFACE ABSORBING ONLY ONE FIFTEENTH OF THE NORMAL RADIATION, AND WITH TEMPERATURE GRADIENT 15 PER CENT OF THE GRADIENT UNDER NORMAL RADIATION

The angular range determines angle α

$$\pi - 2\alpha = \frac{\pi}{6};\ \alpha = \frac{5}{12}\pi = 75°$$

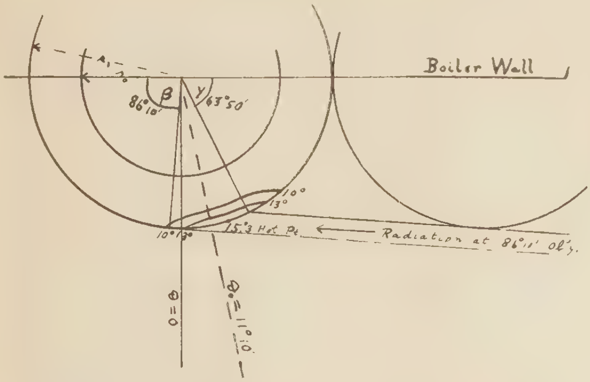

Fig. 5 Radiation at 86°10′ Obliquity, of One Fifteenth the Amount, and Received by One Sixth the Surface Exposed to Normal Radiation

Hence from Equation [6]

$$\cos\beta = 0.067;\ \beta = 86°10' \qquad \gamma = 63°50'$$

and the center line of the exposed surface is at

$$\theta_0 = \frac{\beta - \gamma}{2} = 11°10'$$

where $\phi = 0$.

The temperature expression for the foregoing distribution comes from introducing $\alpha = 75$ deg in Equation [7]

$$T = \frac{Qr_1}{\pi K}\Bigg[0.067 \log\rho + \frac{1}{4}\frac{\rho_1}{\rho}\left(\frac{\rho^2 - 1}{\rho_1^2 + 1}\right)\left\{\left(\frac{\pi}{3} - 1\right)0.9659 \sin\phi + \left(\frac{\pi}{3} + 1\right)0.2588 \cos\phi\right\}$$
$$+ \frac{\rho_1^2}{6\rho^2}\left(\frac{\rho^4 - 1}{\rho_1^4 + 1}\right)(0.0657 \sin 2\phi + 0.3294 \cos 4\phi)$$
$$+ \frac{\rho_1^3}{12\rho^3}\left(\frac{\rho^6 - 1}{\rho_1^6 + 1}\right)(0.129 \sin 3\phi + 0.4829 \cos 3\phi)$$
$$+ \frac{\rho_1^4}{60\rho^4}\left(\frac{\rho^8 - 1}{\rho_1^8 + 1}\right)(0.616 \sin 4\phi + 1.65 \cos 4\phi) + \ldots\Bigg] \quad \ldots[12]$$

Heat Transfer per Degree of Temperature Drop as a Function of Obliquity Angle

The radiant heat transfer per unit tube length is evidently $H_0 = 2r_1Q$ for the normal radiation case where $2r_1$ is the breadth of the beam and Q is its density or heat flux per unit cross section. In case heat comes obliquely, the total amount received per tube is seen to be $H = H_0 \cos\beta$ since the breadth of the beam received by a tube is now $2r_1 \cos\beta$. Note that when only one fourth the normal radiation is received (Fig. 4) over one third the normal receiving area, the temperature drop is only one half that for normal radiation, showing that the heat disposed of per degree of temperature drop between outer and inner tube is one half as great for 75 deg obliquity of radiation as for normal radiation, due possibly to less cramped (from adjoining beams) means of heat disposal. This result, for various obliquity angles, may be graphed most conveniently by plotting H/T against $\cos\beta$. Thus, in Fig. 6 the ordinate is heat transfer per tube per temperature drop between outer and inner tube since $\Delta T = T - T_0 = T$ taking $T_0 = 0$ according to the nomenclature.

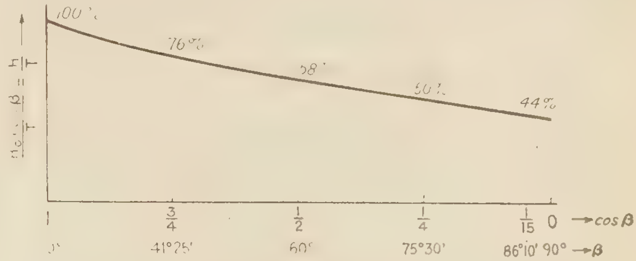

Fig. 6 Decrease in Heat Transfer per Degree of Temperature Gradient as Obliquity of Radiation Increases

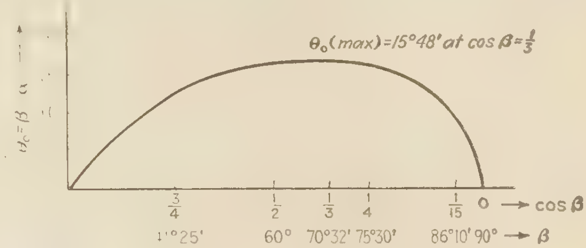

Fig. 7 Shift θ_0 of Center of Received Radiation as a Function of Obliquity and of Cos β

These are shown in percentages of heat transfer per degree, relative to the ratio for normal radiation taken as 100 per cent. They are not efficiencies in the true sense of the term, since there are no heat losses under any of the situations here treated (true efficiencies are all 100 per cent) but are perhaps due to greater spreading of heat through areas not exposed under oblique radiation.

Summary of Results

The distribution of temperature inside boiler tubes in contact under parallel oblique radiation of any obliquity angle β is given by Equation [7], assuming no heat transfer at back or anywhere across outer tube surface, except where it is under radiation from the fire.

The hot point on the boiler tube is near the center of the received radiation beam but shifted toward the surface point where the radiation strikes or would strike (except for shadowing) the surface normally for that obliquity angle.

The temperature drop between outer and inner tube surface is greater for a given quantity of received oblique radiation than for normal radiation, up to angles of extreme obliquity approaching 90 deg. This may be due to greater concentration of heat under oblique radiation, corresponding to greater ease in disposing of it under normal radiation. Fig. 6 shows how the heat transfer per degree of temperature drop decreases under oblique radiation continuously.

The center of the tube-surface area exposed to radiation given by $\theta = \beta - \alpha$ does not shift from its normal position $\theta = 0$ opposite the fire in the same way as the obliquity angle does, due to shadowing effect of adjoining tubes. Its maximum (see Appendix 2) occurs at radiation obliquity of $\beta = 70°32'$, where the central angle θ_0 of surface exposed to radiation is $15°48'$ representing maximum eccentricity of received radiation. When the eccentricity angle θ_0 is plotted against the cosine of obliquity angle β for received radiation, we have a graph of Equation [6] which is shown in Fig. 7

Appendix 1

Derivation of Temperature Distribution Under Oblique Radiation

Taking account of the first assumption or boundary condition, Equation [1], we have for the solutions of Laplace's equation

$$T = (r^m - r_0^{2m} r^{-m})(A_m \sin m\theta + B_m \cos m\theta);\ m \neq 0 \ldots\ldots [13]$$

$$T = c \log \frac{r}{r_0};\ m = 0$$

which includes terms in sin $m\theta$ as well as in cos $m\theta$ because of the unsymmetric type of distribution. We now introduce variable angle ϕ, measured from the center of the area exposed to radiation, and related to θ according to the nomenclature. Then, using new Fourier coefficients A' and B' in Equation [13], we have for our temperature expression

$$T = c \log \frac{r}{r_0} + \sum_{m=1}^{\infty} (r^m - r_0^{2m} r^{-m})(A'_m \sin m\phi + B'_m \cos m\phi) \quad [14]$$

where, according to the nomenclature, the constants are related by

$$\left.\begin{aligned} A_m &= A'_m \cos m(\alpha - \beta) - B'_m \sin m(\alpha - \beta) \\ B_m &= A'_m \sin m(\alpha - \beta) + B'_m \cos m(\alpha - \beta) \end{aligned}\right\} \ldots\ldots [15]$$

The Fourier coefficients of Equation [14] are computed in the usual way taking account of Equations [4] and [5]

$\frac{\partial T}{\partial r}$ at $r = r_1$ is, for the mth term of the series Equation [14]:

(for $m \neq 0$)

$$\left(\frac{\partial T}{\partial r}\right)_{r=r_1} = \frac{m}{r_1}\left(\frac{r_0^{2m} + r_1^{2m}}{r_1^m}\right)(A'_m \sin m\phi + B'_m \cos m\phi)$$

$$= \frac{Q}{K}(\sin\alpha \sin\phi + \cos\alpha \cos\phi)$$

The odd series coefficients are obtained by multiplying the foregoing odd functions by sin $m\phi$ and integrating, and the even terms by multiplying the even functions by cos $m\phi$ and integrating (between limits 0 to π on left and from 0 to $\frac{\pi}{2} - \alpha$ on the right). Thus

$$\frac{m}{r_1}\frac{(r_0^{2m} + r_1^{2m})}{r_1^m} A'_m \frac{\pi}{2} = \frac{Q}{K} \sin\alpha \int_0^{\frac{\pi}{2}-\alpha} \sin\phi \sin m\phi\, d\phi$$

$$= \frac{Q}{2K} \sin\alpha \int_0^{\frac{\pi}{2}-\alpha} [\cos(m-1)\phi - \cos(m+1)\phi]\, d\phi$$

$$\frac{m}{r_1}\left(\frac{r_0^{2m} + r_1^{2m}}{r_1^m}\right) B'_m \frac{\pi}{2} = \frac{Q}{K} \cos\alpha \int_0^{\frac{\pi}{2}-\alpha} \cos\phi \cos m\phi\, d\phi$$

$$= \frac{Q}{2K} \cos\alpha \int_0^{\frac{\pi}{2}-\alpha} [\cos(m-1)\phi + \cos(m+1)\phi]\, d\phi$$

where the integral on the right vanishes according to Equation [4] beyond the indicated upper limit of integration. When the values of A'_m and B'_m so determined are substituted in Equation [14] we have

$$\left.\begin{aligned} T = \frac{Qr_1}{\pi K}\Bigg[& \cos^2\alpha \log\frac{r}{r_0} + \frac{r_1}{2}\frac{(r - r_0^2 r^{-1})}{(r_1^2 + r_0^2)}\Big\{(\pi - 2\alpha - \sin 2\alpha)\sin\alpha\sin\phi + (\pi - 2\alpha + \sin 2\alpha)\cos\alpha\cos\phi\Big\} \\ & + \frac{r_1^2}{6}\left(\frac{r^2 - r_0^4 r^{-2}}{r_0^4 + r_1^4}\right)\Big\{(3\cos\alpha + \cos 3\alpha)\sin\alpha\sin 2\phi + (3\cos\alpha - \cos 3\alpha)\cos\alpha\cos 2\phi\Big\} \\ & + \frac{r_1^3}{12}\left(\frac{r^3 - r_0^6 r^{-3}}{r_0^6 + r_1^6}\right)\Big\{(2\sin 2\alpha + \sin 4\alpha)\sin\alpha\sin 3\phi + (2\sin 2\alpha - \sin 4\alpha)\cos\alpha\cos 3\phi\Big\} \\ & - \frac{r_1^4}{60}\left(\frac{r^4 - r_0^8 r^{-4}}{r_0^8 + r_1^8}\right)\Big\{(5\cos 3\alpha + 3\cos 5\alpha)\sin\alpha\sin 4\phi + (5\cos 3\alpha - 3\cos 5\alpha)\cos\alpha\cos 4\phi\Big\} + \ldots\ldots \end{aligned}\right\} \ldots [16]$$

When the dimensionless variable and constant $\rho = \frac{r}{r_0}$ and $\rho_1 = \frac{r_1}{r_0}$ are used in Equation [16], we obtain series Equation [7].

Appendix 2

Center of Area Exposed to Radiation as a Function of Obliquity Angle

If we solve Equation [6] for α, we have

$$\alpha = {}^1/_2 \text{ arc cos } (2\cos\beta - 1) \ldots\ldots\ldots\ldots [17]$$

and hence for the center of the area exposed to radiation according to Fig. 1 and the nomenclature

$$\theta_0 = \beta - \alpha = \beta - {}^1/_2 \text{ arc cos } (2\cos\beta - 1) \ldots\ldots [18]$$

The derivative of this is readily seen to be

$$\frac{d\theta_0}{d\beta} = 1 - \frac{\sin\beta}{2\sqrt{\cos\beta - \cos^2\beta}} \ldots\ldots\ldots\ldots [19]$$

Setting this equal to zero gives

$$\left.\begin{aligned} \cos\beta &= {}^1/_3 \\ \beta &= 70°32' \end{aligned}\right\} \ldots\ldots\ldots\ldots\ldots\ldots [20]$$

for the obliquity angle of maximum θ_0 being the center of the area exposed to radiation. The position of this maximum shift of the center is obtained by substituting Equation [20] in Equation [18].

$$\theta_{0(\max)} = \beta - {}^1/_2 \text{ arc cos } (-{}^1/_3) = \beta - {}^1/_2\, (109°28')$$

$$\theta_{0(\max)} = 70°32' - 54°44' = 15°48' \ldots\ldots\ldots\ldots [21]$$

This is the most eccentric position of the surface exposed to oblique radiation. Note that it is less than 1 deg out from the center of the radiation beam of 60 deg obliquity ($\theta_0 = 15°$) and of 75°30′ obliquity ($\theta_0 = 15°30'$) of Figs. 3 and 4, and greater than 70°32′ angles of obliquity reduce the eccentricity of the exposed surface. Thus Fig. 5 shows that for an 86°10′ obliquity angle the center of the area exposed to radiation is given by $\theta_0 = 11°10'$, being less eccentric than the distribution of Fig. 2, for 41 deg obliquity of radiation when the eccentric angle $\theta_0 = 11°25'$ gives the shift of the center from its position $\theta_0 = 0$ opposite the fire in case of normal radiation.

Discussion

G. M. Dusinberre.[6] The section, "Heat Transfer per Degree of Temperature Drop as a Function of Obliquity Angle β," opens an interesting line of study, for it shows that the maximum temperature gradient in the metal falls off surprisingly slowly as the heat transfer decreases (owing to location of the tube), under the author's assumptions. As the heat transfer and internal turbulence decrease, the temperature gradient between metal and boiling liquid will no longer be negligible. But according to the studies of Jakob[7] this gradient may well be constant about the inner circumference. If so, the author's temperature distributions would remain qualitatively correct. Now, if the interior surface gradient should become greater than the decrease of gradient in the metal, then a tube at the end of the wall, although receiving less heat, might attain a higher surface temperature than a tube opposite the flame. This high temperature would, moreover, be quite localized.

The practicing engineer who has not the author's thorough command of differential equations will find that solutions of this type of problem are readily obtained by the numerical method outlined by Emmons.[4] The precision of the method depends only upon the time spent, and this is equally true of the analytical solution, for one computes only as many terms of the series as is judged to be necessary for the purpose at hand. For a degree of precision consistent with that of most engineering data, the numerical method offers a tremendous saving of time.

The writer's experience shows that a particular case, such as is represented by one of the author's diagrams, can be solved from the beginning in the time required for the final computations alone in the analytical solution. Thus one saves the time required for solving the differential equation and for calculating the numerical constants in the series. The numerical method is also more flexible. As Emmons has pointed out, there is only slightly more difficulty in the treatment of a fin or stud tube, whereas these would greatly complicate an analytical solution. The same may be said for the effect of a slag layer such as occurs in pulverized-coal firing.

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[7] Reference 5 of paper, p. 585, discussion by Max Jakob.

Author's Closure

The main advantage of the present treatment over the numerical method discussed by Mr. Dusinberre seems to lie in its dual nature in that it supplies first the mathematical formula and secondly the graph for any oblique distribution within the scope of the subject matter, such as the four special series herein and their temperature distribution graphs. Its application involves, of course, the selection of a given obliquity angle β and its incorporation into Equation [7] and corresponding graph as preliminary to any special detailed study of the situation it represents; but now that [7] is supplied, the time involved may be of the same order of magnitude as for obtaining from the beginning a graph by the numerical method. And the present method is then roughly of twice the productivity in that it supplies the mathematical series as well as the graph.

One does not need to enlarge upon the power and completeness of any mathematical solution that is true to the facts of any problem. It usually involves hidden truths to be unraveled by further study of the formulas, as well as often unexpected though mathematically obvious relations such as that herein shown and discussed concerning the temperature drop as a function of the obliquity angle.

The greater flexibility of the numerical method is readily conceded, but flexibility of method is usually accompanied by corresponding handicaps. A more flexible analytical method is usually the more difficult one, witness Weierstrass elliptic functions as compared with the more frequently used Legendre functions. In the present instance the flexibility of the numerical method is obtained at the expense of any mathematical representation at all, and so in this regard it has only half the thoroughgoing completeness of the twofold treatment herein.

Ratio and Multiple-Fuel Controls in the Steel Industry

By HERBERT ZIEBOLZ,[1] CHICAGO, ILL.

The need for accurate control of the furnace atmosphere in the steel industry has brought about the development of a variety of designs and basic circuits to solve this problem. The methods used and the apparatus which were developed are briefly discussed, i.e., (1) the basic ratio control, (2) the control mixing stations, and (3) the controls available for burning several fuels simultaneously with only one common supply for combustion air.

Reason for Use of Ratio Controls in Steel Industry

The development of automatic controls and instruments in the steel industry followed a course somewhat different from that in other industries. Relatively great forces had to be handled. Many of the plants were old and out-dated, and the original designers had rarely considered the requirements of controls and instruments. Consequently, most of the installations had to be designed to meet individual requirements.

The last ten years have fortunately brought about a better understanding of the mutual problems, and the importance of this co-operation can hardly be overemphasized.

Generally speaking, there are three major problems, typical of the steel industry:

1 Chemically changing the material handled.
2 Changing its physical shape.
3 Raising its temperature prior to (1) and (2).

The diagram, Fig. 1, showing the variables entering into problem 3, i.e., the heating of carbon steel, gives a general view of the subject covered by this paper. The vertical Z axis shows the products of combustion of the most expensive fuel in the entire steel plant, i.e., scale. The X axis gives the percentages of excess and deficiency of combustion air, the Y axis gives various steel temperatures. It will also be noticed that the heating time, during which the steel is exposed to these temperatures and atmospheres, is indicated. This diagram gives in concise form the story of heating steel, and it will be observed that if it is required to bring a piece of steel to a given temperature with a minimum and controlled type of scale formation, two variables must be taken into account, as follows:

(a) Rate of temperature increase.
(b) Atmosphere control.

The purpose of this paper is to discuss the latter problem and, in particular, the mechanisms developed for combustion or atmosphere control. This includes the premixing of fuels to be used in the burners.

An attempt will be made to show the reasoning behind various solutions from the viewpoint of the oft forgotten designer, as a more general knowledge of his problems may well be of value in the future development of the art.

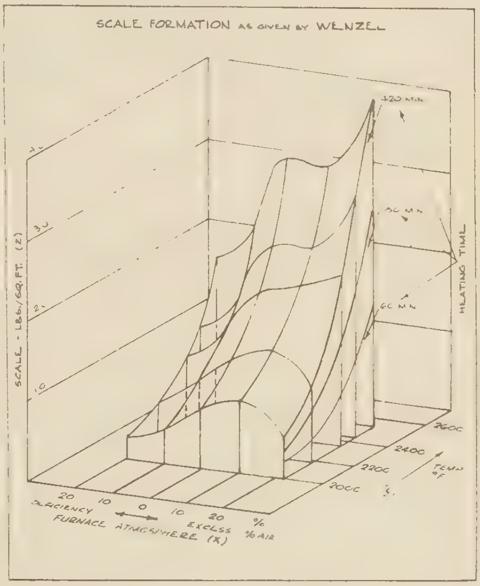

Fig. 1 Scale Formation of Carbon Steel as Function of Atmosphere, Temperature, and Heating Time for Carbon Content of 0.35 to 0.61 Per Cent

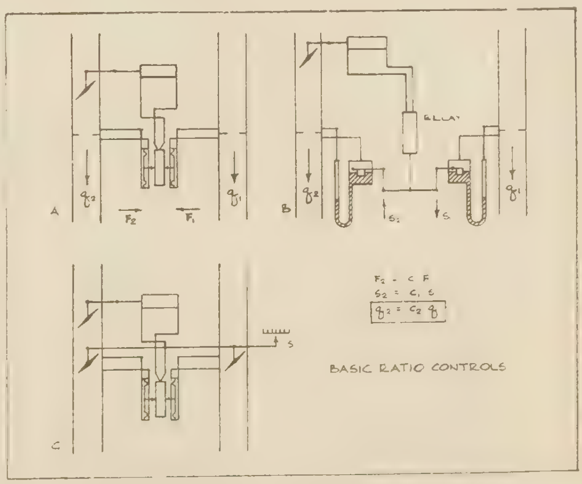

Fig. 2 Basic Ratio Controls

(A, Balancing of forces produced by pressures across fixed orifice plates. B, Balancing of movements created by pressures across fixed orifice plates. C, Balancing of forces produced by pressures across orifice plates.)

Basic Ratio-Control Principles

In order to control the proportion between fuel and air, and thus atmospheres, we must first measure the rates of flow. Two alternate methods are generally available:

1 The use of a fixed orifice with varying differential.
2 The use of a variable orifice with a fixed differential.

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[1] Vice-President, Askania Regulator Company. Jun. A.S.M.E.

Contributed by the Industrial Instruments and Regulators Division and presented at the Spring Meeting, Birmingham, Ala., April 3–5, 1944, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

It should also be borne in mind that most ratio- or mixing-control jobs demand a constant or varying ratio on a weight basis rather than on a volumetric basis. Thus the factor of density enters into the picture, and we can expect that in the future more consideration will be given to its measurement. Keeping this in mind, we can now consider basic solutions of ratio-control problems.

Fig. 2(*A*) shows the typical method of balancing two differentials against each other. As long as the two forces F_2 and F_1 are proportional the corresponding pressure drops and thus the rate of flows q_2 and q_1 are proportional. We shall use this diagram as a symbol rather than as representative of a particular design in the following schematic sketches.

As an alternative, we can match the strokes S_1 and S_2, Fig. 2(*B*), instead of forces, if we make provision that these movements are proportional to the differential pressures. The diagram is self-explanatory.

In Fig. 2(*C*) we find the same ratio control as in Fig. 2(*A*), with the modification that variable orifices are used to increase the range of operation. In this particular instance, butterflies of identical characteristics (with geometric proportionality and with the same Reynolds numbers) are used.

Central Gas-Mixing Stations for Interchangeable Fuels

Fortunately for the steel industry, it frequently has a number of sources of fuel gas at its disposal. To use these fuels without the need for repeated burner- and combustion-control adjustments, it is necessary to provide central or individual gas-mixing stations, in order to provide an interchangeable fuel at all times.

Owing to the wide variations in fuel demand in central mixing stations, it is obvious that only variable orifices can be used in such stations. Fig. 3(*A*) shows a "wide-range mixing station in which the variable orifices are operated by means of a differential controller which keeps constant the pressure drop across one of the variable orifices, q_1. The ratio-controller's job is to proportion the pressure drop produced by the other rate of fluid flow q_2, to that produced by q_1 across the primary variable orifice. Both orifices move simultaneously and are so connected as to give (directly) proportional free areas.

A supervising control, responding to the Btu of the mixture, density, or any other representative variable, is used to correct the ratio by adjusting *s*. Such additional controls serve at the same time as a correction for discrepancies between the flow characteristics of the variable orifices.

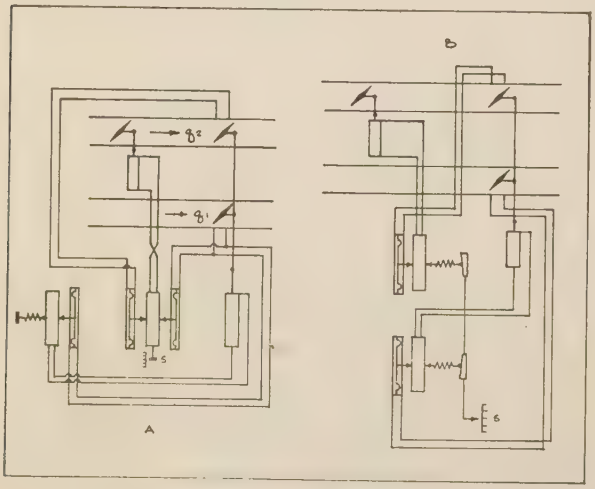

Fig. 3 Wide-Range Ratio Control
(*A*, With one differential and one ratio control. *B*, With two differential controls.)

From the foregoing, it seems logical to develop an alternative scheme, Fig. 3(*B*). If the differential across one of the orifices is kept constant, it is unnecessary to measure it a second time. A spring can thus be substituted for the right-hand diaphragm of Fig. 3(*A*), or to make the control more flexible, a cam can be used which simultaneously changes the differential-pressure settings of the primary- and secondary-flow-rate controllers. An adjustment of the cam *s* will change the ratio between the two flow rates, and thus obtain any desired relationship between *s* and the ratio between the two fluids.

Fig. 4 shows how the change of ratio setting is accomplished.

Typical Means for Varying Ratio Setting

The ratio adjuster as shown in Fig. 4(*a*) and in two typical controller designs has a particular characteristic, i.e., the relationship between its stroke and the ratio established for the opposing forces F_1 and F_2.

This ratio is given in Fig. 4(*b*) and represents a hyperbolic function. Fortunately, the hyperbola coincides to an unexpectedly high degree of accuracy with a logarithmic curve within the range of values of R between 2 and 0.5 in., which is typical for ratio adjustments. This makes an adjustment of the ratio slider possible without a recalibration of scales, as any correction factor can be introduced into the mechanism by means of a shift of the ratio scale. The reason for this is the fact that the log of the product of two variables is the sum of the logs of the two variables. It also allows the mechanical addition of movements representing correction factors, such as barometric pressure, temperature, Btu content, humidity, density, and other variables.

This ratio variation, produced by increasing one of the differentials of a ratio control relative to the magnitude of the other is shown in Fig. 4(*d*), where one orifice is fixed, the other variable, and in Fig. 4(*e*), where the free areas of both orifices change in opposite direction.

Another scheme which may at first glance appear rather elaborate is shown in Fig. 4(*f*). A ratio control establishes a pilot flow of low-pressure air which is directly proportional to the main flow. A double orifice of the type shown in Fig. 4(*e*) is used to split this pilot flow into two proportional branches. The flow through one of these branches is used to produce a signal which controls the rate of flow q_2, by means of a second ratio control. Thus depending upon the position of the double orifice in the pilot line, various ratios of q_1/q_2 can be obtained.

Compensation for Air Infiltration or Constant Amount of Primary-Air Supply

So far, we have only considered the control of one rate of flow proportionally to a second one and have solved our multiple-fuel problem by proportioning the individual fuels to each other, so as to produce a new composite fuel which can then be proportioned to air.

The simplest solution of a multiple-fuel problem is to provide a separate air supply for each fuel. Although this design is quite frequently used, it requires surplus capacity of fans and ducts, as, if 100 per cent of any fuel should be used at any time, the total air must be delivered from its particular source.

In this connection it may be well to recommend thinking of air as a fuel of about 100 Btu per standard cu ft which can be burned with various fluids.

Given two fluids with flows q_a and q_b, which are measured by their respective pressure drops h_a and h_b, with the flow constant c_a and c_b, the problem of controlling a third rate of flow q_c with h_c and c_c proportional to their sum takes the following form

$$(q_a + q_b) = q_c$$
$$(c_a \sqrt{h_a} + c_b \sqrt{h_b}) = c_c \sqrt{h_c}$$

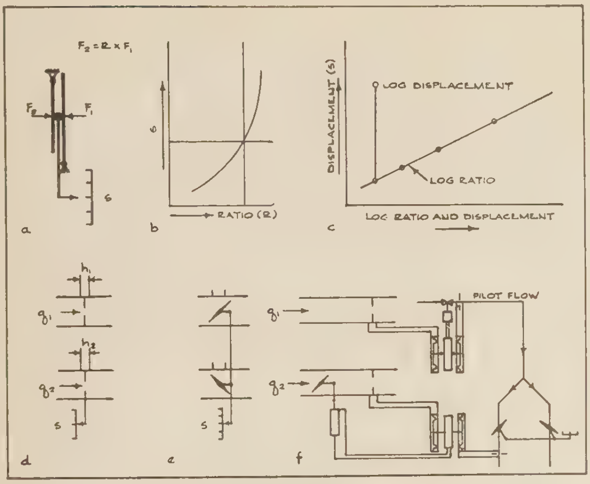

Fig. 4 Ratio-Varying Devices

(*a*, Ratio slider; *b* and *c*, relationship between stroke and ratio; *d* and *e*, ratio change by means of variable orifice; *f*, ratio change by means of controlling pilot flows.)

$$(c_a^2 \cdot h_a + c_b^2 \cdot h_b) + 2 \cdot c_a \cdot c_b \cdot \sqrt{h_a \cdot h_b} = c_c^2 \cdot h_c$$

This equation shows that h_c cannot be made proportional to $c_a^2 \cdot h_a + c_b^2 \cdot h_b$, as is time and again suggested even by engineers who should know better.

One typical example is the case where the fuel has to be proportional to the sum of two air flows. These two air supplies may be as follows:

(*a*) Combustion air + air infiltration; or
(*b*) Combustion air + primary air.

It is in such cases often erroneously suggested to provide an additional spring loading on one of the diaphragms to compensate for or to represent the rate of air infiltration or primary-air flow which is assumed to be constant.

In Fig. 5, the result of this method is shown giving the possible compensation for constant "air infiltration" of 5 per cent and 15 per cent maximum flow q_2.

The diagram, Fig. 6, proves that it is possible to improve greatly the effect of this compensation by using 3 times the equivalent force of the amount of infiltration as the spring tension to be

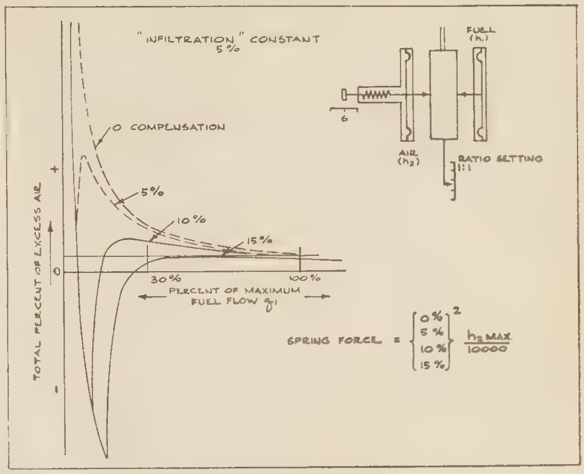

Fig. 6 Effect of Variables or Constant Forces Added to Ratio Regulators for Compensation of Infiltration

added. This gives satisfactory results down to as low as 30 per cent of the maximum rate of flow.

However, since at 10 per cent flow, the force of the diaphragm is equal to $^1/_{100}$ of the maximum force obtained from full flow, it becomes apparent how critical is this adjustment and that it should be used only in emergencies, and by men who know exactly its effect. A variation of infiltration with time, and an attempt to compensate for it by the foregoing method seems therefore not to be too desirable as a plant-maintenance job.

It seems promising to try a solution in which a model of the actual flow distribution is established and the same flow pattern is repeated on a smaller scale. Fig. 7 shows this method. A pilot flow is established, which is directly proportional to the total air requirements of the fuel q_1. Downstream from the pilot-flow valve, an orifice produces a signal differential which is used to act upon the main air-flow regulator. The orifice for the first ratio control, which establishes the pilot flow, is upstream from the pilot-flow valve. By bleeding various amounts of air between this valve and the upstream orifice, an amount of air can be deducted from the downstream pilot flow which is directly proportional to the air infiltration.

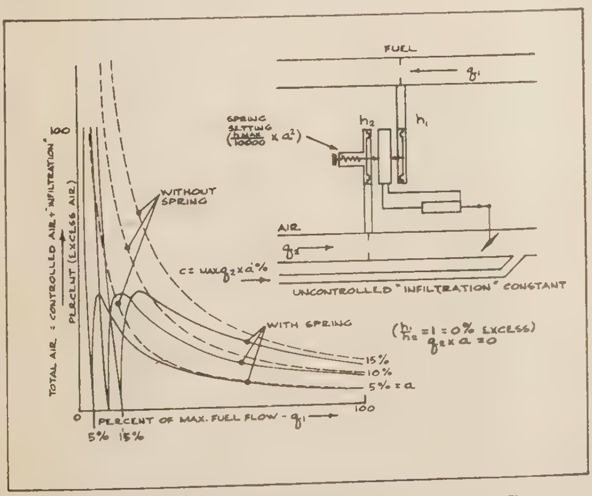

Fig. 5 Correction for Infiltration by Adding Constant Forces on Ratio Regulators

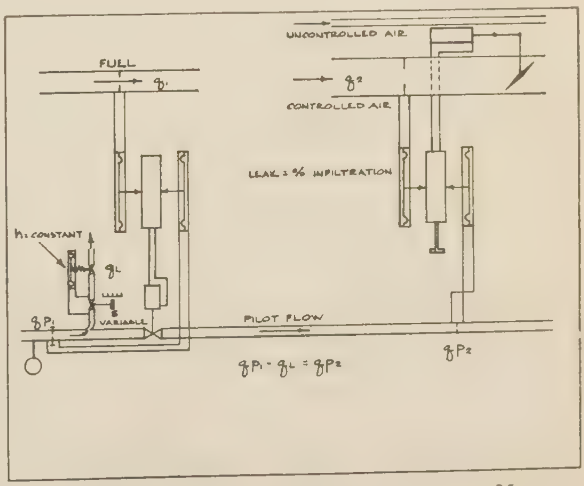

Fig. 7 Correction for Uncontrolled Air Flow by Means of Pilot Flows

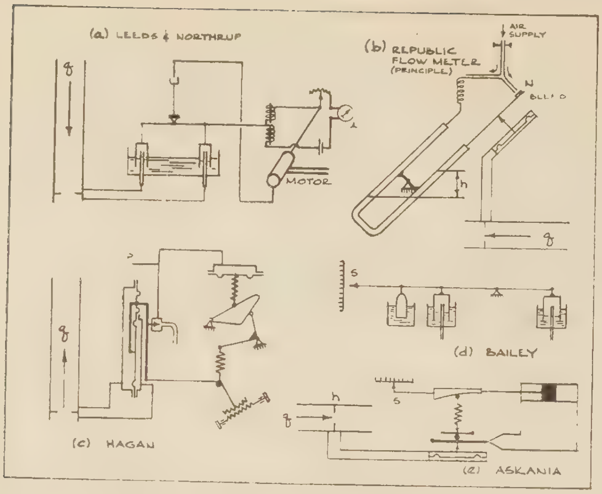

Fig. 8 Various Methods to Extract Square Roots and for Getting Second-Power Functions

A differential regulator is shown making the adjustment of the bleed-off valve a direct function of the bleed-off flow.

The basic reason for the success of this arrangement is the fact that the square functions, i.e., the differential pressures, are converted into linear functions of flow rates. These functions can then be added or subtracted without introducing an error.

Methods and Devices for "Modulating" Signals

It has been shown that the basic problem in multiple-fuel control is the conversion of square functions into linear functions, and the provision of means for adding and subtracting values representing individual flow rates or air requirements.

It seems desirable therefore to study some of the methods developed by various designers to solve this problem before we discuss the complete controls available for multiple-fuel-control problems.

In Fig. 8(*a*), one solution is found in the use of a Kelvin balance (Leeds & Northrup Company) which produces a "pilot" flow of electrical current which is directly proportional to the fuel flow or its air requirements.

As the force developed by the two coils in series is proportional to the square of the current flowing through them, and the force exerted on the pressure bells is proportional to q^2, we obtain a current which is directly porportional to the flow rate q. The current is controlled by a rheostat and a motor which reverses its direction of rotation, depending upon the direction of the unbalanced movement of the instrument. While previously an energy rate of flow was obtained in the form of a fluid pilot flow, the device mentioned produces a directly proportional electric-energy flow rate represented by the current i. This current, as we shall see later, lends itself just as well as the fluid pilot flow to summarizing and subtracting problems.

In Fig. 8(*b*), a mechanical device is shown in which the differential pressure across an orifice (a second power function of the flow rate) is translated into a pressure which is a linear function of the flow. The differential across the orifice tilts a U-tube and thereby simultaneously increases the pressure in a relay nozzle N. The pressure thus produced is transmitted to one leg of the U-tube and thereby causes the liquid level to change by an amount h.

As the counteracting or feedback moment is proportional to h times the distance of the displacement center of gravity $(c_1 \cdot h)$ the moment becomes equal to $(c_1 \times h^2)$, or, when in the balanced condition, h becomes directly proportional to q. The device thus takes the square root out of the flow signal. Reversing the design, it is possible to produce an h_2 which is porportional to the square of h_1 (Republic Flow Meters Company's "modulator").

Another company (Hagan Corporation) introduces the additional parameter s, a stroke. The differential pressure acts upon a diaphragm, Fig. 8(*c*), thus creating a force directly proportional to the square of the flow rate q. This force is transmitted to the outside through a stuffing box (diaphragm type) and controls a relay valve which produces a pressure on top of a spring-opposed diaphragm. For every pressure on the diaphragm, a stroke is obtained which is modified by a cam, and reconverted into a force which finally opposes the primary-flow-rate signal. Thus a feedback establishes a balance between forces and produces a pressure p which, by use of a proper cam, can be made directly proportional to q.

A ratio adjuster working on the force-vector principle allows a change of the proportionality factor. The translation is thus accomplished in steps following the sequence:

From flow energy to differential pressure to $\text{force}_{(I)}$ to stroke; to a modified stroke to a $\text{force}_{(II)}$, with a feedback circuit between $\text{force}_{(I)}$ and $\text{force}_{(II)}$.

We will find such translators in series in most of our multiple-fuel-control solutions. Again the device can be reversed, with slight changes, to produce a differential h^2 for any given p or h.

In Fig. 8(*d*), the same problem is solved (by Bailey Meters Company) with the use of displacement bodies and floating inverted bells.

The translation follows the lines:

Energy flow—differential pressure—$\text{force}_{(I)}$—stroke—$\text{force}_{(II)}$ (counteracting buoyancy). Feedback is established between $\text{force}_{(I)}$ and $\text{force}_{(II)}$ with the result that the stroke s becomes directly proportional to q, the flow rate.

Fig. 8(*e*) shows another method (Askania Regulator Company) in which the translation is obtained by means of a force feedback, represented by a spring counteracting the diaphragm force. By the use of an adjustable cam, which can be easily adjusted in the field, any relationship between h and s can be produced. This is particularly advantageous if the flow signals, as frequently happens, do not follow second power or any other predictable or simple laws.

We recall that the reason for the existence of all these translating devices is that the square of the sum is not equal to the sum

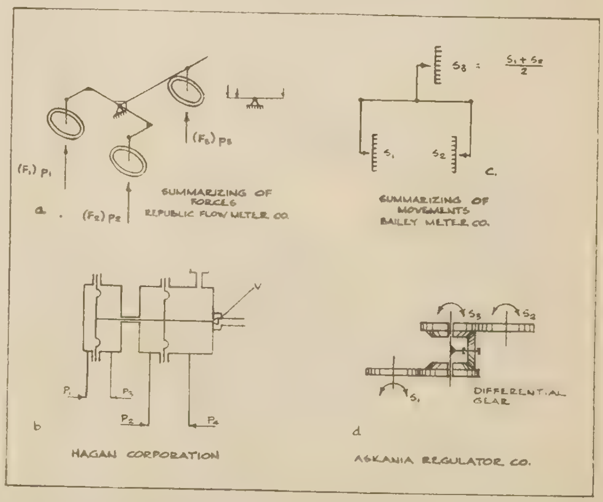

Fig. 9 Various Summarizing Devices for Forces *a* and *b* and Movements *c* and *d*

of the squares, and that directly proportional variables must be obtained for summarization.

Means for Summarization of Signals Representing Flow Rates

Before describing the individual basic solutions of the multiple-fuel problem as they are now available to the steel industry, we shall discuss briefly some typical summarizing devices for pressures and strokes as they are found in many various controllers. As it is impossible to summarize pressures directly, a detour becomes necessary.

Fig. 9(a) uses for that purpose a forked lever with pressures p_1 and p_2 applied to individual diaphragms. A pilot (not shown) produces a pressure p_3, which is used as a feedback into a third diaphragm, producing a balancing force F_3. Then F_3 becomes the sum of $(F_1 \cdot c_1 + F_2 \cdot c_2)$ with c_1 and c_2 as constants (Republic Flow Meters Company). Basically, the same idea is used in design Fig. 9(b), in which two or three pressures are added or subtracted by means of diaphragms in series. Pressure p_4 is the resulting pressure counteracting $(c_1 \cdot p_1 + c_2 \cdot p_2 - c_3 \cdot p_3)$ and controlled by a bleed-off valve v (Hagan Corporation).

In the class of the stroke summarizers, we find the whiffletree, Fig. 9(c) (Bailey Meter Company), and the differential gearbox Fig. 9(d) (Askania Regulator Company). Both of these devices are too well known to need any further explanation.

Methods and Devices for Combustion Controls Using Several Fuels but Only One Air Supply

Thus prepared, we can finally analyze the various designs of actual controllers used in multiple-fuel controls.

We have seen that producing a proportional pilot-flow model offers a particularly flexible solution. In Fig. 10 (Leeds & Northrup Company), we note that the summarization is accomplished by the addition of two (or more) currents which are directly proportional to the individual air requirements.

New in this picture is the use of a positive-displacement meter for measuring the fuel-oil rate of flow by means of a generated voltage directly proportional to the speed of the meter. The generated voltage is balanced by a potentiometer controller against the voltage due to the current (i_2 current) flow in the pilot circuit. The potentiometer adjusts the rheostat to maintain this voltage. This current is added to that produced by the Kelvin bridge. The square-root symbol indicates the linearity of the controlled strokes. No special summarizing devices are necessary,

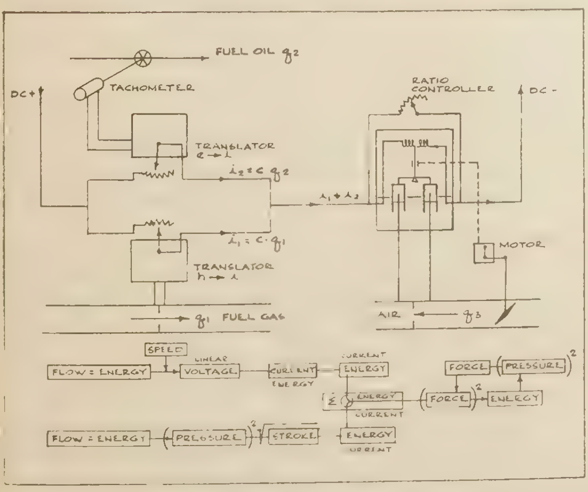

Fig. 10 Multiple-Fuel Control; Leeds & Northrup Company

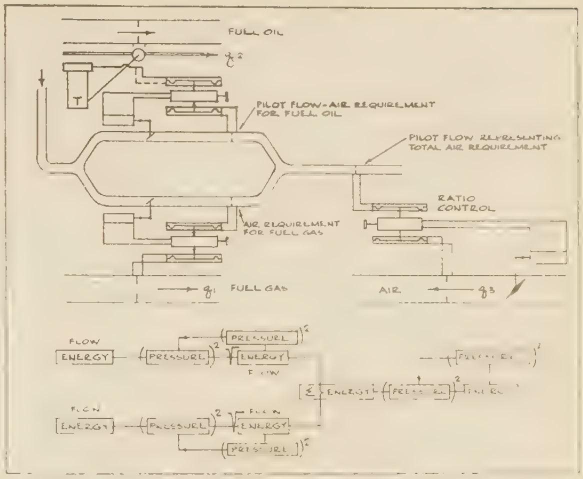

Fig. 11 Multiple-Fuel Control; Askania Regulator Company

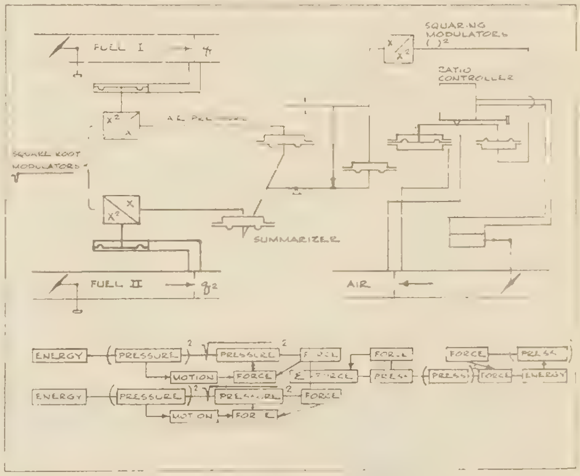

Fig. 12 Multiple-Fuel Control; Republic Flow Meters Company

as the currents flow together in accordance with Kirchhoff's law. The sum total produces a force proportional to i^2 and controls the flow of air needed for combustion.

The ratio controller is basically the Kelvin balance in reverse. The air-flow energy produces a second-power pressure, which, translated into a force, is fed back against the current force.

A ratio adjuster in the form of a rheostat permits the adjustment of the ratio from excess to deficiency.

An attempt is made in Fig. 10 and the following diagrams to present the individual steps of translations in a block diagram.

For want of a definite standard way of using symbols for such translators these diagrams are tentative only and should be modified as soon as a general agreement on these symbols has been or can be reached. For the purpose of the present paper, however, these diagrams may be sufficiently clear to demonstrate the approach to the circuit analysis, although it would be desirable to develop a more universally acceptable way of presentation.

Fig. 11 is basically the same circuit with the exception that fluid pilot flows are used instead of the electrical currents (Askania Regulator Company).

Of interest is the possibility of using alternative fuels by the application of the device which translates the speed of the posi-

tive-displacement oil meter into square-function air pressures. By matching the fluid pressure produced by this device to that obtained from an orifice in the alternate fuel line in such a manner that it represents the same air requirement, the pneumatic tachometer T can be switched off, and its signal can be replaced by the gas differential without any changes in the circuit.

For convenience, the orifices in the pilot lines are adjustable, thus greatly facilitating adjustments for varying operating conditions.

As the two pilot flows can be easily combined and the totalized flow produces a square function across the totalizer orifice in the common conduit, no additional translating devices are needed. Feedback is established by balancing pressure differentials, or to be more precise, forces.

In Fig. 12, the summarization is accomplished by adding pressures or forces which are directly proportional to the flow rates (Republic Flow Meters Company). To do this, "modulators" converting square functions into linear functions (symbol x^2/x) are used, and summarizers as shown in Fig. 9(a). The sum is "modulated" into a square function (symbol x/x^2) and applied to a ratio regulator which controls the total combustion air. The air differential is fed back into the ratio control to re-establish balance.

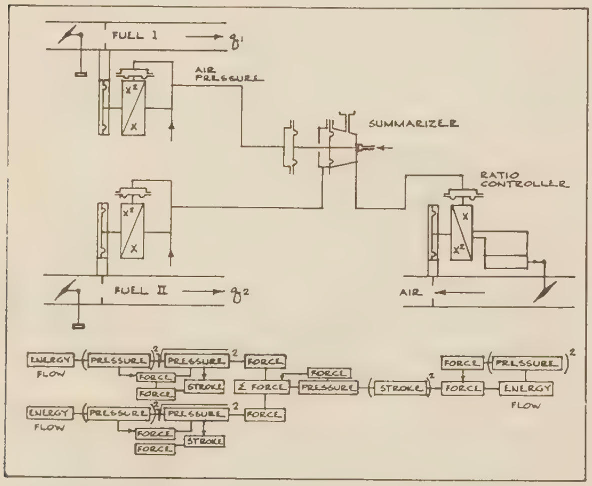

FIG. 13 MULTIPLE-FUEL CONTROL; HAGAN CORPORATION

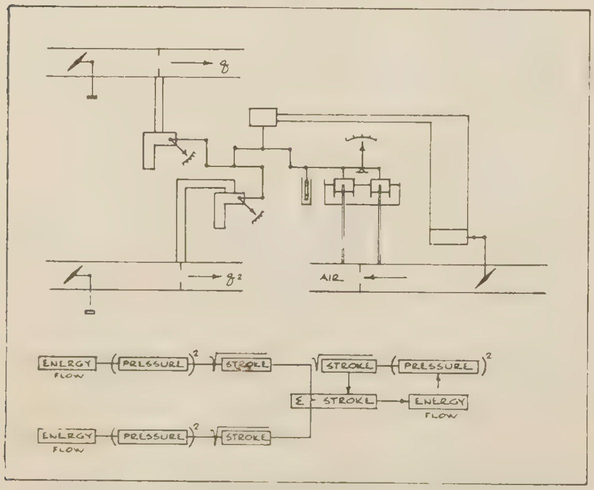

FIG. 14 MULTIPLE-FUEL CONTROL; BAILEY METER COMPANY

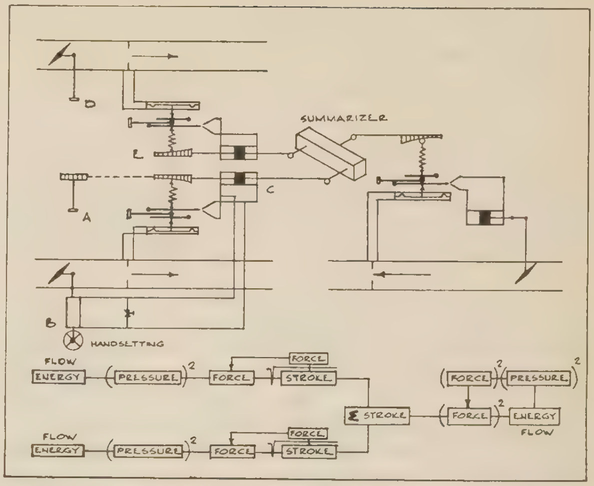

FIG. 15 MULTIPLE-FUEL CONTROL; ASKANIA REGULATOR COMPANY

Basically, the same line of thinking is found in the solution of Fig. 13 (Hagan Corporation), although, as previously shown, the details of the modulator design are different. For the purpose of clarity the same box symbols for the modulators are shown as in Fig. 12.

In Fig. 14, the summarization is accomplished by means of the whiffletree shown in Fig. 9. Individual flowmeters add their pointer movements (without additional power amplifiers) and produce linear movements by suitable float arrangements (stroke modulators) or displacement bodies (force modulators) (Bailey Meter Company). The diagram is self-explanatory. Feedback is accomplished by comparing movements of "strokes."

The last example, Fig. 15, shows a stroke feedback with power amplification (Askania Regulator Company). The modulator, previously discussed in connection with Fig. 8(e), will be recognized.

The diagram shows an alternative adjustment of A or B. If the cam is set by hand, adjustment A, the butterfly valve is controlled, and thus the fuel flow becomes constant, regardless of any variations in the fuel-supply line. If B is set by hand, the flow regulator acts as a "flowmeter." Adjustable cams permit not only correction for second-power flow signals, but permit also:

(a) Correct compensation for any air infiltration (either constant or varying with the load).

(b) The varying of setting of excess and deficiencies of air for different heat inputs.

How this same principle can be applied to wide-range flow-mixing or combustion problems is shown in Fig. 16. With the same number of controls as in Fig. 15, and variable orifices, any turndown range can be handled. In this case again strokes are used as the common parameters, strokes are summarized, and strokes are used in the feedback system.

With these tools available, we are finally in a position to solve practically all problems of multiple-fuel controls. As most of the applications have been previously discussed in papers given by the individual manufacturers, a detailed discussion would only be a repetition of available information.

MAKE-UP FUEL STARTED AT A PREDETERMINED RATE

Recently another problem has become more and more important, i.e., the need for starting the secondary fuel at some minimum flow rate, as there is usually a certain minimum rate below which a burner cannot operate satisfactorily. If a deficiency of pri-

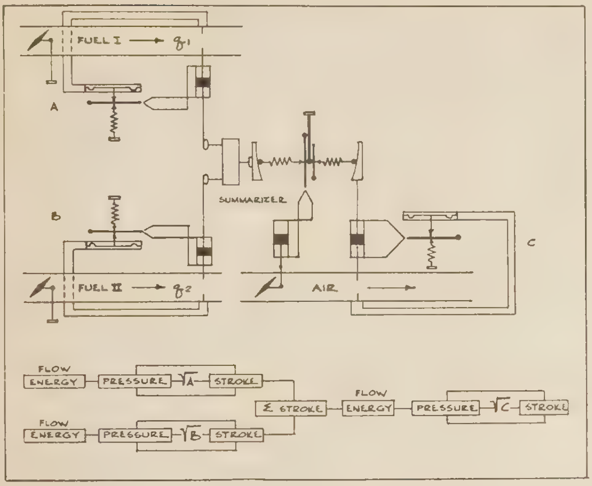

Fig. 16 Wide-Range Multiple-Fuel Control; Askania Regulator Company

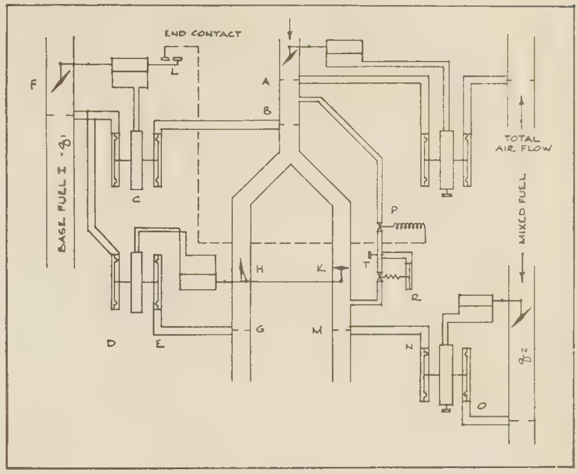

Fig. 17 Multiple-Fuel Control With Minimum Setting of Secondary Fuel

mary fuel therefore demands a start of secondary fuel, means have to be found to start the secondary-fuel flow at once at this minimum rate, and simultaneously to reduce the primary-fuel flow by an equivalent amount of Btu supply.

Fig. 17 shows one possible solution which uses pilot flows to represent air flow or Btu rates per time unit (Askania Regulator Company). The total air flow, or the rate of total Btu supply, is replaced by a pilot flow which branches into two lines representing the Btu rates of the individual fuels.

An orifice in the main pilot line *B* produces a signal representing a certain amount of Btu of fuel I (q_1). As long as ratio regulator *C* can satisfy this demand, and controls butterfly *F*, the differential acting on *D* is equal or slightly greater than that produced by *G*. Under these conditions, butterfly *H* is open and butterfly *K* closed.

Both butterflies are operating in opposite directions *H*, opening when *K* closes and vice versa.

If there is a deficiency of the primary fuel, butterfly *F* opens wide in an attempt to satisfy the demand, and thus closes contacts *L*, which energize a solenoid valve *P*, and thereby a passage for bleeding a definite amount of pilot flow from a point between *A* and *B* to a point ahead of *M*, which determines the rate of fuel demand for the secondary fuel.

This fuel rate is controlled by means of the ratio regulator *N*–*O*. A differential control valve in *R* the by-pass, and an adjustable restriction *T*, assure a definite rate of flow which represents the minimum flow rate for the secondary-fuel burner.

It will be noted that the diversion of flow from the left branch to the right decreases the loading on ratio control *C*. Butterfly *F* therefore reduces the flow rate (q_1) by at least the amount of the by-passed pilot flow, and ratio control *D*–*E* cuts down the pilot flow through *H* and increases the flow through *K* so that the difference is always diverted to orifice *M*. The design is intentionally so arranged that the secondary-fuel burners remain in operation until an operator manually disconnects them from the line. All that is necessary for this purpose is to de-energize solenoid *P*, which incidently can also be done automatically.

It appears that all of the foregoing solutions offer a great help to the operator and simplify his job of operating a furnace.

The question which is still open, and which this paper did not attempt to cover, is: What is the best setting of these calculating and controlling devices assuming optimum operating conditions?

We must not forget that the equipment, even if its performance is beyond reproach, is just as capable of maintaining the wrong operating conditions as it is of maintaining the correct ones.

Acknowledgment

The author is indebted for assistance in the preparation of the paper to Messrs. M. J. Boho, Hagan Corporation; C. J. Geiser, Republic Flow Meters Co.; J. C. Peters, Leeds & Northrup Company, H. E. Weaver, Bailey Meter Company, and P. W. Keppler, Foxboro Co.

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Discussion

C. J. Geiser.[2] The author has contributed a commendable paper, presenting the general problems of ratio and multiple-fuel controls in the steel industry and a brief description of the various designs of systems used to solve these problems.

The problem of uncontrolled air infiltration has always presented difficulties on actual applications. Limitations of burners at low rates of firing and limitations as to the minimum amount of combustion air (cold or preheated) required at the burner at low rates of firing for proper combustion will necessarily limit the extent to which full compensation can be provided. Each individual furnace usually presents an individual problem in this respect requiring extensive test work to determine optimum settings

[2] Republic Flow Meters Company, Chicago, Ill.

for various firing rates and for various phases of the furnace cycle.

The method of compensation described by the author, and illustrated in Figs. 5 and 6, is useful for obtaining a close approximation of theoretically correct total combustion air for the limited firing range specified. Whether or not a furnace will give its best performance with theoretically correct total combustion air (full air-infiltration compensation) at the lower rates of firing, say between 50 and 30 per cent of maximum, can be determined only by actual tests. Due to other considerations, partial compensation (illustrated by the 10 per cent curve of Fig. 6) may possibly give superior over-all results.

It is obvious that higher percentages of air infiltration adversely affect the range of operation of fuel-air ratio control and the attempts to compensate for the infiltration. For example, as an extreme condition, if the air infiltration is equal to 50 per cent of the controlled maximum air flow, the ratio regulator could not possibly control the first $33^1/_3$ per cent of total air flow. Also, using the method of compensation illustrated in Fig. 6, the fuel-flow range would be reduced to a range of approximately 60 per cent to 100 per cent.

The means of compensation shown in Fig. 7 of the paper provides a method for obtaining the correct characteristic for a constant quantity of air infiltration for a particular setting of variable S. It may not be immediately apparent that another means of obtaining exactly the same characteristic is available on the design shown in Fig. 12. Having established an air pressure which is directly proportional and linear with respect to fuel-flow (or total fuel flow on the multiple-fuel control), spring or weight loading of an air-pressure relay (or the summarizer) will exactly compensate for a constant quantity of air infiltration.

Generally, the application of ratio and multiple-fuel controls, particularly on fuel-fired furnace in the steel industry, might be divided into two very broad divisions:

1 The applications on newly designed and built furnaces.
2 The applications on outmoded, existing furnaces.

New furnaces which are carefully designed to minimize air infiltration and to accommodate a wide range of firing, will unquestionably prove the most easily adaptable to the application of these controls. However, the possibilities of applications to the older, existing equipment cannot be overlooked, merely because these applications may introduce difficult problems of maintaining the required combustion analysis due to excessive air infiltration or other limitations. While the older equipment may introduce greater problems for successful application, careful attention by the plant combustion personnel in applying the control, to obtain optimum control settings, will show a great improvement in over-all furnace performance compared to uncontrolled operation.

The vast majority of ratio and similar controls, applied to the steel industry, are entirely dependent upon the judgment of combustion engineers (or other authorized operators) to determine and set the values of pressure, flow, ratio, etc., which the control will maintain throughout the various cycles of the process. As pointed out by the author (and possibly this point cannot be overemphasized), "the control is just as capable of maintaining the wrong operating conditions as it is of maintaining the correct ones." The personnel factor, therefore, enters into the over-all application problem to a large degree and should be given very serious consideration when determining the features which are to be incorporated into any particular design of these controls for a particular application. Considering the physical condition and other limitations of older, existing mill equipment, as well as the deep-rooted precedents which have been established during a long period of use by the operators, the greatest care and attention are required for the successful application on old furnaces.

Summarizing the features of combustion-control equipment which is today available to progressive combustion engineers:

1 Standard fuel-air ratio controls may be compensated for air infiltration, within limits.

2 An almost unlimited number of control-characteristic combinations are available to permit addition and subtraction, multiplication and division of control factors, with designs that can be readily tailor-made to fit the exact requirements of a single application.

3 The extent of control refinements possible is limited in most cases only by the ability of the plant personnel to use the control equipment successfully.

Indexes to A.S.M.E. Papers and Publications

THIS and the following pages will serve as a guide to the current publications of the A.S.M.E.

Regular Society Publications, 1944

Mechanical Engineering, monthly (see index on page RI-55)
A.S.M.E. Transactions, monthly including the *Journal of Applied Mechanics* (see index on page RI-63)
A.S.M.E. Mechanical Catalog and Directory, 1945 edition.

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Design Data on Mechanics
1942 Automotive and Oil Engine Power Cost Reports

American Standards

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Cast Iron Pipe Flanges and Flanged Fittings, Class 250

API-ASME Code for Unfired Pressure Vessels

1944 Supplement

Boiler Construction Code

1944 Addenda to:
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THE technical programs of the meetings of the Society and of its Professional Divisions have been published in *Mechanical Engineering* and may be located by consulting the index on pages RI-55–RI-62. A majority of these papers were published, or will be published, in *Mechanical Engineering* or the Transactions (including the *Journal of Applied Mechanics*) and may be located by reference to the indexes of these publications. Several additional papers and reports included in these 1944 programs were not published during the year in Transactions or *Mechanical Engineering* but were issued in mimeographed or photo-offset form.

Complete sets of these are on file for reference purposes at the office of the Society and the Engineering Societies Library, under the title of "Miscellaneous Papers Presented at A.S.M.E. Meetings, 1944." Photostat copies of any of the papers may be secured from the Library at twenty-five cents a page to members, or thirty cents a page to nonmembers.

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